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ENERGETIC AND EXERGETIC STUDY OF STEADY STATE THREE STEPS FIRE TUBE BOILER OPERATING WITH NATURAL GAS

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Abstract. *The firetube boiler market has required increasing customized designs from manufacturers. In this work, the main objective is the development and the validation of a numerical simulator to perform an energy and exergy analysis of a steady state three passes firetube boiler production capacity of 1000 kg/hr, with working pressure between 1 and 10 bar, with non-submerged turn boxes and fueled with Bolivian natural gas. The impact on the temperature profile of various correlations for convective heat transfer was assessed. A central composite design was also performed to simultaneously evaluate the impact of six design or operating parameters (diameter of the tubes; boiler length; number of tubes in the second and third passes; excess of air in the mixture) and obtain an empirical second-order polynomial model in order to maximize boiler energy and exergetic efficiencies and reduce irreversibilities. A mesh test was also performed and the model was also validated with data obtained experimentally, observing the model is sensitive to the impact of working pressure and the difference between the experimental and theoretical values of the temperatures of the first and second passes. With the robust polynomial models given by the design of experiments approach, it was possible to define the most significant parameters, both for energy and exergetic efficiencies, as well as for exergy destruction. In decreasing order of importance of linear impacts, the excess of air; length of the boiler; the number and the diameter of tubes in the second and third passes and the diameter of the tube in the first pass were significant through linear contributions and some interaction terms. In addition, it was possible to quantify the trade-off between the increase of the heat exchange area and the reduction of flow turbulence level.*

Keywords: *Thermodynamics, Combustion, Design of experiments, Tube bundles, Heat transfer.*

1. INTRODUCTION

The industry uses steam as the main source of heating in various equipment and/or unit operations: chemical reactors, heat exchangers, evaporators, dryers and various thermal equipment. Boilers are the main parts in most industrial plants. They provide the energy source for space heating and cooling for personal comfort and the heat needed for many industrial processes. According to the Department of Energy (2012), the sectors that have the higher number of boilers are the chemical and food industries, plus the sector that has larger capacity boilers are the paper and chemical industries that use boilers to produce hot water or steam. However, the rational use of energy is an essential global concern. The world consumption of energy is constantly increasing, driven by the socioeconomic growth of nations and the increase in the world population.

In this context, fossil fuels will continue to be used for several years (Badcock-Wanson, 2012). Thus, the thermal efficiency of coal fired boilers varies between 81% and 85%, for oil between 78% and 81% and for gas between 76% and 81% (in a HHV basis). Poorly maintained boilers can lose up to 30% of their original efficiency (Einstein & Ernst , 2001). Most of the heat loss is due to high air/fuel ratios, steam generation below rated capacity, surface heat losses, and high flue gas temperatures (Kaya & Eyidogan, 2010). Therefore, an efficient boiler also has a significant influence on heating-related energy savings (Ganapathy, 2002). A substantial amount of energy can be saved by taking energy saving measures and improving the overall efficiency of the boiler. It is worth mentioning that until now few studies have been carried out on the exergetic and energetic analysis of boilers with low working pressure. Therefore, it is difficult to compare the data with other relevant work (Table 1).

Table 1. Results of efficiencies in the literature

Steam production capacity [t/hr]	Energy Efficiency [%]	Exergetic Efficiency [%]	Irreversibility Rate [%]	Reference
3	72.46	24.89	52	Saidur et al. (2010)
637.8	85	41	Not identified	Pattanayak and Ayyagari (2014)
Not identified	82	33	Not identified	Terhan and Comakli (2017)
Not identified	65	Not identified	Not identified	Zeng et al. (1999)
10-11.6	83.79-85.29	13.41-27.97	Not identified	Zhang et al. (2018)
1.1	70.49 – 80.18	16.42 – 19.70	71.37 – 72.44	This study

In the Table 1 shows the comparison of energy and exergetic efficiencies and losses in boilers with other works available in the literature. Thus, the main objective of this work is the development and validation of a numerical simulator that allows performing an energy and exergy analysis of a steady state fire tube boiler with up to three passes operating with natural gas. The validation of the model is based in the comparison of sets of convective heat exchange correlations to evaluate the sensitivity of the model and comparison of predicted temperature of the exit gas of the combustion chamber, second and third pass with collected experimental data of two boilers located in Bolivia. An analysis of the fire tube boiler using a Central Composite Planning (CCD) assessed the impact on energy and exergy efficiencies and irreversibility rate of the diameter of tube in the three passes, the boiler length, the number of tubes in pass 2 and 3 and the excess of air to.

2. BOILER MODELING

A steady-state model is developed for the heat transfer of a 3-pass fire tube boiler, with non-submerged turning boxes. Steam generation process, considered as saturated steam, is isobaric. The boiler is modeled as a set of several heat exchangers in series, submerged in a uniform volume of saturated water. A two-phase water/steam zone, metal zone, and gas zone, respectively. The gas zones of the combustion chamber and tube bundle are subdivided into elementary sections, for each section an energy balance is established considering the power generated and transferred in the volume control of length Δx in the direction of the gas flow, as shown in Figure 1. The temperature, pressure and composition of the gases are uniform in the control volumes. Additionally, both reactants and products are considered ideal gases.

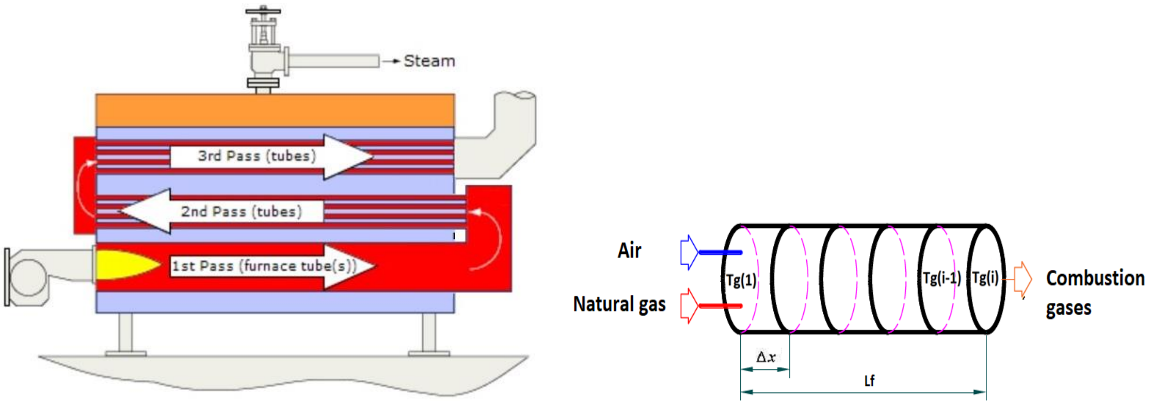

Figure 1. Schematic representation of the three-pass firetube boiler (Spiraxsarco, 2021) and boiler subdivision scheme. (Self-elaboration)

2.1. Combustion modeling

The studied fuel is a Bolivian natural gas (92.10% of CH_4, 4.56% ofC_2H_6, 1.23% of C_3H_8, 0.15% of C_4H_{10}, 0.32 of C_4H_{10}, 0.10% of C_5H_{12}, 0.10% of C_5H_{12}, 0.22% of C_6H_{14}, 0.73% of N_2 and 0.48% of CO_2). The complete combustion process is considered, producing only carbon dioxide (CO_2) and water vapor (H_2O). A gradual combustion model along the length of the furnace is used. The length of the flame is assumed approximately equal to 70% of the length of the boiler (Badcock-Wanson, 2012). The pattern of heat release along the flame (ie, the progress of the reaction and consequently the proportion of gas burned), F, can be described exponentially (Rhine & Tucker, 1991), parabolic (Roesler,

1997) or uniformly. In the present work, the parabolic shape proposed by Roesler is applied (Farhadi, Bahrami, Babaheidari, & Hashemi, 2005) according to the equation Eq. (1)

$$F = \frac{6}{L_f} \cdot \left(\frac{x}{L_f} - \frac{x^2}{{L_f}^2} \right) \quad \text{for} \quad 0 \leq x \leq L_f; F = 0 \quad \text{for} \quad x > L_f \tag{1}$$

where L_f is the flame length $[m]$ and x:is the length within the first pass $[m]$.

The chemical composition of the flue gases varies in each control volume as a function of the gradual combustion pattern.

2.2. Conservation equations

For each control volume, an energy balance is established considering the heat released by the combustion reaction (inside the flame) and the heat transferred to the water. For second and third pass control volume analysis, a single tube is analyzed for each pass. Thus, the mass flow at the tube inlet is divided between the number of tubes in each pass. The energy balance for a generic control volume is shown in Figure 2.

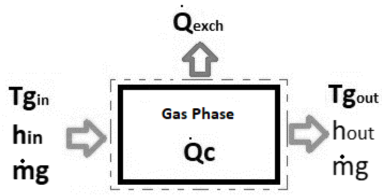

Figure 2. Simplified scheme of transport phenomena in a control volume (Own Elaboration)

The steady-state energy conservation equation for a control volume is given by Eq. (2) which can be positive, null, or negative.

$$\dot{m}_g \cdot \left(h_{g:out} - h_{g:in} \right) = \dot{Q}_c - \dot{Q}_{exch} \Leftrightarrow T_{out} = \frac{\dot{Q}_c - \dot{Q}_{exch}}{\dot{m}_g \cdot cp_g} + T_{in} \tag{2}$$

where m_g is the mass flow of flue gas $\left[\frac{kg}{s}\right]$, $h_{g:out}$ is the output specific enthalpy $\left[\frac{J}{kg}\right]$, $h_{g:in}$ is the input specific enthalpy $\left[\frac{J}{kg}\right]$, $\dot{Q}_c$ is the combustion heat release rate $[W]$, $\dot{Q}_{exch}$ is the rate of heat transfer from flue gas to metal $[W]$, T_{out} is the flue gas outlet temperature $[K]$ and T_{in} is the flue gas inlet temperature $[K]$.

The heat transfer rate $\dot{Q}_{exch}$ between the gas and metal zone is modeled as shown in Eq. (3).

$$\dot{Q}_{exch} = g_{rad} \cdot \sigma \cdot S_{Fi} \cdot [T_{out}^4 - T_{int}^4] + h_g \cdot S_{Fi} \cdot [T_{out} - T_{int}] \tag{3}$$

where g_{rad} is the total radiative heat transfer coefficient [-], σ is the Stefan Boltzmann radiation constant, $5.669.10^{-5}$ $\left[\frac{W}{m^2.K^4}\right]$, S_{Fi} is the internal area of the combustion chamber $[m^2]$, T_{int} is the internal temperature in the combustion chamber wall $[K]$ and h_g is the convection heat transfer coefficient $\left[\frac{W}{m^2.K}\right]$.

2.3. Equations for heat exchange

The total radiative heat transfer coefficient under the assumption of an infinitely long tube without axial radiation can be written as Eq. (4) (Schlunder, 1983), in which the flame emissivity, defined by Talmor (Talmor, 1982), is correlated with the type of fuel, in this case natural gas for fuel gases with a $\frac{C}{H}$ mass ratio between 3.5 and 5.0.

$$g_{rad} = \left[\frac{1}{\varepsilon_m} + \frac{1}{\varepsilon_{gas}} - 1 \right]^{-1} \; with \;\; \varepsilon_{gas} = \sqrt{0.2 \cdot \left(\frac{LHV}{900} \right)} \tag{4}$$

where ε_m is the emissivity of carbon steel SAE 1010, assumed equal 0.8 $[-]$ (Rahmani & Dahia, 2009), ε_g is the emissivity of combustion gases $[-]$ and LHV is the lower heating value of the natural gas $\left[\frac{Btu}{ft^3}\right]$.

For the internal convective heat transfer coefficient h_g, the literature presents a single correlation for laminar flow ($Re_{int} < 2{,}300$) and two options for correlations for turbulent flow ($Re_{int} > 4{,}000$). As the wall temperature is closer to the water temperature than the gas temperature due to the high heat transfer coefficient from the water side (Rahmani & Trabelsi, Numerical investigation of heat transfer in 4-Pass fire-tube boiler, 2014) (Huang, 1988), the temperature of the metal wall only varies slightly along the length of the boiler compared to the temperature of the gas. On the other hand, the gas temperature variation has a direct impact on the heat flow. Therefore, the boundary condition of uniform wall temperature is adopted from Eq. (5) is used to determine the Nusselt number under laminar flow conditions, Nus_{Lam}. In the turbulent regime, the Nusselt number in turbulent internal flow, Nus_{Turb}, can be obtained by the Gnielinski correlation (Kuznetsov, 2015) in Eq. (6) or the Petukhov correlation (1970) in Eq. (7). In the transition region, a linear interpolation between the calculated values for Reynolds numbers equal to 2,300 and 4,000, respectively, based on the value of the Reynolds number, is considered.

$$\mathrm{Nus_{Lam}} = \left[3.66^3 + \left(1.615^3 \cdot \left[\mathrm{Re_{int}} \cdot \mathrm{Pr_g} \cdot \frac{\mathrm{D_{Fi}}}{\Delta \mathrm{x}}\right]^{[1/2]}\right)^3 + \left(\frac{2}{1 + 22 \cdot \mathrm{Pr_g}}\right)^{(1/6)} \cdot \left(\mathrm{Re_{int}} \cdot \frac{\mathrm{Pr_g}}{\Delta \mathrm{x}}\right)^{(1/2)}\right]^3 \tag{5}$$

$$\mathrm{Nus_{Turb}}(\mathrm{Gnielinski}) = \frac{\frac{\xi_{nu}}{8} \cdot [\mathrm{Re_{int}} - 1{,}000] \cdot \mathrm{Pr_g}}{\left|1 + 12.7 \cdot \left[\frac{\xi_{nu}}{8}\right]^{1/2} \cdot [\mathrm{Pr_g}^{2/3} - 1]\right|} \tag{6}$$

$$\mathrm{Nus_{Turb}}(\mathrm{Petukhov}) = \frac{\frac{\xi_{nu}}{8} \cdot [\mathrm{Re_{int}} - 1{,}000] \cdot \mathrm{Pr_g}}{\left|1 + 12.7 \cdot \left[\frac{\xi_{nu}}{8}\right]^{1/2} \cdot [\mathrm{Pr_g}^{2/3} - 1]\right|} \cdot \left[1 + \left(\frac{\mathrm{d}}{\mathrm{L}}\right)^{2/3}\right] \cdot \left(\frac{\mathrm{T_b}}{\mathrm{T_{int}}}\right) \tag{7}$$

where Re_{int} is the internal Reynolds number $[-]$, T_b is the temperature of the gas mixture $[K]$, ξ_{nu} is the friction factor for turbulent flow in smooth cylindrical tubes [-], Pr_g is the Prandtl number of the gas mixture $[-]$, $\mathrm{D_{Fi}}$ is the internal diameter of the combustion chamber $[m]$, Δx is the length of each volume in the combustion chamber $[m]$, d is the tube diameter $[m]$ and L the tube length $[m]$.

For the external convective heat transfer coefficient h_o the literature presents two options for correlations for the combustion chamber pass (single tube) and three options for the tube bundle.

The Cornwell nucleate boiling correlation (Cornwell & Houston, 1994) for the combustion chamber is given by Eq. (8).

$$\mathrm{Nus_{out}} = 100 \cdot \left[\mathrm{D_{Fo}} \cdot \frac{\frac{\dot{\mathrm{Q}}_{\mathrm{exch}}}{\mathrm{S_{Fo}}}}{\mu_{\mathrm{water}} \cdot q_{latent}}\right]^{0.67} \tag{8}$$

where μ_{water} is the dynamic viscosity of water $\left[\frac{kg}{m.s}\right]$, q_{latent} is the latent heat of vaporization of water at 10 $[bar]$ assumed equal to 2,172 $\left[\frac{J}{kg}\right]$, $\mathrm{D_{Fo}}$ is the external diameter of the combustion chamber $[m]$ and $\mathrm{S_{Fo}}$ is the external area of the combustion chamber $[m^2]$.

Alternatively, the Gorenflo nucleate boiling correlation (Baehr, 2006) for the combustion chamber given by Eq. (9).

$$\mathrm{h_o} = 5{,}600 \cdot \left[1.73 \cdot \mathrm{Pr_2^{0{,}27}} + \left[6.1 + \frac{0.68}{1 - \mathrm{Pr_2}}\right] \cdot \mathrm{Pr_2^2}\right] \cdot \left[\frac{\dot{\mathrm{Q}}_{\mathrm{exch}}}{\mathrm{S_{to}} \cdot 20{,}000}\right]^{0.9 - 0.3 \cdot \mathrm{Pr_2^{0.t15}}} \tag{9}$$

where $\mathrm{Pr_2}$ is the reduced pressure $[-]$

Next to bundles of tubes, the boiling heat transfer coefficient is higher than for an individual tube, h_1 following the Gorenflo nucleated boiling correlation (Baehr, 2006) given by Eq. (9). This is due to the agitation of the vapor bubbles which increases heat transfer. Taking this effect into account, the heat transfer coefficient of the convection section tubes is calculated by Eq. (10).

$$\mathrm{h_o} = \mathrm{h_1} \cdot \left[1 + \frac{1}{2 + \frac{\dot{\mathrm{Q}}_{\mathrm{exch}}}{\mathrm{S_{to}} \cdot 1{,}000}}\right] \tag{10}$$

where h_1 is the external convection heat transfer coefficient of a single tube of the tube bundle $\left[\frac{W}{m^2.k}\right]$ and S_{to} is the external area of a single tube $[m^2]$.

The correlation of boiling heat transfer coefficient for Cooper tube bundle (Cooper, 1984) is given by Eq. (11).

$$h_o = 12.96 \cdot P_r{}^{0.12} \cdot (-0.4343 \cdot \mathrm{Ln}(P_r))^{-0.55} \cdot \frac{\dot{Q}_{exch}}{S_{to}}^{0.67} \tag{11}$$

The correlation of boiling heat transfer coefficient for Fritz tube bundle is given by Eq. (12).

$$h_o = 1.95 \cdot \left(\frac{\dot{Q}_{exch}}{S_{to}}\right)^{0.72} \cdot \left(\frac{P}{100{,}000}\right)^{0.24} \tag{12}$$

where P is the boiler working pressure $[Pa]$

2.4. Exergetic analysis

Exergy is a property that allows determining the useful work potential of a given amount of energy in a given state in relation to its environment (Kotas, 1995). Total exergy derives from disordered forms of energy in an idealized device where the current would go through physical and chemical processes while interacting with the environment. It is convenient, however, to separate physical exergy from chemical exergy allowing the calculation of exergy values using standard chemical exergy tables (Kotas, 1995). Thus, the specific exergy rate ε $\left[\frac{J}{mol}\right]$ is:

$$\varepsilon = \varepsilon_k + \varepsilon_p + \varepsilon_{ph} + \varepsilon_0 \tag{13}$$

where ε_k is the specific kinetic exergy $\left[\frac{J}{mol}\right]$, ε_p is the specific potential exergy $\left[\frac{J}{mol}\right]$, ε_{ph} is the specific physical exergy $\left[\frac{J}{mol}\right]$ and ε_0 is the specific chemical exergy $\left[\frac{J}{mol}\right]$. In this work, kinetic and specific exergy potential components are considered negligible. For an ideal gas, the specific physical exergy ε_{ph} assuming the value of the specific heat at constant pressure invariant with temperature is given by the following Eq. (14).

$$\varepsilon_{ph} = cp_g \cdot [T_g - T_0] - T_0 \cdot \left[cp_g \cdot \ln\left(\frac{T_g}{T_0}\right) - R \cdot \mathrm{Ln}\left(\frac{P_g}{P_0}\right)\right] \tag{14}$$

where P_g is the flue gas pressure $[kPa]$ and R is the universal gas constant considered equal to 8.314472 $\left[\frac{J}{mol.K}\right]$.

The studied substance consists of a mixture of gases, considered to be ideal, mixtures of natural gas, air and combustion products. The specific chemical exergy of the gas mixture is given by the following Eq. (15). (Kotas, 1995).

$$\varepsilon_0 = \sum_i x_i \cdot \tilde{\varepsilon}_i + R \cdot T_0 \cdot \sum_i x_i \cdot \mathrm{Ln}(x_i) \tag{15}$$

where $\tilde{\varepsilon}_i$ is the molar exergy of each compound in the gas mixture $\left[\frac{J}{mol}\right]$ (Kotas, 1995) and x_i is the molar fraction each compound of the reactants and combustion oxidant $[mol]$.

To calculate the standard molar specific chemical exergy for gaseous fuel (ε_{fuel}) in $\left[\frac{J}{mol}\right]$, the general expression given by Eq. (16) is used (Kaushik & Singh, 2014).

$$\varepsilon_{fuel} = -\Delta G + \sum n_p \cdot \varepsilon_{Ch,P} - \sum n_R \cdot \varepsilon_{Ch,R} \tag{16}$$

where ΔG is the change in the standard Gibbs function (calculated equal to-1.778×10^6 $\left[\frac{J}{mol}\right]$), $\varepsilon_{Ch,P}$ is the molar specific chemical exergy of the products of combustion $\left[\frac{J}{mol}\right]$ and $\varepsilon_{Ch,R}$ is the molar specific chemical exergy of the combustion reactants $\left[\frac{J}{mol}\right]$. (Kotas, 1995)

3. METODOLOGY

3.1. Boiler Description

In this work, two 3-pass fire-tube boilers with a steam production capacity of $1{,}500\left[\frac{J}{kg}\right]$ are analyzed with the following operating parameters for two boilers: absolute working pressure of $6{,}200\ [kPa]$ and $2{,}000\ [kPa]$ respectively, feed water temperature of $20\ [°C]$, fuel flow of natural gas equal to $0.024\ \left[\frac{kg}{s}\right]$ and excess of air equal to $19\ [\%]$

Based on the meteorological and climatic conditions of Cochabamba – Bolivia, $\mathrm{T_o} = 20\ [°C]$ is considered in this analysis as the ambient temperature and $\mathrm{P_o} = 75$ [kPa] as the atmospheric pressure.

3.2. Validation

For the mesh test, regular meshes with up to 800 volumes were tested in each pass (combustion chamber, second and third pass) to verify the temperature behavior. For the evaluation, 12 combinations of internal – external heat transfer correlations were tested to verify the behavior of the temperature parameter: Gnielinski – Gorenflo and Gnielinski – Cornwell for the combustion chamber; and Gnielinski-Gorenflo with correction factor, Gnielinski-Cooper with correction factor, Gnielinski-Fritz with correction factor, Gnielinski-Gorenflo, Gnielinski-Cooper, Gnielinski-Fritz, Gnielinski-Gorenflo with correction factor, Gnielinski-Cooper with correction factor, Gnielinski-Fritz with correction factor, Gnielinski-Gorenflo, Gnielinski- Cooper, Gnielinski-Fritz for second and third passes.

3.3. Design of experiments

A design of experiments is a series of tests in which the input variables are changed according to a certain rule, in order to identify the reasons for the changes in the output response. The objectives of the experiment include: determining which variables are most influential on the response, determining where to set the controllable influential variables so that the response is almost always close to the desired ideal value, so that the variability in the response is small, so that the effect of uncontrollable variables is minimized (Cavazzuti, 2013).

In this study, a Central Composite Design (CCD), also known as Box-Wilson Composite Central Planning (Anthony, 2003), was performed. The mathematical model with multiple regression analysis in a second-order polynomial fitted model is expressed in Eq.(17).

$$\mathrm{Y} = \beta_0 + \sum \beta_i\, X_i + \sum \beta_{ii}\, X_i^{\,2} + \sum \beta_{ij}\, X_i \cdot X_j \tag{17}$$

Where Y is expected response, β_0 is interception coefficient or constant term, β_i is coefficient of the linear effect of the variable i, β_{ii} is coefficient of the quadratic effect of the variable i and β_{ij} is coefficient of the interaction term (or rectangular) between variables i and j. For each variable, the influence of all factors and their linear relationships are studied. The investigated parameters and their respective levels are shown in Table 2.

Table 2. Variable geometric parameters and values of CCD factors (6 factors)

Symbol	Variable	Unit	-1	0	+1
X_1	D_{Fo} Tube diameter in step 1	[m]	0.5	0.6	0.7
X_2	D_{To} Tube diameter in passes 2 and 3	[m]	0.05	0.0525	0.055
X_3	L_F Boiler length	[m]	2	2.25	2.5
X_4	N_{tub2} Number of tubes in pass 2	[-]	67	72	77
X_5	N_{tub3} Number of tubes in pass 3	[-]	51	56	61
X_6	% Excess of air	[%]	9	19	29

4. RESULTS AND DISCUSSIONS

4.1. Parametric analysis in the objective of optimizing boiler efficiency

A mesh test was performed to verify the temperature behavior at the end of the flame and for each in order to define for each pass the length of the control volume, that is, the number of control volumes for different sets of correlations from the literature. The mesh test results for the combination of Petukhov – Gorenflo correlations, for internal and external heat transfer correlations, respectively, for the combustion chamber and for the tube bundle, the simulation values are far below reality, there is also a lack of convergence in temperature towards the end of the flame and end of the combustion chamber, even simulating with large amounts of control volume. For the second combination of Gnielinski – Gorenflo correlations, the results showed that the temperature stabilized around 1,089 $[ºC]$ for a division of 80 control volumes. Similarly, for the last volume of the combustion chamber, it is observed that the temperature stabilizes around 1,027 $[°C]$ for a division of 80 control volumes. For the temperature in the last volume of the second and third passes, it is observed that the temperature stabilizes around 422 ºC and 277 $[°C]$, respectively, for a division of 200 control volumes. It is important to note that the values in this case the temperatures are within reality, based on the experimental value the value measured at the end was 853 $[°C]$ for the combustion chamber, 453 $[°C]$ for the second pass and 258 $[°C]$ for the third pass. Thus, the correlation that best fits for the internal heat exchange coefficient and Gnielinski and for the external convection heat transfer coefficient the best correlation that fits the model for the combustion chamber is the Gorenflo correlation and the Fritz correlation without correction factor for the tube bundle, using 200 control volumes.

Model validation was performed comparing the predicted gas temperature at the outlets of each pass with the operational data of the two boilers (Boiler 1 and Boiler 2). For boiler 1, for the first step the operational and expected temperatures of the gas at the outlet is equal to 853 $[°C]$ and 897 $[°C]$, respectively, having a difference of 44 $[°C]$, in the second step the temperatures are 453 $[°C]$ and 426 $[°C]$, respectively, having a difference of 27 $[°C]$ and finally for the third step the temperatures are 258 $[°C]$ and 298 $[°C]$, respectively, having a difference of 40 $[°C]$. For boiler 2, the temperatures are 831 $[°C]$ and 857 $[°C]$, respectively, having a difference of 26 $[°C]$, in the second step the temperatures are 430 $[°C]$ and 407 $[°C]$, respectively, having a difference of 23 $[°C]$ and finally for the third step the temperatures are 245 $[°C]$ and 276 $[°C]$, respectively, having a difference of 31 $[°C]$.

It is important to note that for the two boilers losses in the turning boxes were not considered, so with the exception of the second pass, the heat exchange will always be smaller compared to the real case. In an analysis of the sensitivity capacity of the model with the pressure change, it is observed that in the case of boiler 2, it is expected to find a lower temperature and the model results are coherent when compared to the case of boiler 1.

4.2. Parametric analysis in the objective of optimizing boiler efficiency

Table 3. Regression coefficient, Student test and p-value for energy, exergetic efficiency and rate irreversibility (significant parameters are given in red)

Factor	Energy efficiency [%]			Exergetic Efficiency [%]			Rate of irreversibility [W]		
	Coefficient	Test-t	p-value	Coefficient	Test-t	p-value	Coefficient	Test-t	p-value
Mean/Interc.	75.77844	34,563.18	0.000000	18.10178	5,370.629	0.000000	920,903.3	46,677.99	0.000000
(1)X1 (L)	0.05176	45.27	0.000000	0.01559	8.868	0.000000	35.3	3.43	0.003192
X1 (Q)	0.00172	0.40	0.696351	0.00304	0.457	0.653539	-3.7	-0.09	0.925797
(2)X2 (L)	-0.21794	-190.61	0.000000	-0.07147	-40.659	0.000000	-17.6	-1.72	0.104488
X2 (Q)	0.01672	3.86	0.001243	0.00304	0.457	0.653539	-3.7	-0.09	0.925797
(3)X3 (L)	1.88618	1,649.60	0.000000	0.64471	366.771	0.000000	129.4	12.58	0.000000
X3 (Q)	-0.19328	-44.69	0.000000	-0.05196	-7.815	0.000001	-403.7	-10.37	0.000000
(4)X4 (L)	0.28500	249.25	0.000000	0.09824	55.886	0.000000	41.2	4.00	0.000923
X4 (Q)	-0.01328	-3.07	0.006919	-0.00196	-0.295	0.771500	-3.7	-0.09	0.925797
(5)X5 (L)	0.26882	235.11	0.000000	0.08941	50.866	0.000000	0.0	0.00	1.000000
X5 (Q)	-0.00828	-1.92	0.072459	-0.00196	-0.295	0.771500	-3.7	-0.09	0.925797
(6)X6 (L)	-2.13029	-1,863.10	0.000000	-0.72559	-412.784	0.000000	6,123.5	595.15	0.000000
X6 (Q)	0.02672	6.18	0.000010	0.03304	4.969	0.000117	-603.7	-15.51	0.000000
X1(L)-X2 (L)	0.00000	0.00	1.000000	-0.00187	-1.035	0.315245	0.0	0.00	1.000000
X1(L)-X3(L)	-0.00625	-5.30	0.000058	-0.00375	-2.070	0.054034	-25.0	-2.36	0.030661
X1(L)-X4(L)	-0.00188	-1.59	0.130062	-0.00250	-1.380	0.185535	0.0	0.00	1.000000
X1(L)-X5(L)	-0.00188	-1.59	0.130062	0.00125	0.690	0.499575	-6.3	-0.59	0.563405
X1(L)-X6(L)	-0.00250	-2.12	0.048912	-0.00312	-1.725	0.102711	0.0	0.00	1.000000
X2(L)-X3(L)	0.00875	7.42	0.000001	0.00062	0.345	0.734367	93.7	8.84	0.000000
X2(L)-X4(L)	0.00938	7.95	0.000000	0.00437	2.415	0.027307	18.7	1.77	0.095015
X2(L)-X5(L)	0.00687	5.83	0.000020	0.00062	0.345	0.734367	0.0	0.00	1.000000
X2(L)-X6(L)	-0.03250	-27.58	0.000000	-0.00500	-2.760	0.013397	-56.3	-5.30	0.000058

X3(L)-X4(L)	-0.03187	-27.04	0.000000	-0.00375	-2.070	0.054034	-118.7	-11.20	0.000000
X3(L)-X5(L)	-0.02812	-23.86	0.000000	-0.00625	-3.449	0.003062	-100.0	-9.43	0.000000
X3(L)-X6(L)	0.11500	97.57	0.000000	0.00312	1.725	0.102711	481.3	45.38	0.000000
X4(L)-X5(L)	-0.00875	-7.42	0.000001	-0.00500	-2.760	0.013397	-25.0	-2.36	0.030661
X4(L)-X6(L)	0.01187	10.08	0.000000	-0.00063	-0.345	0.734367	81.2	7.66	0.000001
X5(L)-X6(L)	0.01313	11.14	0.000000	-0.00312	-1.725	0.102711	75.0	7.07	0.000002

For the parametric analysis in the mathematical model of the boiler, the respective impacts of 6 parameters on energy and exergy efficiencies, and irreversibility rates are analyzed through the execution of a CCD. The results for the 3-pass fire tube boiler present values between 70.49% and 80.18% for energy efficiency and between 16.42% and 19.70% for exergy efficiency and, finally, for the irreversibility rate., there are values in the range of 913.0 and 926.7 [kW]. Comparing with the literature data (Table 1), the simulation results had the same order of magnitude, the difference is due to the fact that the boiler capacity of the study is smaller, and the geometry is different.

From the simulation values for energy efficiency, exergetic and irreversibility rate, the coefficients of the polynomial model were calculated. Table 3 shows the coefficients of the polynomial model and the coefficients of the t-student in whichever greater than 5% means that the variable or combination is significant they are marked.

4.2.1. Energy efficiency

From the energy efficiency simulation values, the coefficients of the polynomial model were adjusted by the least squares' method, which presents a value of coefficient of determination R^2 and adjusted coefficient of determination $R_{adjusted}$ equal to 1 and 0.99999, respectively. From Table 1 it can be observed that all the variables studied are significant, at least, by their linear term. However, the main significant parameters are excess air (X6), boiler length (X3); the number of tubes in the second pass (X4); the number of tubes in the third pass (X5); the diameter of the tubes in the second and third passes (X2). The linear contribution of the boiler length (X3) is representing the importance of the heat exchange area (directly proportional to this variable) in improving the heat exchange for the water. The same argument applies for the diameter of the tube in the first pass (X1), whose variation impacts the wetted perimeter. The positive interactions between X6 with X3, X5 and X4 point to a weighting of the dilution effect of excess air with the increase of the heat exchange area. The positive interactions between X2 and X4, X3 and X5 are representing the increase in heat exchange with the area and its impact on energy efficiency. For negative interactions between X2 and X6 (dilution effect reinforced by reduced convective exchange in the second and third passes); X3 and X4 and X3 and X5 (even with an increase in the area for heat exchange, the predominant factor indicated by the model is the reduction of the mass flow in each tube of the bundles); X4 and X5 (interaction between passes 2 and 3 related to lower heat exchange due to reduced velocity in the tubes); and X1 and X3 (reduction of the Reynolds number in the first pass) complete the understanding of the physics of the problem. These terms illustrate the difficult trade-off between increasing the heat exchange area and decreasing the flow velocity.

4.2.2. Exergetic Efficiency

Similarly, from the exergetic efficiency simulation values, the coefficients of the polynomial model were adjusted by the least square's method, which presents a value of R^2 equal to 0.99988 and $R_{adjusted}$ equal to 0.99983. It is possible to observe that all the variables studied are significant similarly to energy efficiency. However, the main significant parameters are, in decreasing order of importance: excess of air (X6), boiler length (X3); the number of tubes in the second pass (X4); the number of tubes in the third pass (X5); the diameter of the tubes in the second and third pass (X2) and the diameter of the tube in the first pass (X1). These last four factors only contribute a linear effect. Compared to the energy efficiency model, a quadratic contribution of excess of air was added.

4.2.3. Rate of irreversibility

From the simulation values, the coefficients of the polynomial model were adjusted by the least squares' method. The model presents excellent robustness with values of R^2 equal to 0.99995 and $R_{adjusted}$ equal to 0.99988. It is possible to observe that not all the variables studied are significant. However, unlike the two efficiencies, the main significant parameters are, in decreasing order of importance: excess of air (X6) with a predominant linear effect; with the (mainly linear) contributions associated with the length of the caldera (X3); the number of tubes in the second pass (X4); and the diameter of the tube in the first pass (X1). These last two factors only contribute a linear effect, but Student's t test returns values close to the cut-off value. This observation, together with the non-significant aspect of the variables associated with the diameter of the tubes in the second and third pass (X2) and the number of tubes in the third pass (X5), shows that the significant factors are associated with the first pass where the reaction of combustion takes place.

4.3. Optimization

From the parametric analysis based on the mathematical models of the boiler, an optimization of the efficiency parameters of the boiler can be carried out, following the best operating conditions. The analysis to define the optimized points pointed to conditions outside the experimental domain. Thus, considering the restriction of the experimental domain, for the maximum energy yield equal to 80.18%, maximum exergetic yield equal to 19.70% and the lowest irreversibility rate equal to 913 [kW], the best operational and geometric conditions for the Boiler 1 are a pipe 0.7 [m] diameter in pass 1 and a maximum boiler length equal to 2.5 [m]. For these two parameters, the maximum value means an improvement in the heat exchange to the water. For the diameter of the tubes in passes 2 and 3, the smallest value equivalent to 0.05 [m] was pointed out. This leads to an increase in the velocity of the gases in the tubes and consequently a reduction in the Reynolds number and the intensity of the turbulence. The level of these parameters leading to a reduction in mass flow are balanced with an increase of the number of tubes in pass 2 (maximum value with 77 pipes) and in pass 3 (maximum value with 61 pipes). Finally, the percentage of excess of air must be minimized with a value of 9%. This fact, the only one that can be adjusted without requiring a change in boiler geometry. Such variation agrees with the literature, since energy and exergy efficiency tend to decrease with increasing excess of air (Saidur, Ahamed, & Masjuki, 2010). With the results of the optimized simulation, the exergy flow in the boiler is shown in a Grassman diagram.

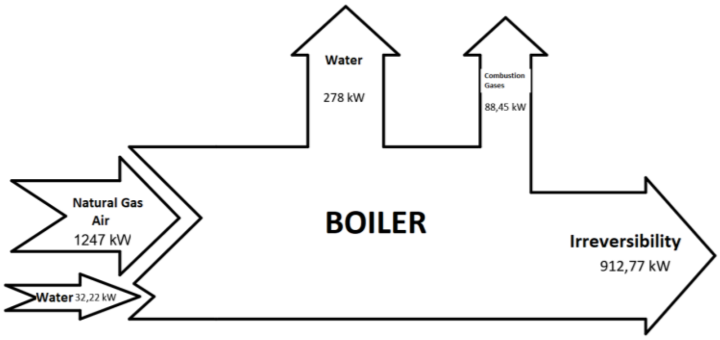

Figure 3. Grassmann diagram for exergy flow in the boiler

For the boiler studied, it can be seen in Figure 3 that the exergy of the water vapor is equal to 21.73% and the exergy of the combustion gases is equal to 6.91%, but the irreversibility is equal to 71.35%. Such value is quite higher when compared to Saidur et al. (2010). Nonetheless, the simulated boiler is 3 times smaller than the model from literature.

5. CONCLUSION

A mesh test was carried out with operational data collected in the field, behavior of the temperature at the end of each step and defining for each step the length of the control volume, number of control volumes. From the mesh test it was also verified that the correlation that best fits the internal heat exchange coefficient is the Gnielinski correlation and for the heat transfer coefficient by external convection it is the Gorenflo correlation and for the tube bundle it is the Fritz Correlation without the correction factor, using 200 control volumes. In the validation of the model for Boiler 1 and Boiler 2, it is important to highlight the lower heat exchange with respect to the real case due to the absence of losses in the dump boxes that were not simulated, so the temperature values are always slightly below above the experimental values because the second step in the simulation leads to an offset above the experimental one. Regarding the parametric analysis in order to optimize the efficiency of the boiler, 6 geometric and process parameters are evaluated to evaluate the most significant factors in the energy and exergetic efficiencies and in the rate of irreversibility through a complete factorial planning. For energy and exergetic efficiency, the coefficients of the polynomial model were adjusted by the method of least squares, which presents the following values of R^2 equal to 1 and $R_{adjusted}$ equal to 0.99999 and R^2 equal to 0.99988 and $R_{adjusted}$ equal to 0.99983, respectively. From the Table 1 for both efficiencies all the variables studied are significant, at least, for their linear term. However, the main significant parameters are excess air (X6), boiler length (X3); the number of tubes in the second step (X4). The linear contribution of the length of the boiler (X3) represents the importance of the heat exchange area in improving heat exchange by water in the same way for the diameter of the tube in the first step (X1), whose variation effects on the wet perimeter. For both the positive and negative interactions it can be seen that they illustrate the difficult balance between increasing the heat exchange area and decreasing the flow velocity. For the rate of irreversibility, have R2 equal to 0.99995 and Radjusted equal to 0.99988. From the Table 1 it is possible to observe that not all the variables studied are significant. However, the main significant parameters are, in decreasing order of importance: excess air (X6) with a strong linear effect; with the two mainly linear contributions associated with the length of the caldera (X3); the number of tubes in the second step (X4); and the diameter of the tube in the first step (X1). Finally, the analysis to define the optimized points pointed to conditions outside the experimental domain. Thus, considering the restriction of the experimental domain, the best combination of experimental conditions for the maximum energy yield equal to 80.18%, maximum exergetic yield equal to 19.70% and the lowest rate of irreversibility equal to 913 [kW] the best Operating and geometric conditions for Boiler 1 are a pipe diameter in step 1 increased by a value of 0.7 [m] and a maximum boiler length equal to 2.5 [m]. For the diameter of the tubes in steps 2 and 3, the smallest value was indicated, which is equivalent to 0.05 [m], the maximum number of tubes in step 2 with 77 tubes and in step 3 the maximum value with 61 tubes. Finally, the percentage of excess air in combustion must be minimized with a value of 9%.

Additional studies can be carried out to refine the model to include more phenomena and investigate other aspects that impact performance, such as modeling the losses in the turning boxes; including in the modeling of elements such as corrugated surfaces and/or deflectors: and implementing a more refined model to represent incomplete combustion. The current model can be applied to other boilers to perform a comparison between the efficiencies. Additionally, the modeling of the transient operation can help determine peak load capacity.

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