

Heat Transfer Study in 3-Pass Fire-Tube Boiler During a Cold Start-up

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Abstract: In this work, we are interested to simulate the transient thermal-hydraulic behaviour of 3-pass fire-tube boiler during a cold start-up. A transient thermal model has been developed based upon energy balance equations of the main components of the boiler and the relevant correlations describing heat transfer phenomena taking place. The variation of the main thermal parameters describing the boiler dynamic behaviour, namely: temperatures of combustion gas, tube walls and water as well as heat transfer rates are also studied. The model validation has been performed by comparing the simulation results against experimental data obtained during the boiler start-up. The comparative study shows a good agreement between the simulation and the boiler operating data. Indeed, the maximum error in predicting the main parameters of the boiler was found to be acceptable and the deviations are mainly due to simplifications introduced in the model. The agreement between the simulation and experimental data reveals the accuracy and the suitability of the proposed model to estimate the dynamic behaviour of the studied boiler and other similar designs.

Keywords: Fire-Tube Boiler, Transient Simulation, Start-up, Heat Transfer

1. Introduction

Fire-tube boilers are widely used for hot water or steam generation for heating and technological purposes [1-5]. Due to the long-time service of these systems, they undergo a series of degradation processes that could expose the system structural integrity to serious hazards and, consequently, huge economic and human losses could occur [6, 7]. The cold start-up is inevitable operation, during the first ignition phase, when the boiler is lighted in the morning or after a long period. It is the operating period before the boiler reaches stable combustion conditions in which the system undergoes an extreme transformation from ambient condition to operating conditions. The start-up involves for the boiler a significant thermal and mechanical stresses in comparison to what they are subjected during the normal condition. Therefore, the boiler structural integrity is subject to fatigue and failure. Accurate prediction of heat transfer in the boiler

during start-up phase is of significant importance for the safe and economic operation [8]. Therefore, a transient thermal-hydraulic modeling of a fire-tube boiler can be a helpful tool in defining the optimal operating conditions in order to keep the structural integrity of the boiler. Nowadays, the safety analysis of thermal systems such as fire-tube boilers is mainly based on numerical simulation. Where, transient simulation of fire-tube boilers has reached remarkable development, through the use of CFD techniques and other advanced thermal-hydraulic system codes [9, 10].

The purpose of this work is the transient simulation a cold start-up of a 3-pass hot water generation fire-tube boiler. A transient thermal model has been developed based upon energy balance equations of the boiler components, namely the combustion gases, the fire-tube metal and the water volume in the boiler.

A set of empirical correlations is used to estimate the heat transfer rates in each part in the system. These correlations

are selected due to their validity over a wide range of operating conditions. A computer calculation program is performed using Matlab software to solve the governing differential equations describing the boiler thermal-hydraulic behaviour. The model validation has been made by comparing calculated results with boiler experimental data. The comparison shows that the calculated results are in good agreement with the boiler operating data. The time variation of main temperatures of the boiler namely: combustion gas, wall and water as well as the heat exchanged power in each pass are estimated.

2. Boiler Description

The studied unit is a single furnace, 03-passes fire-tube boiler used for heating applications and fired with natural gas fuel [11]. It containing several horizontal fire-tubes mounted in a pressure shell that is filled with water. The main characteristics of the boiler are:

- Heating power: 90KW
- Total water amount: 182 liters,
- Furnace length & internal diameter: 836mm/334mm.
- 2nd pass length & internal diameter: 652 mm/146.4 mm.
- 3rd pass length & internal diameter: 826 mm/36.4 mm.

More details about the boiler geometry and operating conditions can be found in Ref. [11]. Figure 1 shows a cross sectional view of the studied boiler.

From heat transfer point of view, the boiler can be divided in two parts: the radiation section and the convection sections [1]. The radiation section, also known as the first pass, is a

horizontal cylindrical furnace in which the Fuel and air are force into the furnace by the burner to produce heat. Heat transfer within the radiation section is mainly by radiation, although convection may contribute to less than 10% [12]. From the radiation section, the combustion gases flow out toward the convection section, crossing the 2nd and the 3rd pass. The convection section tubes are arranged as horizontal tube banks located above the combustion chamber, and they recover additional heat from the furnace gases.

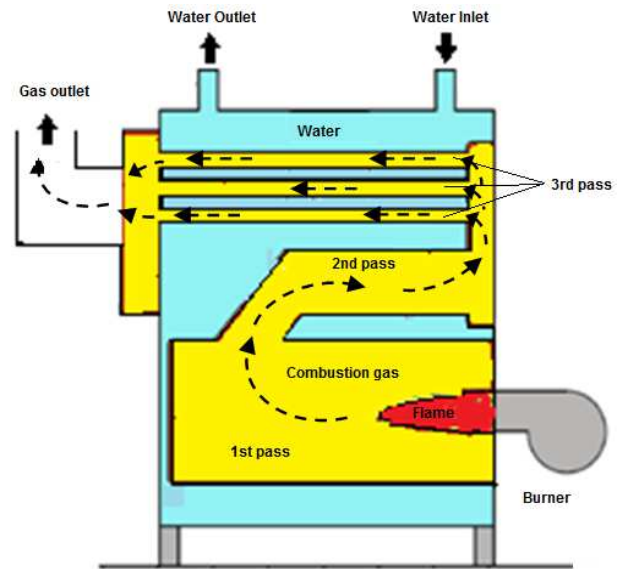


Figure 1. Longitudinal cross sectional view of the 03-pass fire-tube boiler [11].

3. Boiler Modelling and Simulation

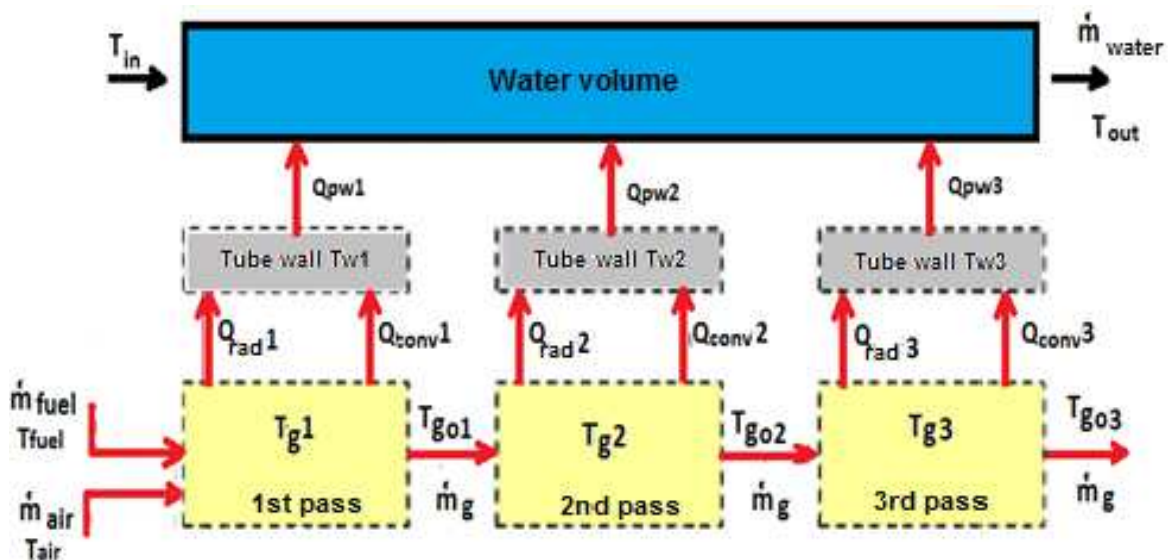


Figure 2. Modelling topology proposed for the 03-pass fire-tube boiler.

The modeling approach presented in the scope of this work is similar to that followed by Bisetto *et al.* [8, 9, 11]. It consists in dividing the boiler in seven control volumes (Figure 2), each one represent a specific state of the boiler.

For each volume an energy balance is established taking into account the generated and the transferred power in the control volume. The transient thermal-hydraulic behaviour of the fire-tube boiler is described through seven (07)

differential equations corresponding to the main component of the boiler, namely: flue gas in each pass, fire tube walls and the water mass. Energy balance equations describing the thermal behaviour of the main components of the boiler are obtained according to the first principle of thermodynamics. The following assumptions have been considered in the boiler modeling:

- The temperature of combustion gases is uniform in the control volume.
- the inner tube walls are gray surfaces,

$$(\rho V C_p)_{g1} \frac{dT_{g1}}{dt} = \dot{Q}_{NG} + \dot{Q}_{air} - \dot{Q}_{g1o} - \dot{Q}_{rad(g1-w1)} - \dot{Q}_{conv(g1-p1)} \quad (1)$$

$$\dot{Q}_{NG} = \dot{m}_{NG} [LCV + C_{p,NG} (T_{NG} - T_{ref})]$$

$$\dot{Q}_{air} = \dot{m}_{air} \cdot C_{p,air} (T_{air} - T_{ref})$$

$$\dot{Q}_{g1o} = \dot{m}_{g1} \cdot h_{g1} = \dot{m}_g \cdot C_{p,g1} \cdot (T_{g1} - T_{ref})$$

LCV is the lower calorific value of the fuel and \dot{Q}_{g1o} , is the energy transferred from the combustion chamber to the 2nd pass. The thermal energy exchanged by radiation towards the inner wall of the combustion chamber tube is given by:

$$\dot{Q}_{rad(g1-w1)} = h_{rad} A_{i1} (T_{g1}^4 - T_{w1}^4)$$

The convective heat transferred rate from the combustion gases to the furnace internal wall can be given by:

$$\dot{Q}_{conv(g1-w1)} = h_{conv} \cdot A_{i1} \cdot (T_{g1} - T_{w1})$$

The mean gas temperature of the 1st pass (furnace) is

$$(\rho V C_p)_{w1} \frac{dT_{w1}}{dt} = \dot{Q}_{rad(g1-w1)} + \dot{Q}_{conv(g1-w1)} - \dot{Q}_{conv(w1-water)} \quad (2)$$

$\dot{Q}_{conv(w1-water)}$, is the convective heat transfer rate exchanged at the furnace tube water side between the water and the furnace external wall, expressed by:

$$\dot{Q}_{conv(w1-water)} = h_{c(w1-water)} A_{o1} (T_{w1} - T_{water})$$

3.2. Convection Section (2nd Pass)

The energy balance equation for the combustion gases flowing inside the 2nd pass tube is given by the following equation:

$$(\rho V C_p)_{g2} \frac{dT_{g2}}{dt} = \dot{Q}_{g1o} - \dot{Q}_{g2o} - \dot{Q}_{rad(g2-w2)} - \dot{Q}_{conv(g2-w2)} \quad (3)$$

The exchanged power rates in this case have the same formulation like in the 1st pass as:

$$\dot{Q}_{g2o} = \dot{m}_g \cdot h_{g2} = \dot{m}_g \cdot C_{p,g2} \cdot (T_{g2} - T_{ref})$$

$$\dot{Q}_{rad(g2-w2)} = h_{rad} \cdot A_{i2} \cdot (T_{g2}^4 - T_{w2}^4)$$

$$\dot{Q}_{conv(g2-w2)} = h_{conv} \cdot A_{i2} \cdot (T_{g2} - T_{w2})$$

The mean gas temperature in the 2nd pass is calculated by the mean temperature between the inlet and outlet temperatures as: $\bar{T}_{g2} = (T_{g1o} + T_{g2o})/2$.

For the wall temperature of the 2nd pass, the energy balance equation is given by Eq. (4). The energy gained from the combustion gases by radiation and convective will be transferred to the flowing water at the tube wall external area.

$$(\rho V C_p)_{w2} \frac{dT_{w2}}{dt} = \dot{Q}_{rad(g2-w2)} + \dot{Q}_{conv(g2-w2)} - \dot{Q}_{conv(w2-water)} \quad (4)$$

- 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 as $\epsilon_p = 0.8$.

3.1. Combustion Chamber (1st Pass)

The boiler radiation section has been modeled using the flowing equation:

expressed by the mean temperature between the adiabatic temperature and the gas temperature at the furnace outlet as: $\bar{T}_{g1} = (T_{ad} + T_{g1o})/2$. We suppose that the gas temperature at the furnace inlet is equal to that of the flame temperature. T_{ad} , refers to the theoretical (fictitious) flame temperature assuming no heat losses through the walls [13]. It is computed by equating the lower heating value of the fuel to the enthalpy of combustion products corresponding to a unit mass fuel:

$$T_{ad} = T_a + \frac{\dot{m}_{NG} LCV + \dot{m}_{Air} (T_{NG} - T_a)}{\dot{m}_g C_{pg}}$$

C_{pg} , is the specific heat capacity of the combustion gases. The mass flowrate of the combustion gases is the summation of the air and the natural gas flowrate as: $\dot{m}_g = \dot{m}_{NG} + \dot{m}_{Air}$.

The energy balance equation for the furnace wall is given by:

$$\dot{Q}_{conv(w2-water)} = h_{c(w2-water)} \cdot A_{o2} \cdot (T_{w2} - T_{water})$$

3.3. Convection Section (3rd Pass)

The energy balance equation for the combustion gas flowing inside the 3rd pass tube bank is given by:

$$(\rho V C_p)_{g3} \frac{dT_{g3}}{dt} = \dot{Q}_{g2o} - \dot{Q}_{g3o} - \dot{Q}_{rad(g3-w3)} - \dot{Q}_{conv(g3-w3)} \quad (5)$$

$$\dot{Q}_{g3o} = \dot{m}_g \cdot h_{g3} = \dot{m}_g C_{p,g3} (T_{g3} - T_{ref})$$

$$\dot{Q}_{rad(g3-w3)} = h_{rad} A_{i3} (T_{g3}^4 - T_{w3}^4)$$

$$\dot{Q}_{conv(g3-w3)} = h_{conv} A_{i3} (T_{g2} - T_{w3})$$

The mean gas temperature in the 3rd pass is averaged between its inlet and outlet as: $\bar{T}_{g2} = (T_{g1o} + T_{g2o})/2$.

The energy balance equation for the 3rd pass wall temperature is expressed by the following equation:

$$(\rho V C_p)_{w3} \frac{dT_{w3}}{dt} = \dot{Q}_{rad(g3-w3)} + \dot{Q}_{conv(g3-w3)} - \dot{Q}_{conv(w3-water)} \quad (6)$$

The exchanged power at the external tube bundle of the 3rd pass is expressed by:

$$\dot{Q}_{conv(w3-water)} = h_{c(w3-water)} \cdot A_{o3} \cdot (T_{w3} - T_{water})$$

3.4. Water Mass

The energy balance equation for the water volume inside the boiler is given by Eq. (7). The energy gained by convective from the tube walls of the three passes will be transferred to the flowing water.

$$(\rho V C_p)_{water} \frac{dT_{water}}{dt} = \dot{Q}_{conv(w1-water)} + \dot{Q}_{conv(w2-water)} + \dot{Q}_{conv(w3-water)} - \dot{Q}_{Water} \quad (7)$$

The heat transferred rate (gained power) to the water side can be expressed as a function of the water temperature at the inlet and the outlet of the boiler.

$$\dot{Q}_{Water} = (\dot{m} C_p)_{water} (T_{water_o} - T_{water_i})$$

3.5. Heat Transfer Coefficients

The radiation heat transfer from the combustion gases to the tube walls is expressed by:

$$h_{rad} = \sigma g_{rad} \frac{T_g^4 - T_w^4}{T_g - T_w}$$

Where, g_{ray} is the total transfer factor for radiation from the gas to the heat sink.

$$g_{rad} = \frac{A_r}{\frac{1}{\varepsilon_w} + \left(\frac{1}{\varepsilon_g} - 1\right)}$$

The convective heat transfer coefficient, at the tubes internal side, depends on the flow geometry, the thermodynamic properties of the fluid (Viscosity, density, conductivity,...), as well as the flow regime (laminar or turbulent). For the laminar flow regime ($Re \leq 2300$), the convection heat transfer coefficient is determined by the B. Spang equation [14]:

$$Nu_g = 4.364 \left[\frac{0.086 \left(Re P_r \frac{D}{L} \right)^{1.33}}{1 + 0.1 P_r \left(Re \frac{D}{L} \right)^{0.83}} \right]$$

For transition and turbulent flow regimes ($Re > 2300$), Petukhov Correlation [14] was used, where all properties of the fluid are evaluated at the mean temperature $(T_g + T_w)/2$.

$$Nu_g = \left(\frac{f_g}{8} \right) Re P_r \left/ \left[1 + 12.7 \sqrt{\frac{f_g}{8}} (P_r^{2/3} - 1) \right] \right.$$

To apply the Petukhov correlation in the tube inlet region and for the transition regime, Gnielinski's modified equation is used by replacing Reynolds number by $(Re - 1000)$:

$$Nu_g = \frac{\left(\frac{f_g}{8} \right) (Re - 1000) P_r}{1 + 12.7 \sqrt{\frac{f_g}{8}} (P_r^{2/3} - 1)}$$

This correlation is used to estimate the convective heat transfer coefficient between the flow gases and the tube wall of the combustion chamber; it is multiplied by a correction coefficient which takes into account the D/L ratio.

$$Nu_g = \frac{\left(\frac{f_g}{8} \right) (Re - 1000) P_r}{1 + 12.7 \sqrt{\frac{f_g}{8}} (P_r^{2/3} - 1)} \left[1 + \left(\frac{D}{L} \right)^{2/3} \right]$$

At the water side, the convective heat transfer coefficient is determined from the water flow regime by:

$$Nu_{water} = \frac{h_f}{D_o} \left\{ \left[0.6 + 0.387 \left(\frac{Ra}{1 + \left(\frac{0.559}{Pr} \right)^{9/16}} \right)^{16/9} \right]^{1/6} \right\}^2$$

4. Boiler Start-up Simulation

A computer calculation program has been established and developed in Matlab software for solving the heat balance equations and predicting the dynamic behaviour of the boiler during the start-up. The Runge–Kutta 4th/5th-order method was used to solve the governing equations for a total calculation time of 1300s. The initial conditions for the calculation program represent initial starting of the boiler, namely: temperatures of flue gas, tubes wall's and water mass were assumed to the ambient temperature (Figure 3). Also, the time variation of the fuel temperature, inlet water temperature and water flow rate, is introduced in the model as boundary conditions [11].

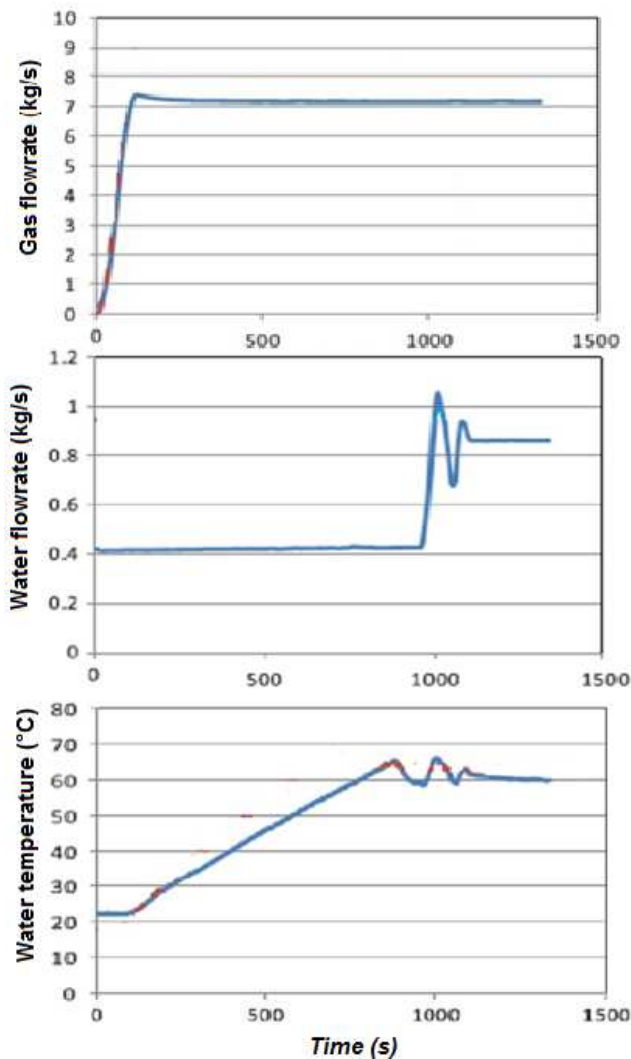


Figure 3. Initial and boundary conditions used in the simulation.

5. Results and Discussion

5.1. Modeling Validation

In order to validate the proposed thermal model, a comparative study was conducted between simulation results and boiler experimental data given in [11]. Figure 4 shows a comparison between the boiler operating data and the calculation of the flue gases temperatures at the 2nd pass, at the 3rd pass outlet and the boiler outlet water temperature during the cold start-up. The comparison shows that the simulation results agree well with the experimental data. The maximum error in predicting the boiler operation statements was found to be acceptable and the deviations with the operating data are mainly due to simplifications introduced in the model.

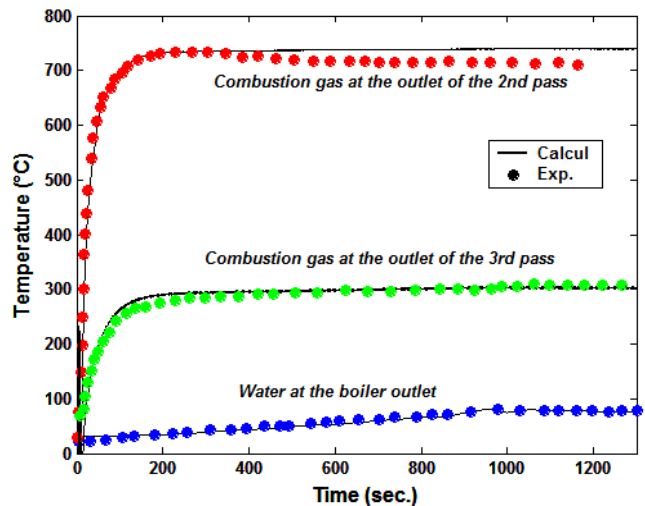


Figure 4. Comparison between the boiler experimental data and the calculated parameters during the boiler start-up.

5.2. Thermal Analysis of the Boiler

5.2.1. Tubes Wall Temperatures

The prediction of wall temperature and precisely that of the combustion chamber is of major importance in boiler safety analysis. Knowing that the metal in this location is subject to sever conditions on both sides. At the gas side, the flame temperature can achieve 1500 to 1800°C and this just a few seconds after the boiler start-up. At the water side, the fluid is at the ambient temperature. Figure 5 illustrates the evolution of the tube wall temperatures of each pass in the boiler. It is clear that the wall temperature is dependent to that of the flue gases in circulation. The maximum temperature difference between the tube wall of the combustion chamber and the water is 20.6°C. In the 2nd pass,

the temperature difference is 15.5°C. While for the 3rd pass, the difference in temperature is 5.47°C. The study shows that the maximum temperature difference between the combustion chamber wall and the water is below the allowable limit and that the boiler operates under normal conditions.

5.2.2. Exchanged Power

The time variation of the thermal power exchanged in each pass in the boiler is given in Figure 6. a. The calculations show that heat exchange is greater at the 1st pass (combustion chamber) with a rate of 51.74%, followed by the third pass by 34.68%. While the 2nd pass provided 12.58% of the total power exchanged. It is clear that the heat exchange in the furnace (0.9 m²) is the most important in the boiler. This is mainly due to the excessively high temperature of the combustion gas at this location that reaches 1030°C. It is found that the 3rd pass is more efficient in terms of heat exchange compared to the 2nd pass.

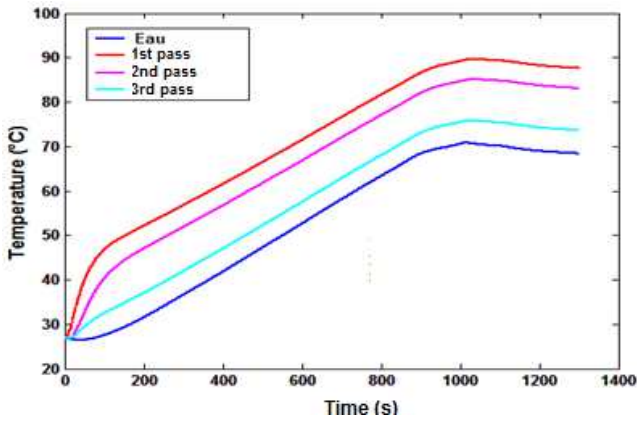
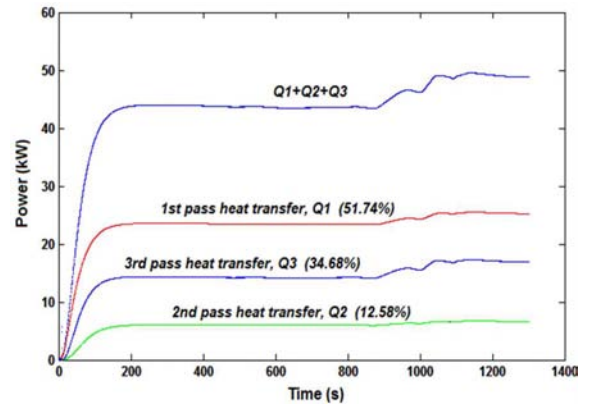
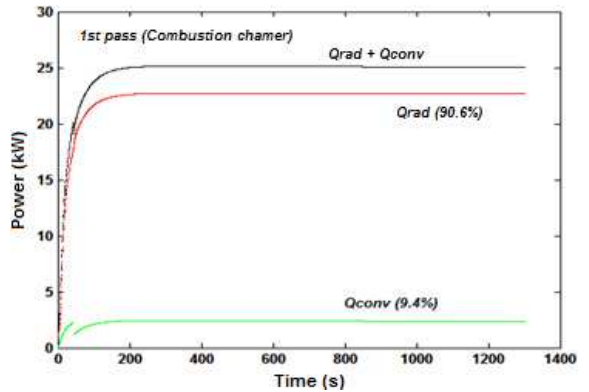


Figure 5. Wall temperature variations in each pass during the boiler start-up.

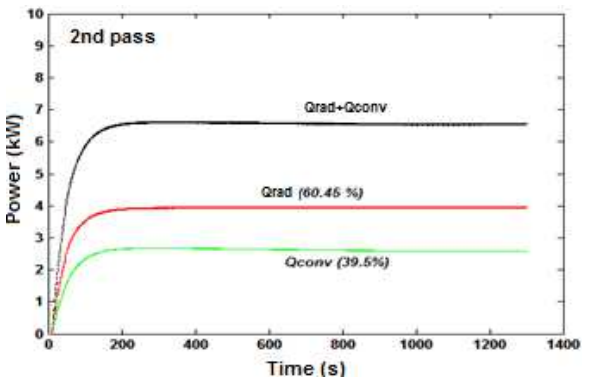
This is not the case in other boilers of 3-pass boilers where the power exchanged decreases along the flow path [1, 2]. This is due to the heat transfer surface of the 3rd pass (2.3m²) which represents the surface of 22 tubes which is larger than that of the 2nd pass (0.32 m²) formed by a single tube. In other hand, this design (01 tube in the 2nd pass) does not favor heat transfer at this location, which translates into a relatively high temperature of 712°C at the exit of the 2nd pass (Figure 5). This indicates that the thermal radiation is present in the 2nd and 3rd pass (Figures 6. c & 6. d). The heat transfer analysis for each pass shows that the heat transfer between the flue gases and the tube walls along the boiler is done simultaneously by radiation and convection. In the combustion chamber (Figure 6. b), thermal radiation is the most dominant mode of heat transfer with a rate of 90.6%, while gas-wall forced convection accounts for only 9.4%. This is explained by the effect of the high temperature of the flame, knowing that the thermal radiation is proportional to the 4th power of the temperature.



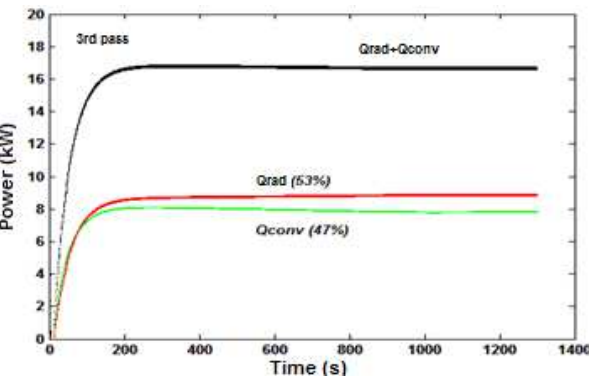
(a)



(b)



(c)



(d)

Figure 6. Convective and radiative power distribution in the boiler during the cold start-up.

In the second pass (Figure 6. c) and following the heat transfer deficit, the flue gas temperature remains high and the radiation remains dominant with an exchange rate of 60.45% compared to 39.5% for the convection. Figure 6. d shows the variation of the exchanged powers by radiation and convection at the 3rd pass. The radiation still exists due to the high temperature of the combustion gases enter the 3rd pass. The thermal radiation slightly exceeds convection with 53% against 47%. Usually, in 03-pass Fire-tube boilers with tube bundle, the thermal radiation is negligible in the 3rd pass.

6. Conclusion

A complete dynamic model of a 3-pass hot water fire-tube boiler has been developed based on the energy balance equations. A computer code has been developed in Matlab software for solving the heat balance equations and predicting the boiler dynamic behaviour during a cold start-up. The modeling validation is performed by comparing the simulation results against experimental data obtained during a cold start-up of the boiler. The comparison shows a reasonable agreement between the simulation results and the experimental data. The agreement between the predicted and experimental data reveals the accuracy and suitability of the proposed model to predict the dynamic behaviour of the studied fire-tube boiler and for other similar designs. The proposed model may be used as an effective tool for safety analysis of such boiler during normal and accidental conditions.

Nomenclature

A	Area (m ²)
C _p	Heat capacity (J/kg °C)
D	Flow hydraulic diameter (m)
f	Friction factor, dimensionless
g	Acceleration due to gravity (m/s ²)
Gr	Grashof number
H	Hydrogen percentage in the fuel
h _{cw}	Convective heat transfer coefficient (W/m ² °C)
h _{rad}	Radiative heat transfer coefficient (W/m ² °C)
k	Thermal conductivity (W/m °C)
L	Tube length (m)
m	Mass moisture percentage of the flue gas
ṁ	Mass flow rate (kg/s)
n	Air index
P _r	Prandtl number
Re	Reynolds number
Q	Heat power (Watt)
T	Temperature (°C)
t	Time (s)
Greek symbols	
β	Thermal expansion coefficient (k ⁻¹)
ρ	Density (kg/m ³)
μ	Dynamic viscosity (kg/ms)
ν	Kinematic viscosity (m ² /s)
ε	Emissivity

σ	Stefan-Boltzman constant (5.6697 × 10 ⁻⁸ W/m ² k ⁴)
Subscripts	
a	Ambient
g	combustion gases
i	Inner, internal
NG	Natural gas
o	Outer, external
r	Relative
rad	Radiation
ref	Reference
conv	Convection
w	Wall

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