A NETWORK APPROACH APPLIED IN MODELLING THE HEAT TRANSFER AND FLUID FLOW IN A SUPERHEATER HEAT EXCHANGER

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ABSTRACT

A 1-D thermo-fluid model for a superheater in coal-fired boilers was developed using the network approach encapsulated in the commercial software Flownex. This model brings to light the relationship between the boiler operations, the thermo-fluid processes and the metal temperatures, since it captures the fluid flow as well as the heat transfer. An understanding of this relationship is crucial at both the design and operational stages of a boiler. At the design stage, the model can be used to minimize the material cost while ensuring adequate performance. For an operational boiler, such a model can be combined with condition monitoring and finite element analysis techniques. In turn, it can be used to improve predictive and preventative maintenance of the plant, to analyse the performance of the heat exchanger as well as doing root cause analyses in case of a failure.

In order to qualify the model, three boiler operation anomalies were modelled using a tubesheet of superheater with a complex geometry and flow arrangement. For each of these anomalies: outer fouling, inner fouling and flow blockage, three cases were studied. The model successfully predicted that the performance of the heat exchanger drops if the outer fouling layer increases. It also showed that the tube metal temperatures increase if the inner fouling layer increases or the flow throttling increases due to the reduction in the amount of coolant (steam) flowing in the tubes.

KEY WORDS

Boiler tube failure; anomalies; flow maldistribution; 1-D network approach; thermo-fluid model; heat transfer.

1. Introduction

Boiler tube leaks are a major contributor to unplanned capacity losses in South African coal fired power plants, especially in superheater and reheater heat exchangers which operate at the high pressures and temperatures. These unplanned capacity losses increase the maintenance costs of the plant, since repairs should be done as well as the loss of revenue due to the down time of the plant. Sometimes it is even possible to repair a tube leak and put the boiler back online, only to be forced offline by another tube leak. This emphasizes the need to identify and correct the root cause of the problem. Some knowledge of the different kinds of tube failures (their visual characteristics) and their causes does assist in the root cause analyses of the problem.

These tube failures include the following: caustic attack, oxygen pitting, stress corrosion cracking, waterside corrosion fatigue, superheater fireside ash corrosion, high-temperature oxidation, water fireside corrosion, fireside corrosion fatigue, dissimilar metal weld failure, tube erosion, short-term overheating, long-term overheating and thermal fatigue [1]. They are a result of operational anomalies such as slagging on the water walls of the boiler, fouling convective pass, tube erosion and scaling inside tubes. Overheating may be caused by a gradual build-up of oxide scales on the inner tube surfaces or by reduced steam flow due to blockages. This may also affect the steam flow and temperature distributions into the outlet headers. Such flow and temperature maldistribution may lead to additional thermal stresses. Furthermore, fatigue may be caused by transient operations such as start-up and shutdown. Nowadays, the increasing need for load changes and low load operation of coal fired power plants implies that there is an increased risk of failure from thermal fatigue and creep-fatigue, where these mechanisms interact. This could significantly reduce the lifetime of components.

Thus, as part of predictive and preventative maintenance it is necessary to understand the relationship between the boiler operations, the thermo-fluid processes and the metal temperatures. For this purpose, mathematical models are
Thermo-fluid process models are often employed for this application but in many instances the complete heat exchanger is lumped together and viewed as a generally counter-current flow, co-current flow, cross-flow or hybrid flow arrangement. Complex flow arrangements are often not accounted for and models are usually not set up in such a way that the flow and temperature maldistribution amongst the tubes and within the headers can be identified. In order to do this, thermo-fluid process models are required that can predict the flow and heat transfer of the steam inside the heat exchanger tubes and headers, the heat conduction through the tube and header walls, as well as the flue gas flow outside the tubes, in an integrated manner. These models can be used during the design stage to minimize the material cost while ensuring adequate performance. For an operational boiler, such models can be combined with condition monitoring and finite element analyses techniques to improve predictive and preventative maintenance of the plant, to analyse the performance of the heat exchanger as well as doing root cause analyses in case of a failure.

Taler et al. [2] presented a numerical model of a dual-tube 12-pass primary steam superheater with a complex flow arrangement that forms part of a 50 MW coal-fired utility boiler. They discretized the heat exchanger into a network of one-dimensional control volumes representing increments in the flow path connected in a way that represents the actual complex cross-flow arrangement. For each pass the two tubes in parallel were lumped together and therefore modelled as a single larger tube with equivalent flow and heat transfer areas. This implied that flow maldistribution between the two parallel tubes cannot be captured. Their model employed the one dimensional (1-D) partial differential equations for mass and energy discretised using the finite volume method (FVM) and solve iteratively using the Guess-Seidel method. In a follow-up paper by Trojan & Taler [3] the same 12-pass superheater was used as a case study to investigate the effects of various anomalies that may occur. These include the effects of fouling on the outside of the tubes, scaling on the inside of the tubes, and uneven gas temperature profile across the width of the superheater.

Coelho [4, 5] modelled the complete convection chamber of a utility boiler. On the flue gas side he applied the porous medium Computational Fluid Dynamics (CFD) approach that was proposed by Patankar & Spalding [6]. He therefore solved the mass, energy and momentum conservation equations simultaneously and thereby obtained a fully coupled flow and heat transfer solution on the flue gas side. The heat transfer from the flue gas to the steam flowing inside the tubes were taken into account by following the flow path of the steam through the various control volumes and performing heat balances to account

This paper describes the application of a one-dimensional thermo-fluid network approach to model a tubesheet of superheater heat exchanger with complex flow arrangement, including the effects of flow and temperature maldistribution. The thermo-fluid network solver will be applied using the commercial software package Flownex® [7].

2. Flownex

The integrated process modelling capability encapsulated within Flownex is based on a network methodology which entails the simultaneous solution of the transient one-dimensional forms of the conservation equations for mass, energy and momentum, combined with the applicable closure relations, boundary values and initial values. The closure relations include models for the component specific characteristics and all the modes of heat transfer as well as the fluid property relationships. The network methodology can be described in terms of the node and element schematic shown in Figure 1.

![Figure 1: Network of nodes and elements applied in Flownex.](image)

In this approach an element is essentially a control volume which may represent any type of physical component such as a pipe, valve, heat exchanger, boiler, turbine, etc. Each element has one inlet and one outlet and the properties within the element are assumed to be represented by a single weighted average value between the inlet and the outlet. An element may also represent a single subdivision or increment of a physical component that is discretized into a number of control volumes. It may therefore represent a pipe increment or heat exchanger increment.

The nodes represent the connection points between elements, which may also be a physical reservoir or tank. Nodes may therefore have multiple inlets and outlets with the properties within a node assumed to be homogeneous and represented by a single averaged value. In the solution of the integrated network all the fluid volume, and therefore
all the fluid mass, is assumed to be contained within the nodes, but the net change of momentum between all the inlets and outlets of a node is assumed to be negligible. Mass and energy may also be added to or removed from a node via mass sources or sinks or via power and heat transfer terms.

The FVM is applied in the discretization of the conservation partial differential equations. The spatially integrated transient partial differential forms of the mass and energy conservation equations without internal sources are solved for each node are respectively given in simplified format by

\[
\frac{\partial \rho}{\partial t} = \frac{1}{V} \left( \sum \dot{m}_{i} - \sum \dot{m}_{e} \right) \tag{1}
\]

\[
\frac{\partial h_{b}}{\partial t} = \frac{1}{\rho V} \left( \left( \sum \dot{m}_{i} \left( h_{w} + g z_{i} \right) \right) - \left( -\sum \dot{m}_{e} \left( h_{w} + g z_{e} \right) \right) \right) + \dot{Q} - \dot{W} + V \left( \frac{\partial \rho}{\partial t} - \frac{\rho \partial v}{\partial t} \right) \tag{2}
\]

Here \( \rho \) is the density and \( h_{w} \) the specific stagnation or total enthalpy of the fluid defined as \( h + \frac{1}{2} v^{2} \). \( h \) is the specific static enthalpy, \( v \) the weighted average velocity in the control volume, \( V \) the volume, \( t \) the time, \( \dot{m} \) the mass flow rate, \( z \) the elevation and \( p \) the static pressure. Wherever there is no subscript it refers to the averaged property value within the control volume, while the subscripts \( i \) and \( e \) refer to the properties at the inlets and outlets respectively. \( \dot{Q} \) is the rate of heat transfer to the control volume, \( \dot{W} \) is the power output from the control volume and \( g \) is the gravitational acceleration.

The transient one-dimensional form of the momentum conservation equation (for incompressible flow) that is solved for each element is given by

\[
\frac{\partial \dot{m}}{\partial t} = \frac{A}{L} \left( \left( p_{w} - p_{w} \right) + \rho g \left( z_{i} - z_{e} \right) \right) + \frac{\Delta p_{ow} - \Delta p_{ol}}{L} v + \frac{\partial p}{\partial t} \tag{3}
\]

Here \( p_{w} \) is the total pressure defined as \( p + \frac{1}{2} \rho v^{2} \), \( A \) the average cross-sectional flow area, \( L \) the representative length and \( \Delta p_{ow} \) and \( \Delta p_{ol} \) are the total pressure increase or decrease due to work and losses respectively. For compressible and homogeneous two-phase flows equation (3) will contain additional terms.

For elements such as pipes, ducts, bends and valves the total pressure loss term is often expressed in the form of a loss coefficient, \( K \) multiplied by the dynamic pressure represented as follows

\[
\Delta p_{ol} = K \left( \frac{1}{2} \right) \rho v^{2} \tag{4}
\]

Equations (1), (2) and (3) may be integrated over a discrete time step \( \Delta t \) using an Euler integration with a weighting factor between 0.0 (fully explicit) and 1.0 (fully implicit) for the source terms at the previous time step and current time step. This therefore determines the degree of forward or backward Euler integration. A weighting factor of 0.7 is often applied since it produces a good balance between accuracy and stability.

The numerical solution scheme applied in Flownex is an Implicit Pressure Correction Method (IPCM). It is based on combining the mass and momentum conservation equations to obtain a total pressure matrix which, after a Newton-Raphson-type linearization, produces a total pressure correction solution matrix. A more detailed description of the numerical solution scheme is presented by Greyvenstein [8]. Besides solving the transient form of the conservation equations, Flownex also has the ability to directly solve the steady-state form of the equations, i.e. with the \( \partial / \partial t \) terms set equal to zero.

3. Case study: the superheater

To demonstrate the application of the network approach in modelling cross-flow heat exchangers, a tubesheet of the primary superheater of a 50 MW coal fired power plant is modelled. This is the heat exchanger shown on the right in Figure 2 and is the same as the one presented by Taler et al. [2]. It consists of 12 dual-tube passes in a cross-flow configuration. Each pass consists of two tubes in “parallel” and these inline tubes are connected to same inlet and outlet headers forming one tubesheet. Therefore a tubesheet is made up of 24 tube passes with a complex arrangement along the length of the flue gas flow path.

The inlet header is connected to the 8th pass from the flue gas inlet side and the tubes then follow the complex arrangement with the outlet header connected to the 1st pass at the flue gas inlet side. The superheater consists of 74 such tubesheets in parallel across the width of the flue gas duct, and the width of the flue gas flow path associated with each tubesheet is 104 mm. The tubes all have an outer diameter of 42 mm and wall thickness of 5 mm. The heights of the first and the last tube with respect to the flue gas flow are 5.34 m and 4.46 m, respectively. The height of tubes in between is linearly distributed.
At nominal operation the steam capacity of the boiler is $210 \times 10^3$ kg/h. The steam parameters are: pressure of 9.61 MPa and temperature of 540 °C.

4. The model

4.1. Flow element network

Figure 3 shows a schematic of the one-dimensional network model of a single tubesheet. It consists of 24 rows of tubes along the flue gas flow path, with each of the tubes discretized into five increments along the length of the tube. The specific number of five increments is for illustrative purposes only and therefore in this case a single tubesheet is represented by 120 individual heat exchanger increments. A representation of one such a heat exchanger increment is shown in Figure 4. The flow path layout on the steam side reflects the complex layout shown in Figure 2.

From the single node representing the extraction point on the inlet header the flow splits into two separate nodes, each representing the inlet of one of the two tubes in the 8th pass. Each of the two tube outlets in the 1st pass are also represented by a node which then converge into a single node representing the connection to the outlet header. The gauge-like symbols connected to the inlet and outlet nodes represent boundary values. Although the outlets of the two tubes converge into a single node, there may be different resultant mass flow rates and temperatures emanating from each. We may therefore detect flow and temperature maldistribution between the two tubes due to different heat transfer and pressure drop conditions that may occur.

On the flue gas side the overall inlet and outlet are each also represented by a single node. At the inlet this implies an assumption of a homogeneous pressure and temperature distribution over the height of the flue gas duct. This assumption is temporarily adopted for demonstration purposes but can be relaxed later on. From the single inlet node the flow splits to individual inlet nodes to each of the flue gas increments along the height of the flue gas duct. The boundary value specified for the single node at the outlet the flue gas flow path is a mass sink. The single node implies that a homogeneous pressure distribution is assumed, which may also be relaxed later on. This single overall outlet node is fed by a convergence from the outlet nodes of each of the flue gas increments. The mass flow rates and temperatures coming from the outlet of each of the flue gas increments may be different due to different heat transfer and pressure drop conditions. Possible flow and temperature maldistribution along the height of the flue gas duct may therefore also be detected.

Figure 2: The convective pass of a 50-MW coal-fired boiler system [13].
Note that for now it is also assumed that there is no mixing in the vertical direction between the flue gas increments as it flows through the heat exchanger. This assumption may also be relaxed by adding interconnecting flue gas flow path elements in the vertical direction between the various inlet and outlet nodes of the flue gas increments. In order to obtain the appropriate detailed flue gas flow pattern within the tubesheet it will require the specification of appropriate representative flow resistances within the horizontal and vertical flow path elements. However, for the purpose of demonstrating the modelling methodology the vertical mixing will be assumed to be negligible, as was done by Taler et al. [2].

4.2. Heat exchanger increment sub-network

The heat exchanger increment shown in Figure 4 is made up of a sub-network of nodes and one-dimensional flow and heat transfer elements as shown schematically in Figure 5.

The heat transfer through the pipe wall is assumed to be axially symmetric and for the thin-walled superheater tube it is assumed that heat conduction is in the radial direction only. The vertical arrows represent the steam ($s_t$) and flue gas ($fg$) flow paths respectively which belong to the tubesheet flow network shown in Figure 3.

The details of the model contained inside each of the increments in the heat exchanger network will be described next. Heat transfer correlations for convection, conduction and radiation are also highlighted.
4.3. Inside convective heat transfer element

The inside convective heat transfer is calculated using the Gnielinski correlation for heat transfer in tube flow [9]. The equations are incorporated in the element using the steam side C# script. This correlation is given by

\[ N_u = \left( \frac{(\xi/8) \text{Re} \text{Pr}}{1 + 12.7 \sqrt{\xi/8} (\text{Pr}^{0.4} - 1)} \right) \times \left[ 1 + (d_i/l_i)^{2/3} \right] \tag{5} \]

where

\[ \xi = (1.8 \log_{10} \text{Re} - 1.5)^{-2} \tag{6} \]

Here, \( N_u \), \( \text{Re} \) and \( \text{Pr} \) are the dimensionless Nusselt, Reynolds and Prandtl numbers, respectively. \( d_i \) and \( l_i \) are the fouled inner tube diameter and the length of the tube respectively.

4.4. Conductive heat transfer elements

The thermal conductivity of the tube material was given by Taler et al. [2] as

\[ k_t = \left( 35.54 + 0.004084 \cdot T \right) \left( -2.0891 \times 10^{-5} \cdot T^2 \right) \tag{7} \]

The thermal conductivity of the outer fouling ash layer was also taken from Taler et al. [2] and its value is \( k_{as} = 0.07 \text{ W/(m K)} \). The inner scaling thermal conductivity was taken from Trojan & Taler [3] and its value is \( k_{asd} = 0.15 \text{ W/(m K)} \).

4.5. Outside convective heat transfer element

The outside convective heat transfer is calculated using the Gnielinski correlation for cross-flow around a single tube [10] incorporated using the flue gas side C# script. This correlation is given by

\[ Nu_l = 0.3 + \sqrt{Nu_{l,lm}^2 + Nu_{l,turb}^2} \tag{8} \]

where

\[ Nu_{l,lm} = 0.664 \sqrt{\text{Re}_f^{1/3} \text{Pr}} \tag{9} \]

and

\[ Nu_{l,turb} = \frac{0.037 \text{Re}^{0.8} \text{Pr}}{1 + 2.443 \text{Re}^{-0.6} (\text{Pr}^{0.4} - 1)} \tag{10} \]

with

\[ l = \frac{\pi}{2} d_o \tag{11} \]

Here, \( Nu_l \), \( Nu_{l,lm} \) and \( Nu_{l,turb} \) are the resultant laminar and turbulent Nusselt numbers. \( l \) is the streamed length and \( d_o \) is the fouled tube outer diameter.

4.6. Outside radiative heat transfer element

The radiation heat transfer from the flue gas to the outer surface of the outer fouling layer is accounted for using Taler & Taler’s [11] radiation heat transfer coefficient given by

\[ h_{ro} = \frac{\sigma \cdot \varepsilon_{eq} \cdot \alpha \cdot S}{\alpha \cdot S + \varepsilon_{eq}} \left( \frac{T_{fg}^4 - T_{fo}^4}{T_{fo}^4 - T_{ro}^4} \right) \tag{12} \]

with \( \sigma \) the Stefan-Boltzmann constant given as \( 5.67 \times 10^{-8} \text{ W/(m}^2 \text{K}^4) \). The equivalent emissivity of the tube, \( \varepsilon_{eq} \) is given by

\[ \varepsilon_{eq} = \frac{2 \varepsilon_w}{2 - \varepsilon_w} \tag{13} \]

with \( \varepsilon_w \) the emissivity of the tube wall or the outer fouling layer. The geometric mean beam length, \( S \) is given by

\[ S = \frac{C V}{A} \tag{14} \]

\( V \) is the volume around the tube occupied by the flue gas and \( A \) is the outer surface area of the fouled tube. According to Jones [12] in many practical systems the constant \( C \) is taken as 3.6. The absorptivity coefficient \( \alpha \) is given by

\[ \alpha = -\frac{\ln(1 - \varepsilon_g)}{S} \tag{15} \]

where \( \varepsilon_g \) is the total emissivity of the gas.
The default radiation heat transfer element within Flownex solves the standard radiation heat transfer equation given by

\[ \dot{Q}_{rad} = A_o F \sigma (T_{fg}^4 - T_{fo}^4) \]  
(16)

with \( A_o \) the outer fouled tube surface area and \( F \) the view factor. This standard radiative heat transfer element is used in the current model but with the view factor given by

\[ F = \frac{\sigma \cdot \varepsilon \cdot \alpha \cdot S}{\alpha \cdot S + \varepsilon \cdot \sigma^{-1}} \]  
(17)

Using this view factor together with eq. (16) is equivalent to employing the effective radiation heat transfer coefficient given by eq. (12).

5. Results

This section presents some results to demonstrate the capability of the model. Steady-state simulation results are presented for three anomalies that often occur in superheaters. These are fouling on the tube outer surface, scaling on the tube inner surface and flow blockage/throttling in one of the pipes. The input conditions for each case are shown in Table 1.

Table 1: Input conditions for a clean superheater at which full load duty is calculated.

<table>
<thead>
<tr>
<th>Details</th>
<th>Units</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet steam temperature</td>
<td>°C</td>
<td>337.7</td>
</tr>
<tr>
<td>Inlet steam pressure</td>
<td>kPa</td>
<td>9600</td>
</tr>
<tr>
<td>Steam mass flow rate</td>
<td>kg/s</td>
<td>46.2</td>
</tr>
<tr>
<td>Inlet flue gas temperature</td>
<td>°C</td>
<td>632.6</td>
</tr>
<tr>
<td>Inlet flue gas pressure</td>
<td>kPa</td>
<td>100</td>
</tr>
<tr>
<td>Flue gas mass flow rate</td>
<td>kg/s</td>
<td>64.5</td>
</tr>
</tbody>
</table>

5.1. Outer fouling

The accumulation of ash particles on the outer surfaces of superheater tubes is one of the major operational anomalies experienced in coal fired power plants. Ash deposition occurs in two ways: slagging and fouling. Slagging is when molten ash is deposited on the outer surface of the tubes and it normally occurs on furnaces and radiant superheaters. Fouling occurs when soot and fly ash deposits on the outer surfaces of tubes in the convective pass. For simplicity we refer to both slagging and fouling of heat exchangers in the convective pass as outer fouling. Outer fouling reduces the amount of heat transferred from the flue gas to the tube, thus reducing the overall effectiveness of the heat exchanger and ultimately also the efficiency of the power plant. During normal plant operations sootblowers are activated to remove the ash build-up.

Using the new model of the 50 MW coal-fired boiler’s primary superheater tubesheet described in the subsections above, we now investigate three cases with different outer fouling layer (OFL) thickness. Even though leading tubes experience more outer fouling than the other tubes, in this work a uniform OFL thickness is assumed for each case. In addition, the same thickness is assumed on all the tubes. The cases are as follows: CASE 1 – OFL thickness of 1.02 mm, CASE 2 – OFL thickness of 2 mm and CASE 3 – OFL thickness of 3 mm. The same inlet conditions are used for all three cases.

**CASE 1 – 1.02 mm OFL thickness**

This is the same case that was compared to Taler et al. [2]. The trends in the respective temperatures are shown in Figure 6.

The difference between the average steam temperature and outer surface temperature of the tube demonstrates that the tube material has a very low thermal resistance compared to that of the outer fouling layer. Heat is transferred from the flue gas to the outer fouling layer outer surface via convection and radiation heat transfer. The contribution of each mode of heat transfer for the fouled tubesheet for CASE 1 is shown in Figure 7. The radiation contribution on the flue gas side is smaller than that of convection and it also decreases with the flue gas temperature. Since this heat exchanger is at the back of the boiler convective pass it is not unexpected that convection contributes more than radiation. The total of the convection and radiation heat transfer is conducted through the outer fouling layer as shown in Figure 7. The amount of heat transferred by a complete superheater with all tubes having an outer fouling layer of 1.02 mm thickness is 10.6 MW.
A sensitivity study was performed to determine the effect of discretization along the length of each tube by comparing the results obtained with one, two, three, four and five increments respectively. The results obtained for the 12th pipe situated in the sixth pass of the tubesheet was chosen for comparative purposes. The results are summarized in Table 2. It can be seen that the change in temperatures from the model with one heat exchanger increment to that with five increments is less than one percent. This implies that for steady-state scenarios the effect of discretization is negligible if uniform inlet flow conditions and uniform outer fouling are assumed. Thus, for convenience a one increment heat exchanger model is used for studying the effect of anomalies in this section.

CASE 2 – 2 mm OFL thickness

The results obtained for CASE 2 are shown in Figure 8. It shows that the increase in the thickness of the outer fouling layer reduces the average steam temperatures. Thus, less heat is extracted from the flue gas. Due to less heat being extracted from the flue gas side, the flue gas exit temperature for CASE 2 is higher. Furthermore, the increased thermal resistance of the outer fouling layer is evident from the increase in the temperature differences across it. The amount of heat transferred in the complete superheater for this case is 7.6 MW.

CASE 3 – 3 mm OFL thickness

The results of CASE 3 are shown in Figure 9. It further demonstrates the trends observed in comparing CASE 2 to CASE 1. For this case the heat transferred in the complete superheater is 5.9 MW.

In comparing the heat transferred in CASE 1, CASE 2 and CASE 3, the drop in the amount of heat transferred per mm increase of outer fouling layer thickness demonstrates the importance of sootblowing applied to superheaters. In order to quantify the loss in heat transfer effectiveness due to the presence of outer fouling in all three cases, the heat transferred in each case is compared to heat that would be transferred in a case with clean tubes for the same inlet conditions. The percentage reduction in heat transfer for each case is shown in Figure 10. If an outer fouling layer of 3 mm thickness is present on all the superheater tubes the impact in this case is roughly a 60% reduction in the overall heat transfer rate.

Table 2: 1.02 mm OFL thickness - The sensitivity of the outlet steam, tube, outer fouled layer and outlet flue gas temperature to level of discretization of the superheater tube sheet with uniform inlet flow conditions.

<table>
<thead>
<tr>
<th>Number of heat exchanger increments</th>
<th>Outlet steam temperature (°C)</th>
<th>Average temperature for pipe 12 on the 6th pass (°C)</th>
<th>Outlet flue gas temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>400.79</td>
<td>373.27</td>
<td>501.89</td>
</tr>
<tr>
<td>2</td>
<td>400.70</td>
<td>372.78</td>
<td>373.70</td>
</tr>
<tr>
<td>3</td>
<td>400.51</td>
<td>372.68</td>
<td>373.69</td>
</tr>
<tr>
<td>4</td>
<td>400.76</td>
<td>372.65</td>
<td>373.57</td>
</tr>
<tr>
<td>5</td>
<td>400.78</td>
<td>372.49</td>
<td>373.41</td>
</tr>
</tbody>
</table>
In a case of a boiler such as this one where there is no reheat downstream, this loss may not be recovered in the air-heater. For a boiler with reheat, the re heater tubes will experience higher temperatures than that for which the material was designed, thus increasing the chances of tube failure due to overheating.

5.2. Scaling: Inner fouling

Superheaters operate at elevated temperatures and sometimes the temperature can be above the design limit. The elevated temperatures provide a conducive environment for oxidation/corrosion of the inner tube wall to occur. This leads to oxide scale build-up on the tube surface and we refer to this as an inner fouling layer (IFL). The inner fouling layer insulates the tube from being cooled by the steam flowing in the tube. Consequently, there is a gradual increase in the tube temperatures as the scale deposit thickness increases. This may lead to long term tube overheating and subsequently lead to tube rupture.

The same primary superheater tubesheet model was used to investigate the performance of the superheater in the presence of inner scaling. Three cases are studied. CASE 1 postulates the same IFL thickness of 2 mm in all the tubes; CASE 2 postulates an IFL of 2 mm thickness in one of the two tubes (but including all the tube passes); and CASE 3 postulates an IFL thickness of 2 mm in the second tube at the outlet side of the steam (Pipe number 2). The same inlet conditions are used for all three cases.

**CASE 1 – IFL thickness of 2 mm in all tubes**

Figure 11 shows the trends in the respective temperatures. The temperature difference across the inner fouling layer demonstrates the impact of the high thermal resistance. Comparing the metal temperatures in Figure 11 to those of the CASE 2 – 2 mm OFL thickness in Figure 8 shows that the presence of an inner fouling layer negatively impacts the metal temperatures. Inner fouling affects the metal temperatures because it insulates the inner tube surface from being cooled by the steam.

**CASE 2 – IFL thickness of 2 mm in one pipe**

The results for CASE 2 are shown Figure 12. The impact on the metal temperatures of the inner fouled tubes is evident. By comparing temperatures in tubes in the same pass it can be seen that the steam temperatures for inner fouled tubes is lower than that of a clean tubes. The outlet steam temperature in pipe number 1 is 441.8 °C compared to pipe number 2 with 397.1 °C. This is despite the fact that the mass flow in the fouled tube (0.263 kg/s) is less than the mass flow in the clean tube (0.361 kg/s). The ability to quantify the distribution of mass flow in the two tubes effectively demonstrates the advantage of modelling each tube separately.
Figure 12: CASE 2 – 2 mm IFL thickness in one pipe: The temperature results on each tube for a one heat exchanger increment model with inner fouling.

CASE 3 – IFL thickness in the second tube at steam outlet

Figure 13 shows the temperature results of a case with an inner fouling layer of 2 mm thickness only in pipe number 2.

The metal temperature of pipe number 2 is higher than that of the other pipes. Despite having the same minimum flow area, the mass flow rate in pipe number 2 for CASE 2 is 0.263 kg/s yet for CASE 3 its 0.304 kg/s. This ≈15% difference in the flow rate is purely due to the difference in heat transferred which impact on the density and other fluid properties. Since the preceding tubes in CASE 2 are fouled, less heat is transferred to the steam compared to CASE 3. This is evident from the outlet steam temperature of 445.4 °C in pipe number 2 for CASE 3.

5.3. Flow blockage/throttling

If the steam flow in one pipe is blocked/throttled, that particular pipe will have less coolant and thus the metal and steam temperatures will increase sharply. This short-term overheating on the metal temperatures may lead to tube ruptures. Flow blockages may possibly result from over attemperation where the water in the superheater tubes partially blocks the flow until it has been evaporated.

The tubesheet model can be employed to predict the effects of throttling the steam flow in one pipe, since both tubes in each pass are modelled. To incorporate this throttling effect, an orifice model that is incorporated in the pipe component is used.

In this study the flow into pipe number 2 is throttled accordingly to analyse the effects on the temperatures. Again, three cases are demonstrated in this section. CASE 1 has an orifice diameter ratio of 0.5, CASE 2 has an orifice diameter ratio of 0.3 and CASE 3 has an orifice diameter ratio of 0.1.

CASE 1 – Orifice diameter ratio of 0.5

Figure 14 shows the effect that flow throttling in one pipe has on the temperatures of the other tubes. There is a noticeable difference on the outlet steam temperatures. The outlet steam temperature of pipe with flow throttling is higher because there is less mass flow in that pipe. The throttled pipe had a mass flow rate of 0.268 kg/s compared to 0.356 kg/s for the clean pipe.

CASE 2 – Orifice diameter ratio of 0.3

With an increase in the flow throttling the difference in steam and metal temperatures between the throttled and
clean pipes becomes even more pronounced as shown in Figure 15.

In this case it can be seen that the metal temperatures are as high as the flue gas temperatures in most of the blocked tubes. The postulated mass flow rate distribution is as follows; 0.021 kg/s for throttled pipe and 0.603 kg/s for the clean pipe.

6. Conclusions

This paper presented the application of the one-dimensional network approach to model a tubesheet of superheater heat exchanger with complex geometry. This approach consists of iteratively solving the mass, energy and momentum equations for each of the one-dimensional increments that are used to construct the complex arrangement of the geometry.

The heat exchanger geometry is discretized along the flue gas flow path as well as along the steam flow path. Each heat exchanger increment contains the geometrical information and thermal resistance characteristics for the steam flow in the tubes as well as the flue gas flow outside of the tubes. Empirical correlations were employed to model both the inner and outer convective heat transfer. An effective heat transfer coefficient was used to account for radiation heat transfer.

To illustrate the application of the model the results of several case studies were presented. The case studies looked at specific anomalies which include outer fouling, inner scaling and flow blockages/throttling. The model successfully predicted that the performance of the heat exchanger drops if the outer fouling layer increases. It also showed that the tube metal temperatures increases if the inner fouling layer increases or the flow throttling increases due to the reduction in the amount of coolant (steam) flowing in the tubes. In the case of flow blockage, the capability of the model to capture maldistribution of flow between the two parallel tubes was illustrated.

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