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# ANALYSIS OF HEAT TRANSFER AND OVERHEATING FAILURES ASSOCIATED WITH WALL TEMPERATURES IN STEAM SUPERHEATERS

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***Abstract.*** *This paper is focused on the heat transfer of the steam superheaters into a coal-fired power plant already existing. Theoretical results as well as measured temperatures have been carried out. High temperature probes were developed and installed for monitoring the metal temperatures of the superheaters. The experimental results are also compared with numerical results to check the thermal behavior of the probe themselves, once they are subjected at high temperatures. The results so far obtained with field probe indicate temperatures slightly higher than the expected values of the order of 40 to 60°C. This difference are quite small when the characteristics of employee thermocouple and unfavorable conditions of the measurement site are concerning, however significantly large if consider that the purpose of the work is controlling the real temperature allowed by the material selected for superheater and reheaters steam tubes.*

***Keywords:*** *Steam power plant, coal-fired boiler, steam superheaters, overheating failures, heat transfer*

## 1. INTRODUCTION

Boiler tube failures are leading cause of forced outages in coal-fired plants. Superheater and reheater tube failures play an important role in forced stopping of large steam generating system, mostly due to high temperatures. The major problem is associated with the progressive degradation of pipe material, involving graphitization alloy, the material creep, and thermal fatigue in critical regions of the superheater. For all materials used in steam superheaters, the strength decreases as temperature increases. The boilers tubes must be maintained within design parameters and for this a balancing is required within the design limits. So tubes are heated by the flame or hot flue gases and simultaneously cooled by water or steam flow. Evidently other causes should be considered. Dominguez et al (2008) reported a metallurgical analysis of failed tubes showing that overheating of material, high temperature oxidation, bulging, cracks and creep degradation are also result of internal deposits on tube wall. To determine the root cause, a comprehensive assessment of actual boiler conditions was carried out, considering operation variables and procedures, maintenance and inspection programs, combustion and control system modifications.

A substantial number of technical reports and papers point to such problems, both in old boilers and in modern boilers designed to operate at supercritical pressures and steam temperatures above 550 ° C. As previously stated by French (2002) the operation of a boiler is a dynamic balance between heat flow from the combustion of a suitable fuel and either steam formation within the furnace or steam super heating within the superheater or reheater. For all materials used in boiler construction, the strength decreases as temperature increases. Operational problems can lead to different short-term high-temperature failures, either in the waterwall as in the superheater and reheater.

Thielsch and Cone (2002) reported details of a number of failures that have occurred in superheater and reheater outlet headers. About one-half of these failures resulted in forced outages. The other half were discovered during scheduled outage inspections. The paper discusses the methods used to repair those headers, on occasion on a temporary basis. It also discusses the various inspection techniques which, if utilized as part of a routine inspection program, are capable of identifying conditions likely to result in failures.

Díez et al (2010) showed semiempirical model simulation of the process of heat transfer to the superheater units intended for large thermoelectric power generation with pulverized coal. The obtained results demonstrate the possibility of reducing calculation errors combining the use of conventional methods with advanced CFD, particularly when one wants to evaluate the process of heat transfer in steam superheaters and reheaters. The load variation has also been considered a key factor for the variation of the wall temperature by some authors.

Yanzhi et al. (2010a, 2010b) presented a method of measuring the wall temperature of up to 600MW supercritical boiler, in order to investigate the factors which influence the rise in temperature which often leads to failures in superheaters and reheaters. The authors measure a difference of up to 130 ° C above the temperature of superheated steam, where the load variation was a key factor.

Rahman and Kadir (2011) reported a failure analysis of failed high temperature superheater tubes of an existing PC fired power plant. Collected failed samples were undergone several experimental investigations including visual inspection, thickness measurement, Vickers hardness testing, and microstructure evaluation. They also state that failures of superheater tubes occur most frequently.

It is evident the difficulty and need to solve such problems and even identify critical regions that may be monitored by installing high temperature sensors, in order to reduce the power plant outages. The commercially available computational tools, such as CFD (Computational Fluid Dynamics), does not satisfactorily resolve the problem because of the limitation of the models, mainly associated to the combustion process and mechanisms of direct radiation and also in absorbing and emitting medium. It is also notorious the difficulty associated with uncertainties related to input data, that are approximated by the design engineer, compromising the results found from the simulation of the process of heat transfer. There is an actual need for validation of the results obtained using the instrumentation and accurate measurement of the temperatures of the tube wall and the combustion gases.

With this concern, financial resources provided by development agencies and public and private companies in the energy sector has ensured the development of research also in Brazil. Recently, the possibility of access to a boiler in operation opened the way for the development and installation of sensors shielded, fitted with type K thermocouples for monitoring the thermal behavior of superheater and reheater at different loads of operation of the burners in the combustion chamber (Silva et al, 2008 and Bazzo, 2010). To ensure reliability, experimental results obtained in the laboratory were compared with numerical results in order to verify also the thermal behavior of the sensors themselves, when subjected to high temperature environments. The results were not definitive because of the uncertainties associated with models of combustion as well as the direct and indirect thermal radiation of combustion chamber due to the presence of gases participants. Also, due to the uncertainties associated with boundary conditions and not always known properties of coal used and materials used to manufacture the boiler. However, the availability of such results contributed to the identification of critical regions of the superheater, where there are high wall temperatures as a consequence of higher flue gas velocities. Naturally, it is imperative to validate the results obtained using the instrumentation and accurate measurement of the temperatures of the tube wall and the combustion gases. With the ability to identify critical areas that can be monitored by installing sensors in high temperatures and even expecting to permanently resolve such problems, it is expected to significantly reduce outages caused by thermal gradients in the power plant. In the course of the work, however, it became clear that the readings of the temperature did not correspond with the real values, as previously observed in sensors available on the market today. So, new sensors were assembled, with new features in order to minimize uncertainties associated with readings taken and installation problems. The sensor is comprised of a base to support the pipe wall, a protective thermal radiation from the combustion gases to a guide wire access compensation, and an upper cover, providing the necessary insulation to the thermocouple. Theoretical results have indicated divergent values compared to the measured values, suggesting further studies in order to adapt the boundary conditions that best represent the real problem. Depending on boundary conditions and simplifying assumptions, the thermal resistance of the fluid and the pipe wall may be small and may lead to wall temperatures approximately equal to the internal fluid. The results depend on the flow velocity, and material properties considered temperature of the steam and hot gases.

A complete simulation of heat transfer to the utility boiler could be undertaken as a set of heat exchangers. This paper is focused on the heat transfer of steam superheaters in an existing PC power plant. A semi-empirical model is proposed in order to calculate the wall temperature of the tubes of a superheater located in the critical passage of the combustion gases of the boiler in operation. A sensitivity analysis was also carried out in order to determine the influence of the temperature and velocity of the hot gases, as well as the effect of oxide layer thickness over the wall temperature. Experimental data were collected to for comparison with theoretical results.

## 2. MODELLING OF THE STEAM SUPERHEATER

The steam generator considered in the analysis is a front coal-fired unit of natural circulation designed with a nominal continuous rating of about 100 kg/s, 130 bar of pressure and 520°C of outlet steam temperature. The fuel is low ranking powered coal. The steam generator has three superheaters and two reheaters. The focus of this paper is the superheater 3.

A conventional lumped model for heat exchangers between two streams is first considered, according to the following equations:

$$q = \dot{m}_g\left(h_{g,i} - h_{g,o}\right) = \dot{m}_s\left(h_{s,o} - h_{s,i}\right) \qquad (01)$$

$$q = UA \cdot F \, \frac{\left(T_{g,i} - T_{s,o}\right) - \left(T_{g,o} - T_{s,i}\right)}{ln \dfrac{\left(T_{g,i} - T_{s,o}\right)}{\left(T_{g,o} - T_{s,i}\right)}} \qquad (02)$$

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where Eq. (1) represents the thermal balance and Eq. (2) the heat transfer from hot stream (gases) to the cold stream (steam). The factor F in Eq. (2) depends on the geometry and complexity of the heat exchanger (Pignotti and Cordero, 1983 apud Díez et al, 2010). Here it was considered close to unity, due to the high number of tube passes. The coefficient *UA* is modeled as:

$$UA = \left[ \frac{1}{A_s \cdot h_{c,s}} + \frac{1}{A_g (h_{c,g} + h_{r,g} + h_{dr,g})} + \frac{ln\left(\frac{D}{D_i}\right)}{2\pi \cdot L \cdot k_t} + \frac{ln\left(\frac{D_i}{D_i - 2e_{ox}}\right)}{2\pi \cdot L \cdot k_{ox}} \right]^{-1} \quad (03)$$

where $h$ means the heat transfer coefficient (W/m$^2$K). Thermal resistance due to fouling in water/steam side has been considered as well. Convective coefficients for gases $h_{c,g}$ and steam $h_{c,s}$ were estimated considering empirical correlations available in the literature (Bazzo, 1996; Incropera et al, 2010). Concerning in-tube thermal radiation, the apparent coefficient was calculate as follows,

$$h_{r,g} = \frac{q_{r,g}}{A_g \cdot \Delta T_m} \quad (04)$$

where $\Delta T_m$ is the mean temperature. The coefficient $h_{r,g}$ depends on gas temperature and wall temperature, thus an iterative procedure was adopted. As stated by Hausen, apud Díez et al (2010), to select an arithmetic mean for gas temperature between inlet $T_{g,i}$ and outlet $T_{g,o}$ does not seem correct, since radiative transfer depends on the fourth power of temperature. Hausen (1983) recommends an empirical correction in case of significant gas temperature variations through the heat exchanger. In this work, considering both methods, the final result differs less than 1%. The radiative heat transfer from the gases was modeled by the isothermal approach, also provided in the basic literature:

$$q_{r,g} = 5.67 \cdot 10^{-8} A \cdot f \cdot (\varepsilon_g \cdot T_g^4 - \alpha_g \cdot T_w^4) \quad (05)$$

where $A$ is the heat transfer area, $f$ is the gray-body correction factor, $\varepsilon_g$ is the gas emissivity and alfag is the gas absorvity.

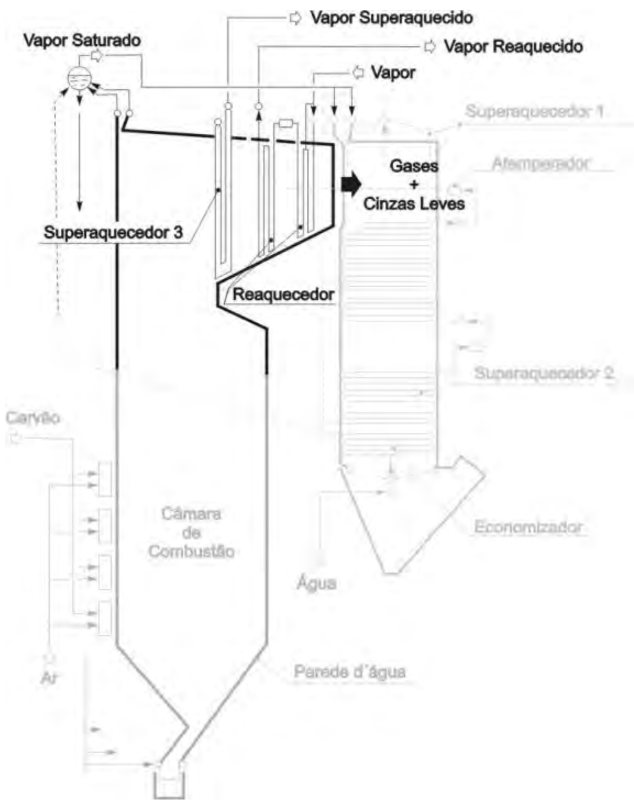


Figure 1- Steam generator scheme

In the case-study, as shown in Fig. 1, although is partially hidden by the nose, the superheater 3 “sees” partially the flame and so is exposed to a significant directed radiation. The classical treatment for this question is

$$q_{r,d} = \dot{m}_s \cdot (h_{s,o} - h_{s,i}) - q_{c,g} - q_{r,g} \quad (06)$$

All thermodynamic calculations and simulations of the process of heat transfer in the boiler were performed on the platform (EES Engineering Equation Solver). All properties of steam and hot gases were also calculated on the platform ESS according to the corresponding temperatures. The available heat in the furnace was calculated considering the Lower Calorific Value of coal used in the boiler (LHV = 17,473 kJ/kg) and inlet temperature of the combustion air equal to 300 ºC, according to data measured in the plant. An amount equivalent to 3% of the heat available as losses through the walls to the environment, purges and ash removed through the bottom of the furnace. The coal burned has 37% ash, of which the percentage equivalent to 30% falls as bottom ash in the furnace, while others 70% as fly ash are entrained by the combustion gases. Still, according to company data, the presence of unburned in the ashtray varies between 2 and 4% of the material falling into the water seal. In this work it was admitted the value of 3%.

Combustion gases were calculated according to the stoichiometric ratios for solid fuels and 25% excess air, corresponding to 14.4%vol CO2 on a dry basis at the base of the chimney.

The calculations were carried out considering the boiler operating at rated load (100 kg/s, 130 bar and 520°C). The following simplifying assumptions were considered:

(i) Steady state operation;
(ii) Gas velocities leaving the furnace uniform and equal to 7 m/s;
(iii) Average temperature of the gases leaving the furnace equals to1084°C;
(iv) Height effective at the inlet region of the superheater equivalent to 8 meters;
(v) Superheated steam uniformly distributed in the superheater panels;
(vi) Steam temperature at inlet superheater constant and equal to 440°C;
(vii) Oxide layer thickness up to 1 mm.

The assumptions (ii), (iii) and (iv) were established following numerical results of CFD simulation, as well as mass and balances energy applied to the steam generating unit, including the waterwalls, superheaters, reheaters and economizer. The hypothesis (iv) takes into account only the effective area for heat exchange from hot gases to steam, excluding the region corresponding to the partial recirculation of the gases at the bottom of the superheater.

According to a technical report presented by Bernardini et al. (2000), oxide layers thickness ranging from 0.51 (cold side) to 0.93 mm (hot side) were identified along the steam reheater. Certainly the presence of oxidized layer is also significant in the superheater and was considered in the analysis.

Usually lumped models do not account for possible effects of irregular gas velocity profiles. The presence of vortex and reversed-flows is common in large steam generators, especially around the nose. Although this question is often ignored, the thermal balance is exact if an adequate value for gas average temperature is assumed. The solution to this problem by means of simple models is difficult and has not been yet attained. Anyway, the uncertainties arise in estimating convective coefficients need to be corrected accounting for inlet effects. Today, a more accurate method to deal with this problem is based on CFD techniques. The physical model considered here for fluid flow analysis considers the geometry of the superheater immersed in a medium with the same characteristics gas boiler, as shown in Fig. 2. The calculation domain for solution flow and heat transfer is defined in the region of passage of the gases at the top of the combustion chamber, including superheater 3 and reheaters. In the simulation model it was considered the RNG k-epsilon turbulence method. Both the superheater and reheaters were modeled as porous elements with equal volume of 99%. The known values of gas flow rate, the heat transferred to the superheater and reheaters (see Fig. 1), as well as imported data from an existing measurement system based on acoustic pyrometry installed at the top of the combustion chamber were considered as boundary conditions. The mass flow rate of hot gases was previously calculated and therefore also considered as input data equal to 165 kg/s. The superheater has 40 panels, each one consisting of seven steel pipe ASTM A213-T22, arranged in line relating to flow of the hot gases.

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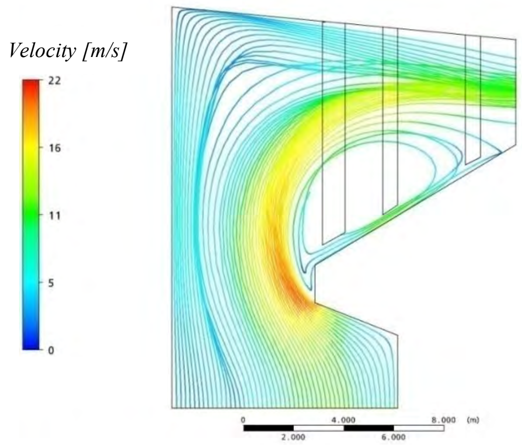


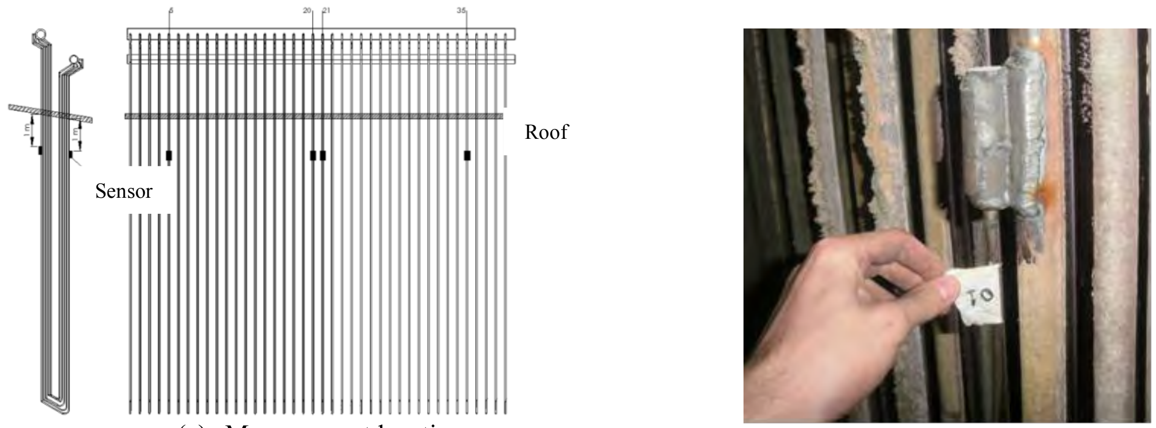

Figure 2. Field of gas velocities in the critical region of the steam generator.

## 3. INPUT DATA AND WALL TEMPERATURE MEASUREMENTS

The mass flow rate of steam pressure and temperature were read directly on the plant, including input and output in the evaporator, superheaters, reheaters and also in economizer. The gas temperatures and volumetric fraction of oxygen after the economizer were also measured directly in the plant. The mass fraction of unburned carbon was measured from samples taken in the ashtray and also in the electrostatic precipitator. All the collected data were first used to calculate the thermal efficiency of the steam generator, and therefore the consumption of coal.

Special attention was reserved for measurements of the wall temperatures in the superheater, the main object of study in this work. The identification of critical locations for probe installation was based on the results obtained through CFD simulation considering the velocity field and gas temperatures in the zone of the steam superheater and reheaters. The data acquisition system consisted basically of a laptop and a data acquisition system for reading and recording the temperatures. The location of the probes followed the results of this simulation, searching critical spots or streamlines as identified in Fig. 2. Eight temperature sensors were installed on the external tubes corresponding to four panels of the superheater 3, as shown in Fig. 3 (a) and (b).

To improve the reliability of the temperature sensors, studies were also performed considering the features of sensor installed on bundle tubes immersed in the hot gases. The problem was solved using a CFD turbulence model SST (Shear Stress Transport) applied either to the side of the gas and to the side of the steam. The results obtained from the simulation are satisfactory, but not conclusive, due to limitations of the software that impede a depth analysis of the process of heat transfer by radiation.




(a) Measurement location (b) Temperature sensor

Figure 3. Wall temperature measurement at the superheater.

It is important to highlight the influence caused by the temperature sensor in the gas flow, especially in the wake downstream of the tube as well as the changing of the gas velocities along the walls of the tube, on the order of 20 m/s in the case of free-flowing and 10 m/s in the presence of the sensor. This difference must inevitably lead to a reduction in the heat exchange of gases to the tube wall by convection, by the local measurement. On the other hand, the presence of the temperature sensor acts as extended surface, intensifying the heat transfer from the gases to the tube wall by conduction. Figure 4 (a) and Figure 4 (b) show respectively the velocity and temperature field in the transverse plane of the tube considering the scenario with the sensor installed.

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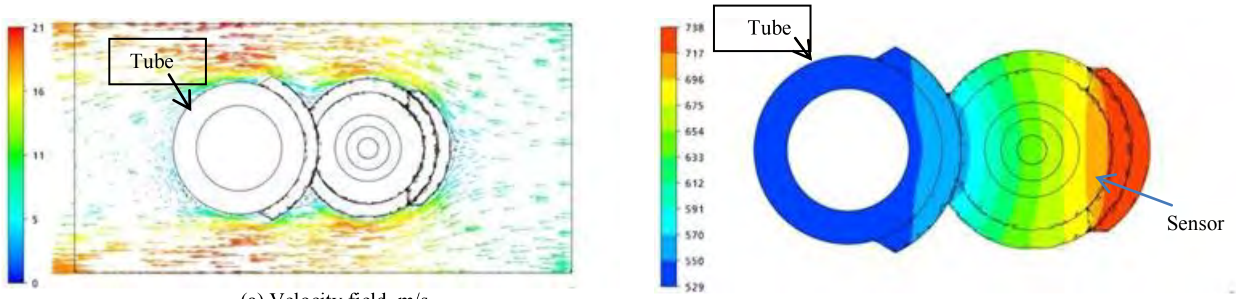


(a) Velocity field, m/s

(b) Temperature field, ºC

Figure 4. Scenario of velocity and corresponding temperature fields around the temperature sensor

At the entrance of the superheater, the flow of hot gases were considered perpendicular to the sensor, with variable speeds in the range of 6 to 8 m/s, corresponding to a mass flow rate 0.020 kg/s at pressure of 101 kPa and an average temperature of 1200°C. Inside the tube, the mass flow rate of steam was considered equal to 0.390 kg/s, corresponding to speeds of about 24 m/s. At normal operation (124 bar and 520°C) the mass steam flow, corresponding to 109 kg/s, was considered uniformly distributed along 280 tubes of the superheater. Under these conditions the thermal power superheater is 30 MW. The outer diameter of the tube is 31.8 mm, with a wall thickness of 5.6 mm. The material considered for the tube wall and also the wall of the temperature sensor was 310 stainless steel, with a coefficient of thermal conductivity of 60.1 W/mK. For the ceramic fiber used as insulation it was considered the thermal conductivity coefficient of 0.04 W/mK.

According to the results, in the absence of the sensor, the temperature gradients in the tube wall correspond to values in the range from 529 to 550°C, at most 35°C above of the steam temperature considered as boundary condition. After the installation of the sensor, significant change is observed in both the thermal gradient, as well as the maximum temperature of the tube wall, in this case in the range from 529 to 570°C, so 14-55°C above the steam temperature. Here, for simplicity, it was not considered ash deposits around the outer surface and even oxide layer on the inner wall of the tube. The presence of ash on the walls of the superheater is also an object of study, but is not being considered in this work.

The sensors used were calibrated following recommendations set out in the available literature (ASTM STP 470A, 1974; Kerlin and Shepard, 1982; Nicholas and White, 2001). But regardless of calibration, it has been also imperative to correct the value indicated by the sensor, once the reading is inevitably higher than the actual temperature, depending on the geometry and material used in manufacturing the sensor. According to results, it is also important to note that the tip of the thermocouple itself can also result in significant differences between the reading and the actual temperature of the tube wall. In this particular case, such tip has been located in the region between 550 and 570°C. The highest temperature was identified on the sensor, with values in the range 768-795°C. The cover of the shield is subjected to temperatures of 714°C to 741°C. The results also show that where the sensor is attached is subjected to a higher temperature, ratifying the hypothesis of overheating referred to above. However, in case the superheater under the set conditions, assuming the steam temperature equal to 515°C, the tube wall temperature does not exceed 570°C. In the absence of the sensor, the same value does not exceed 550ºC.

The measurements were performed with eight sensors (numbered 201 to 208), monitoring the external tubes corresponding to panels 5, 20, 21 and 35 of the superheater, at the distance of 1 meter from the roof of the furnace. As shown in Fig. 5, after 200 minutes, the measured temperatures converged to values in the range from 580 to 660°C, which is higher than the initial expectation. One of the thermocouples did not respond positively, indicating the possibility of cable rupture during starting of the boiler. Considering the results obtained from the numerical simulation of the heat transfer process in the field of the sensor, the expected temperature should not exceed 570°C, but correspond to the values calculated when considering the presence of oxide layer thicknesses in the range 0.50 to 1 mm. However other factors not identified need to be investigated, in view of the uncertainties associated with the effect of the sensor as extended surface, increasing the heat transfer from hot gases to the measurement site.

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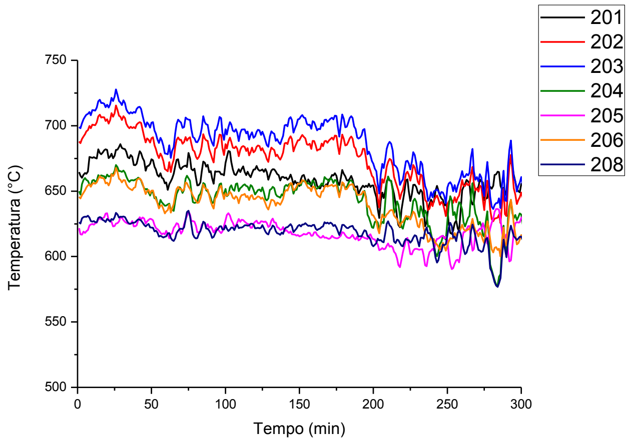


Figure 5- Wall temperatures measured in the superheater.

## 4. RESULTS AND DISCUSSION

Considering now the application of the semi-empirical correlations presented in Chapter 2, but considering the calculation domain effectively as a tube bundle, the wall temperature corresponded to 545ºC, again lower than the temperatures measured by the sensors. On the side of the combustion gases, to calculate the wall temperatures were considered the thermal resistances corresponding to the convective heat exchanged using the correlations proposed by Zukauskas, apud Incropera et al (2010), heat transferred partially by direct radiation from the flame, and heat transferred by gas radiation ($CO_2$ and $H_2O$), following the methodology proposed by Hottel, apud Incropera et al (2010). The portion relating to the direct radiation was calculated by the difference between the total heat received by the steam and the sum of the heat provided by gas convection and gas radiation. The thermal resistances of the tube wall and oxide layer by conduction, as well as the steam side by convection (heating) were also considered.

Finally a sensitivity analysis has been carried out in order to determine the influence of the hot gases temperature and velocity, as well as the effect of oxide layer thickness over the wall temperature. Figure 6 shows the wall temperatures calculated for different values of the oxide layer thickness, considering an average temperature and uniform gas velocities leaving the furnace, equal to to1084°C and 7 m/s, respectively, as well as the boiler operating at nominal load (100 kg/s, 130 bar and 520°C). According to Fig. 6, the wall temperature rises to 580°C for thicknesses of the order of 0.51 mm and even higher values in the order of 634°C for thicknesses about 0.93 mm.

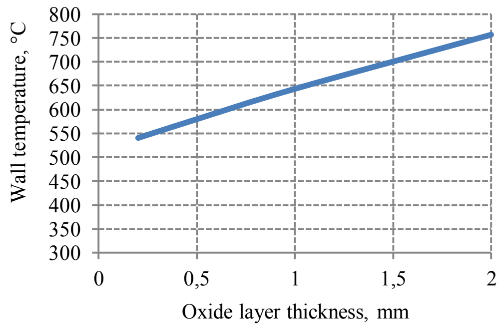


Figure 6- Influence of the oxide layer thickness on the wall temperature.

## 5. CONCLUSION

The wall temperatures has been the object of study of this work, focused on the heat transfer analysis of the steam superheater tubes, using standard procedure with the application of correlations recognized in the literature. Results were compared with the temperatures measured in the wall superheater of a steam generating unit in operation. The differences are relatively small, on the order of 40°C, but significantly large in the order of 560°C if considered the maximum allowed temperature of the specified material in the project of the superheater.

Experimental data point to relatively high values in the range of 580-660°C, reaching in some cases even higher values, of course calling attention to possible problems associated with the superheater tube itself as well as the temperature sensor by interfering in the process of heat transfer. Possible interference associated with the boundary conditions may be adversely acting in the temperatures readings, such as insulation, surface heat exchange by radiation, fixing of the thermocouple and other factors of lesser influence. The biggest issue is directly radiation on the sensor, also subject of analysis using CFD simulation.

Special attention was given also to the oxide layer inside the tubes of the superheater. In this case, considering the insulating effect of the oxide layer, evidently the calculated temperatures increased significantly, agreeing fully with the measured temperatures, but still requiring experimental confirmation.

## 6. ACKNOWLEDGEMENTS

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## 8. RESPONSIBILITY NOTICE

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