THERMAL-HYDRAULIC MODELING OF THE STEADY-STATE OPERATING CONDITIONS OF A FIRE-TUBE BOILER

by

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In this work, we are interested to simulate the thermal-hydraulic behavior of three-pass type fire-tube boiler. The plant is designed to produce 4.5 tons per hour of saturated steam at 8 bar destined principally for heating applications. A calculation program is developed in order to simulate the boiler operation under several steady-state operating conditions. This program is based upon heat transfer laws between hot gases and the fire-tube internal walls. In the boiler combustion chamber, the heat transfer has been simulated using the well-stirred furnace model. In the convection section, heat balance has been carried out to estimate the heat exchanges between the hot gases and the tube banks.

The obtained results are compared to the steady-state operating data of the considered plant. A comparative analysis shows that the calculation results are in good agreement with the boiler operating data. Furthermore, a sensitivity study has been carried out to assess the effects of input parameters, namely the fuel flow rate, air excess, ambient temperature, and operating pressure, upon the boiler thermal performances.

Key words: thermal-hydraulic modeling, fire-tube boiler, radiation heat transfer

INTRODUCTION

Fire-tube boiler is the original boiler design which brought the tide of power to the marine world. This type of boiler was used in driving steam engines, locomotives, and in marine applications, such as in the river boats. In the past few years, improvement in boiler designs and more efficient fire-tube layout have resulted in more compact, less costly packages which are more accessible for cleaning and inspection [1].

Several numerical and experimental investigations [2-6] have been performed to understand the thermal-hydraulic characteristics in conventional steam boilers. In the past, traditional methods relied heavily on expensive experimentation and the building of scaled models, but now a more flexible and cost effective approach is available through greater use of mathematical modelling and computer simulation. The recent advances in the computer technology have made it possible to perform complex calculations efficiently and even faster than real time. The modeling and simulation of fire-tube boilers has reached remarkable development, through the application of CFD techniques [7, 8] and other advanced thermal-hydraulic system codes [9].

Fire-tube boilers are named so because the combustion flames and flue gases circulate inside the boiler tubes. They can be powered by gas, oil, or solid fuel. Actually, fire-tube boilers are used widely for industrial, commercial steam and hot water generation. Modern fire-tube boilers can produce steam at the pressure of up to 25 bar (1 bar = 100 kPa), and through-puts up to about 25 tons per hour [10]. Packaged fire-tube boilers are selected due to the following advantages [11]:

– space requirements for these units are frequently less than that for other types due to their compact design and low headroom needed,
– they require no special foundation like masonry or refractory setting,
– large diameter shells allow greater steam storage space and steam releasing area,

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– large water content provides large steaming capacity for the space occupied which is excellent for process load applications that require quick response to load demands,
– this design provides simplicity in removing scale and silt precipitation, and
– at the maximum load, these units can attain more than 80% of the thermal efficiency.

Due to long-time use, fire-tube boilers may undergo a series of degradation processes that could expose the system structural integrity to serious hazard and, consequently, huge economic and human life losses could occur [12, 13]. Micro-structural and mechanical analysis shows that [14] under prolonged use of the fire tube or severe operating conditions, material degradation and embrittlement and structural changes may occur. Therefore, a thermal-hydraulic modeling of a fire-tube boiler can be a helpful tool in defining the optimal operating conditions in order to keep the integrity of the steam generator.

The main objective of this study is to elaborate a calculation program in order to simulate the thermal-hydraulic behavior of a three-pass fire-tube steam boiler under various operating conditions. A computer calculation has been performed in order to investigate the heat transfer characteristics between the combustion gases and the saturated water surrounding the fire tubes. A brief description of the boiler design and the general aspects of heat transfer in each part are given. The calculation of heat transfer in boilers is usually based on the coarse approximation of the temperature and emissivity in the combustion chamber [4]. Therefore, a set of simplifications may be considered in order to reduce the randomness and the complexity of the boiler geometry. The boiler is modeled in three zones. The first zone represents the radiation section (or furnace), and the second and the third zones are used to simulate the convection section. Each zone represents isothermal gas volumes enclosed by isothermal surfaces. The well-stirred furnace model is used to simulate the heat transfer characteristics in the boiler combustion chamber. In the convection section, the heat transfer is calculated through the heat balance between the hot gases and the tube banks heat exchanger.

The modeling validation has been made by comparing the calculated gas temperatures exiting the boiler with the measured ones for several steady-state operating conditions. In addition, a qualitative analysis has been made in order to analyze the effect of the operational parameters: the fuel flow rate, air excess, ambient temperature, and operating pressure on the boiler thermal performances.

BOILER DESCRIPTION

The studied unit is a single furnace, three-pass fire-tube boiler. It belongs to the COCHRAN Wee-Chieftain boiler family. Heavy gas-oil fuel is burned to produce 4.5 tons per hour saturated steam used for steam heating applications [15]. Figure 1 shows a cross-section of a typical fire-tube boiler arrangement. It contains several horizontal fire-tubes mounted in a pressure shell that is partially filled with water covering the tubes [16, 17].

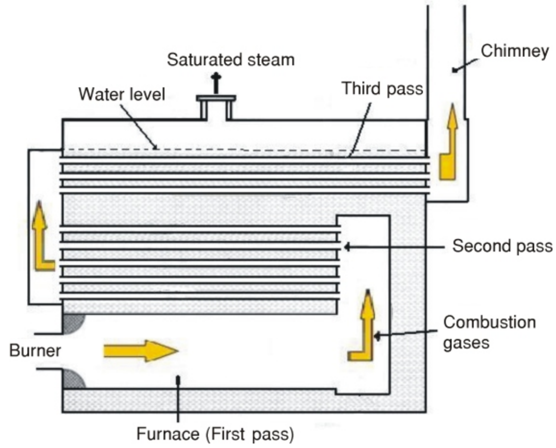

Figure 1. Schematic representation of a typical 3-passes fire-tube boiler [10]

From the heat transfer point of view, it consists mainly of two parts: the radiation section and the convection sections. The radiation section, also known the first pass, is a horizontal cylindrical furnace chamber within which the fuel is burned and heat is generated. Heat transfer within the radiation section is performed mainly by radiation, although convection may contribute to less than 10% [18]. The convection section tubes are arranged as horizontal tube banks located above the combustion chamber, and they recover additional heat from the furnace gases at a lower temperature level better than the radiant section tubes. All the tubes are surrounded by water which absorbs the heat enough to change into a vapor state [19].

The principal mode of heat transfer in this section is convection. Fuel and air are forced into the furnace by the burner to produce heat. From there, the hot flue gases travel throughout the boiler tubes. The number of tubes in each pass is selected to give similar velocities in each pass. Because of the forced-draft fan, the boiler is pressurized, and there may be a slight positive pressure at the flue gas outlet [19]. This also permits higher gas velocities in the unit. The combustion gases flow out of the radiation section toward the convection section, crossing the second and the third pass and finally escape by the chimney to the atmosphere. The main geometric specifications of the steam boiler are presented in tab. 1.

BOILER MODELING

The modeling procedure consists of subdividing the boiler in three control volumes, each one repre-

Table 1. Physical data of the steam boiler

Description	Values
Heat generated power [MW]	3.5
Operating pressure [bar]	8.0
Steam generation [t/h]	4.5
Total water amount [m^3]	7.5
Maximum exhaust gas temperature [°C]	240
Furnace length [mm]	2.6
Furnace internal diameter [mm]	825
Furnace external diameter [mm]	837.8
Tube number in the 2 second pass	76
Tube number in the 3 third pass	92
Mean tube length [m]	2.6-3
Tube internal diameter [mm]	53
Tube thickness [mm]	3.5

senting a gas-water heat exchanger immersed in the saturated water as illustrated in fig. 2. The heat transfer will be considered between gas volume and internal surfaces of the boiler tubes. On these surfaces, energy balance is established taking into account the radiation and convection heat transfer. In the combustion chamber the radiative heat transfer has been modeled using a well-stirred furnace model [10]. At the tube external walls, the nucleate boiling heat transfer mechanism is considered.

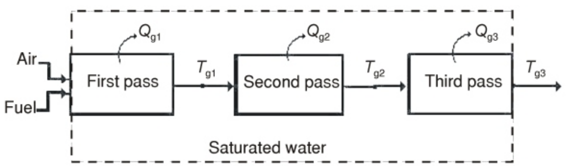

Figure 2. Nodalization diagram of the steam boiler

In order to simplify the calculations and reduce the number of the variables, a set of simplification assumptions was considered:

– the temperature of combustion gases is uniform in the control volume,
– in the convective section, the radiation heat transfer is neglected,
– the heat transfer at the reversal chamber, transfer box, and the smoke box is neglected,
– the total effective emissivity of the furnace wall is considered to be $\varepsilon_p = 0.85$, and
– the refractory thermal resistance is estimated at 0.2 m²K/W.

Radiation section heat transfer

Well-stirred model description

The well-stirred (single gas) model is rather general and practical for any combustion chamber configuration and to all fuels. Due to its generality, relative simplicity and predictive potential, this model is extensively used for the preliminary design of process heaters and boiler radiant sections [10]. In addition, the model may be readily used to investigate the furnace thermal performances during the changes in operating parameters such as fuel flow rate, air preheat, and air excess. In the well-stirred model, the furnace is modeled in three zones, namely, the central hot gas zone (the flame and combustion gases), a heat sink, and the refractory walls. Heat is transferred to the sink surface by radiation and convection from the hot gases and radiation from refractory surfaces [8].

The chemical reaction for gas-oil fuel combustion is given in eq. (1), where e is the air excess and Q_f is the combustion released energy

$$\mathrm{C_{21}H_{44}} \quad 32(1 \quad e)(\mathrm{O_2} \quad 3.76\,\mathrm{N_2}) \quad 21\mathrm{CO_2}$$
$$22\,\mathrm{H_2O} \quad 32e\,\mathrm{O_2} \quad 120.32(1 \quad e)\mathrm{N_2} \quad Q_\mathrm{f} \tag{1}$$

The rate of the heat generation Q_f, is realized by the fuel combustion reaction, and it is expressed as the mass flow rate of the combustion fuel, M_f, multiplied by its lower heating value (*LHV*):

$$Q_\mathrm{f} \quad M_\mathrm{f} LHV \tag{2}$$

The adiabatic temperature, T_{ad}, refers to the theoretical flame temperature assuming no heat losses through the walls [20]. It is computed by equating the lower heating value of the fuel to the enthalpy of combustion products corresponding to a unit mass fuel

$$T_\mathrm{ad} \quad T_\mathrm{a} \quad \frac{Q_\mathrm{f} \quad M_\mathrm{air}(T_\mathrm{pr} \quad T_\mathrm{a})}{M_\mathrm{g} Cp_\mathrm{g}} \tag{3}$$

The mass flow rate of the combustion gases, $M_g = M_{air} + M_f$, and the specific heat capacity, Cp_g, can be computed from

$$Cp_\mathrm{g} \quad \frac{21Cp_{\mathrm{CO_2}} \quad 22Cp_{\mathrm{H_2O}} \quad 32Cp_{\mathrm{O_2}} \quad 120.32(1 \quad e)Cp_{\mathrm{N_2}}}{M_\mathrm{tot}} \tag{4}$$

heat transfer from the combustion product to the furnace internal walls by convection and radiation, as

$$Q_\mathrm{p} \quad g_\mathrm{ray}\sigma(T_\mathrm{g}^4 \quad T_\mathrm{pi}^4) \quad h_\mathrm{c} A_\mathrm{p}(T_\mathrm{g} \quad T_\mathrm{pi}) \tag{5}$$

where g_{ray} is the total transfer factor for radiation from the gas to the heat sink, which can be evaluated by

$$g_\mathrm{ray} \quad \frac{A_\mathrm{T}}{\dfrac{1}{\varepsilon_\mathrm{p}} \quad C \quad \dfrac{1}{\varepsilon_\mathrm{g}} \quad 1} \tag{6}$$

where C is the fraction of metallic wall area of the furnace to the total area ($C = A_p/A_t$) and ε_p is the effective emissivity of the combustion chamber tube wall.

The convective component in eq. (1) can be reformulated as

$$h_c A_p (T_g \quad T_{pi}) \quad \frac{h_c A_p}{4\sigma T_{gp}^3}(T_g^4 \quad T_{pi}^4) \tag{7}$$

T_{gp} may be approximated as the arithmetic mean temperature of T_g and T_{p1}. By substituting eq. (7) in eq. (5), the net rate of heat transfer in the 1st pass is defined by

$$Q_p \quad g_{rc}\sigma(T_g^4 \quad T_{pi}^4) \tag{8}$$

where

$$g_{rc} \quad g_{ray} \quad \frac{h_c A_p}{4\sigma T_{gp}^3} \tag{9}$$

For laminar flow regime, the convective heat transfer coefficient is computed from Spang correlation

$$\mathrm{Nu}_g \quad 4.364 \frac{0.086 \quad \mathrm{Re}\,\mathrm{Pr}\,\frac{D}{L} \quad ^{1.33}}{1 \quad 0.1\mathrm{Pr} \quad \mathrm{Re}\frac{D}{L} \quad ^{0.83}} \tag{10}$$

Then for fully developed turbulent and transition flow, the convective heat transfer coefficient is estimated using Petukhov-Gnielinski correlation [21]

$$\mathrm{Nu}_g \quad \frac{\frac{\zeta}{8}(\mathrm{Re} \quad 1000)\mathrm{Pr}}{1 \quad 12.7\sqrt{\frac{\zeta}{8}}(\mathrm{Pr}^{2/3} \quad 1)} \quad 1 \quad \frac{D}{L} \quad ^{2/3} \tag{11}$$

where $\xi = (1.82 \mathrm{Log}_{10} \mathrm{Re} - 1.64)^{-2}$. The furnace internal wall temperature T_{pi}, is computed from

$$T_{pi} \quad T_{sat} \quad \varphi \quad \frac{1}{h_w} \quad \frac{d_o \log \frac{d_o}{d_i}}{2k_p} \tag{12}$$

h_w being the nucleate boiling heat transfer coefficient calculated using Gorenflo correlation [22]

$$h_w \quad 5600 F_{PF} \quad \frac{\varphi}{20000} \quad ^{0.9 \quad 0.3\,\mathrm{Pr}^{0.15}} \tag{13}$$

where

$$F_{PF} \quad 1.73\,\mathrm{Pr}^{0.27} \quad 6.1 \quad \frac{0.68}{1 \quad \mathrm{Pr}} \quad \mathrm{Pr}^2 \tag{14}$$

The heat transfer calculation is refined taking into account the heat losses through the wall. The heat loss by conduction through the refractory wall is given by

$$Q_r \quad U_r A_r (T_g \quad T_a) \tag{15}$$

where U_r is the overall heat transfer coefficient from the gas to the exterior through the refractory and A_r is the area of the refractory surface. Therefore, the total exchanged power Q_{g1}, from the combustion product to the furnace wall as well as the heat transfer loss may be expressed as

$$Q_{g1} \quad Q_p \quad Q_r \tag{16}$$

The general equation giving the thermal performances of the furnace can be expressed by a non-dimensional form [10]

$$\frac{Q_{g1}}{d} D \; d \quad T_p^{\;4} \quad 1 \quad \frac{Q_{g1}}{d} L_{loss} \quad 1 \quad \frac{Q_{g1}}{d} \quad T_a \tag{17}$$

where reduced firing density D , reduced furnace efficiency and the refractory loss factor L_{loss}, is defined [23] as

$$D \quad \frac{Q_f}{\sigma g_{rc} T_{ad}^3 (T_{ad} \quad T_a)} \tag{18}$$

$$L_{loos} \quad \frac{U_r A_r}{g_{rc}\sigma T_{ad}^3} \tag{19}$$

Using the adiabatic (fictitious) flame temperature T_{ad}, the reduced temperatures are defined as

$$T_a \quad \frac{T_a}{T_{ad}} \tag{20}$$

$$T_p \quad \frac{T_p}{T_{ad}} \tag{21}$$

The reduced furnace efficiency Q_{gl}, was obtained by solving the non-linear eq. (17), and then the new gas temperature T_g was calculated by the following equation

$$T_g \quad T_{ad} \quad 1 \quad \frac{Q_{gl}}{d} \tag{22}$$

The furnace exit temperature is smaller than the central hot gas temperature T_g, by an amount T_g. For a wide range of furnace type and operating conditions, an empirical correlation gives

$$\Delta T_g \quad T_{ad} \quad 1 \quad \frac{1}{d} \quad Q_g \tag{23}$$

For most practical cases, the empirical value of $d = 1.2$ is recommended [8, 18]. Then, the gas temperature exiting the combustion chamber T_{g1}, can be obtained by eq. (24). In fact, this temperature will be considered as the inlet gas temperature for the convection section

$$T_{g1} \quad T_g \quad \Delta T_g \tag{24}$$

Convective section heat transfer

The heat exchanged rate between the gas flowing inside the convection tubes and the surrounding saturated water can be obtained by the general equation

$$Q_{\mathrm{g}} = \frac{T_{\mathrm{g}} - T_{\mathrm{sat}}}{\frac{1}{A_1 h_{\mathrm{g}}} + \frac{\log \frac{r_o}{r_l}}{2\pi L k_{\mathrm{p}}} + \frac{1}{A_{\mathrm{o}} h_{\mathrm{w}}}} \tag{25}$$

For laminar regime, the gas side heat transfer coefficient, h_{g}, is calculated using the Seider-Tate equation

$$\mathrm{Nu}_{\mathrm{g}} = 1.86 \left(\mathrm{Re}\,\mathrm{Pr} \frac{D}{L} \right)^{1/3} \tag{26}$$

and for turbulent and transition regime, the Dittus-Boelter correlation is used [22, 23]

$$\mathrm{Nu}_{\mathrm{g}} = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \tag{27}$$

Heat boiling transfer coefficients on the outside of tube banks are greater than those for an individual tube. This is due to the vapor bubbles agitation enhancing the heat transfer. As a result of this, the mean heat transfer coefficient of the bundle is significantly larger than that of a single tube. According to Gorenflo [24] it holds that

$$h_{\mathrm{w}} = h_{\mathrm{ST}} \left(1 + \frac{1}{2 + \frac{\varphi}{1000}} \right) \tag{28}$$

where h_{ST} is the heat transfer coefficient of the lowest tube row given in eq. (13). The fume gas temperature exiting each pass in the convection section is determined from eq. (29), where $T_{\mathrm{g,inl}}$ and $T_{\mathrm{g,out}}$ are the inlet and the outlet gas temperature in each pass, respectively

$$T_{\mathrm{g,out}} = T_{\mathrm{g,ml}} - \frac{Q_{\mathrm{g}}}{M_{\mathrm{g}} Cp_{\mathrm{g}}} \tag{29}$$

RESULTS AND DISCUSSIONS

Program validation

The validation of the elaborated calculation program consists in comparing the obtained results with the plant data at different stationary operating conditions of the boiler (tab. 2). The steady-state operating data have been collected by the on-line reading from the boiler. Each steady-state (test) is characterized by a specific operating pressure of the steam boiler. The maintaining of this pressure is achieved manually by actuating the main steam outlet valve. Table 3 summarizes the calculated outlet gas temperature, T_{g3} and the measured one in the boiler on 09/03/2008. As can be seen, good agreement with the measured data is obtained. Indeed, the maximum error in predicting the gas temperature was 12.71%. The discrepancies are probably due to simplifications introduced in the model such as the total emissivity of the furnace tube walls, which is a key parameter in the exact determination of the net rate radiation heat transfer. It is also observed that all the calculated results are higher than the measured ones. This could be due to the heat transfer at the transition between the passes.

Table 2. Operation parameters of the steam boiler

Test No.	Pressure [bar]	Fuel flow rate [kg/s]	Air excess [%]
1	0.5	0.0275	7.1
2	1	0.0275	7
3	1.5	0.0275	7
4	2	0.0275	7
5	2.5	0.0275	7
6	3	0.0275	7
7	3.5	0.0275	6.9
8	4	0.0275	4
9	4.5	0.055	3.8
10	5	0.055	3.8
11	5.5	0.055	3.8
12	6	0.055	3.8
13	6.5	0.055	3.7
14	6.75	0.055	3.7

Table 3. Comparing calculated results with measured data

Test No.	Measured temperature [°C]	Calculated temperature [°C]	Error [%]
1	154	161.2	4.67
2	163	168.71	3.5
3	165	174.86	5.97
4	173	180.11	4.11
5	178	184.72	3.77
6	184	188.85	2.63
7	188	192.6	2.44
8	190	196.16	3.24
9	206	232.19	12.71
10	212	235.03	10.86
11	216	237.69	10.04
12	219	240.2	9.68
13	224	242.57	8.29
14	228	243.7	6.9

Effects of input parameters

In this section, a qualitative analysis is made in order to investigate the influence of the steam boiler input parameters, namely the fuel mass flow rate, air excess factor, ambient temperature, and system operating pressure on gas temperature exiting the boiler, thermal powers exchanged, and boiler efficiency.

Effect of fuel flow rate

Figures 3, 4, and 5 show that the fuel flow rate variation affects considerably the gas temperature at the steam boiler outlet. An increase in the fuel flow rate increases the outlet gas temperature. This effect is explained by an increase in the gas flow rate crossing the convection section. An increase of 10% in the fuel flow rate leads to an increase of 15.973% in the gas flow rate which entails an increase of about 7 °C in the outlet gas temperature.

Figure 6 shows the fuel flow rate effect on the radiation and convection exchanged power in the combustion chamber. It is clear that the heat transferred by radiation is more significant than the heat transferred by convection. Furthermore, the variation in fuel mass flow rate does not have an effect on the radiation heat transfer. The fuel flow rate effect on the heat exchanged power in each pass in the boiler is illustrated in fig. 7. Generally, the flow rate increase enhances the heat exchange in every pass. A significant heat transfer is observed in the second pass, due to the elevated gas temperature relatively to the third pass and due to the big heat exchanged area relatively to the first pass. Figures 8 to 10 show that the steam boiler thermal efficiency decreases with the fuel mass flow rate increases shown in figs. 3 to 5. This decrease is mainly due to the heat rate dissipated by the exhaust gases to the atmosphere. For instance, the increases of 10% in the fuel mass flow rate produce a diminution of 3 to 9% in the boiler efficiency.

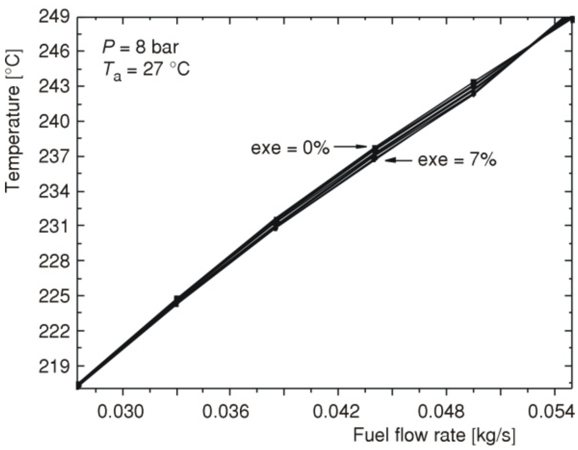

Figure 3. Effect of excess air on the outlet gas temperature

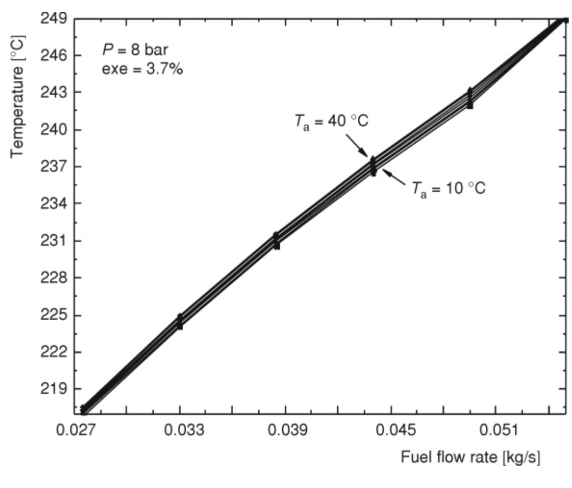

Figure 4. Effect of the ambient temperature on the outlet gas temperature

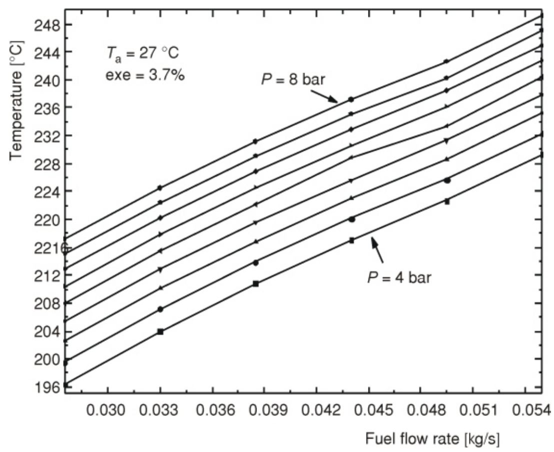

Figure 5. Effect of the operating pressure on the outlet gas temperature

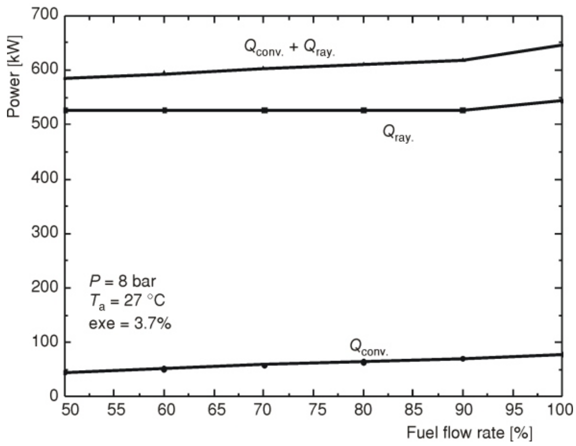

Figure 6. Heat exchanged power variation in the combustion chamber

Effect of air excess factor

Generally, the excess air is used to ensure that the complete combustion takes place. In fact, the excess air is often introduced to control the gas temperature and maintain it within the limits set by the materials in the system. Figure 5 shows that the air excess factor in-

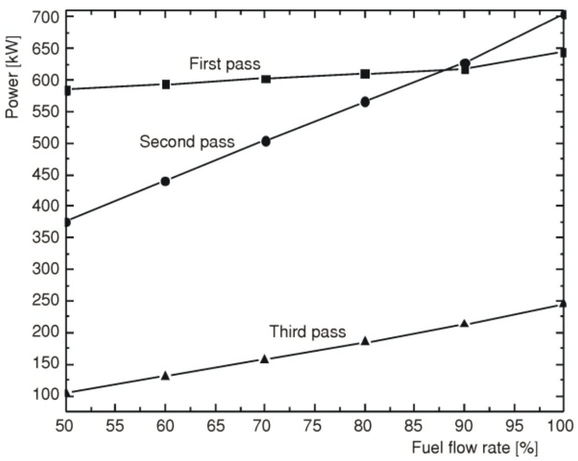

Figure 7. Heat exchanged power variation in each pass in the steam boiler

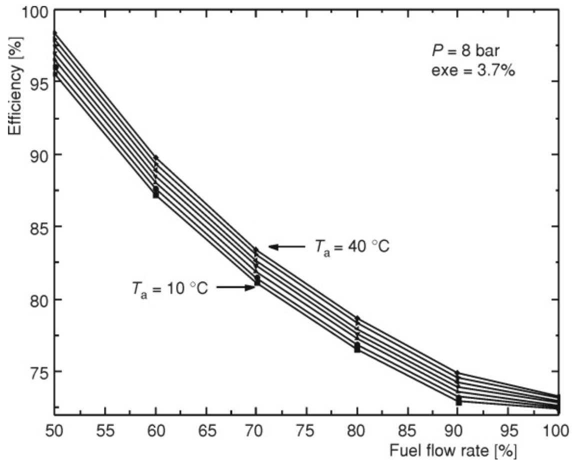

Figure 9. Effect of the ambient temperature on the boiler efficiency

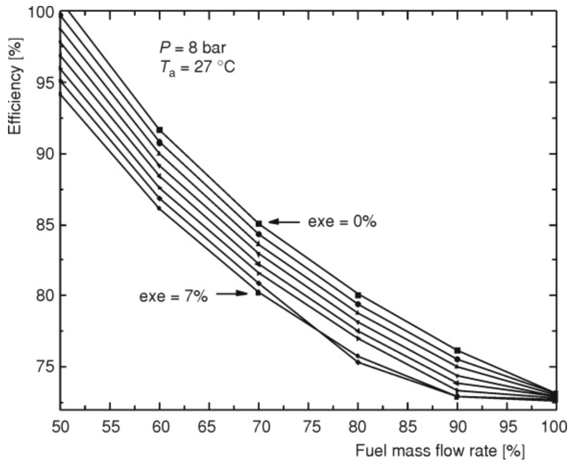

Figure 8. Effect of excess air on the boiler efficiency

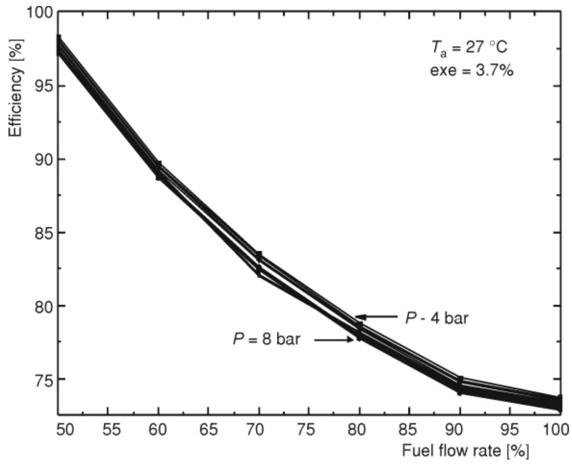

Figure 10. Effect of the operating pressure on the boiler efficiency

fluences slightly the outlet gas temperature. An elevation of 1% of the air excess increases the gas flow rate by about 0.93%, and consequently, weak influence on the convective heat transfer. In addition, the boiler efficiency varies slightly and inversely with the air excess variation (fig. 8).

Effect of ambient temperature

Figure 4 shows that the ambient temperature influence on the gas temperature at the steam boiler outlet is negligible. The increase ambient temperature enhances slightly the boiler thermal efficiency (fig. 9).

Effect of operating pressure

The boiler operating pressure influences the water physical properties, and consequently, the boiling heat transfer. Figure 5 shows that the gas outlet temperature varies proportionally with the system pressure. For example, an increase of 0.5 bar leads to an elevation of 3 °C in the gas temperature. Regarding the boiler efficiency, fig. 10 shows that the variation of the operating pressure does not have a considerable effect on the steam boiler thermal efficiency.

CONCLUSIONS

In this study a simulation of a three-pass fire-tube steam boiler is performed using a computer program. The modeling approach consists of subdividing the system in three heat exchangers. The calculation program is based upon the gas fuel combustion and heat transfer laws from hot gases to the boiling water.

The numerical predictions are compared with the measured data recorded from the plant for different steady-state operating conditions. The comparison shows that the simulation results agree well with the steam boiler operation statements. The maximum error in estimating the gas temperature was found to be acceptable and the deviations with the operating data are mainly due to simplifications introduced in the model.

A sensitivity study was carried out by varying some key input parameters, such as the air excess factor, ambient temperature, and system operating pressure. The analysis shows that the fuel mass flow rate has a big effect on the boiler thermal behavior. The increased mass of the exhaust gas materially lowers the boiler efficiency, knowing that higher exhaust gas temperatures lead to low operating efficiency. The developed program constitutes a versatile tool for engineers and operators to get useful exploitation information and estimate the most efficient operating conditions. However, the limiting embedded hypothesis could be reduced especially if the transition-box, situated between the boiler passes, is modeled. In addition, the furnace modeling can be enhanced with more appropriate advanced models, such as the plug-flow model and multizone furnace model [10].

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ТЕРМОХИДРАУЛИЧНО МОДЕЛОВАЊЕ РАВНОТЕЖНИХ РАДНИХ СТАЊА ПАРНОГ КОТЛА СА ГРЕЈНИМ ЦЕВИМА

У овом раду настојимо да симулирамо термохидраулично понашање парног котла са грејним цевима са три пролаза. Постројење је пројектовано да произведе 4,5 тоне на час засићене паре под притиском од 8 бар, углавном намењене за грејну примену. Развијен је рачунарски програм са циљем да опонаша рад котла у неколико равнотежних радних стања. Програм се заснива на законима преноса топлоте између врелих гасова и унутрашњих зидова грејних цеви. У ложишту котла, пренос топлоте опонашан је коришћењем модела пећи са добрим мешањем. У конвекционој деоници, спроведен је биланс топлоте ради процене измене топлоте између врелих гасова и блокова цеви.

Добијени резултати упоређени су са равнотежним радним подацима разматраног постројења. Компаративна анализа показује да се резултати прорачуна добро слажу са радним подацима котла. Осим тога, изведена је студија осетљивости ради процене утицаја улазних параметара – брзине тока горива, вишка ваздуха, температуре средине и радног притиска – на термичке карактеристике котла.

Кључне речи: термохидраулично моделовање, парни котао са грејним цевима, пренос топлоте зрачењем