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EVALUATION OF A THREE-DIMENSIONAL MODEL FOR THE PREDICTION OF HEAT TRANSFER IN POWER STATION BOILERS

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SUMMARY

This paper reports an evaluation of a full three-dimensional mathematical model of a power station boiler with emphasis on the heat transfer phenomena. The model is based on the numerical solution of the equations governing conservation of mass, momentum and energy and transport equations of scalar quantities. The radiative heat transfer is modelled using the discrete transfer method. The model was applied to a power station boiler of the Portuguese Electricity Utility where an experimental study was recently carried out. Measurements of gas temperature, species concentrations and incident heat fluxes to the walls were compared with the predictions for standard operating conditions, partial load operation and low excess air conditions. Comparison of the results with the measurements has shown that in general a good agreement was achieved, but in some cases only qualitative agreement was found. More detailed measurements would be needed to allow a better evaluation of the model and to identify the sources of discrepancy. Nevertheless, the model proved to be a useful tool for the analysis of the heat transfer in utility boilers.

KEY WORDS mathematical modelling; power station boilers; radiative heat transfer; discrete transfer method

INTRODUCTION

Prediction of heat transfer to the walls of a power station boiler is a relevant engineering problem. The walls of the combustion chamber are constituted by vertical welded tubes, the vaporization panels, where the water preheated in the economizer is boiled. The steam is separated from the water in the drum and returns to the boiler where it is superheated in heat exchangers, the superheaters, and delivered to the turbine. In reheat cycles the steam expands in a high-pressure turbine to an intermediate pressure and is passed back to another bank of tubes in the boiler, the reheater. There, it is reheated, usually to the original superheat temperature, and guided to the low-pressure turbine. Among these heat transfer phenomena, in this study we shall examine only those that take place in the combustion chamber. There, radiative heat transfer is the dominant mechanism of heat transfer to the vaporization panels (Viskanta and Mengüç, 1987).

The heat transferred to the vaporization panels dictates their temperature and may influence the lifetime of the tubes. If the heat fluxes are too high and the film of water at the internal surface of the tubes vaporizes the so-called dry-out phenomenon occurs. The convective heat transfer coefficient at the internal surface of the tube drops and the unwetted wall temperature rises sharply. This may cause tube failure. On the other hand, if the heat fluxes are too low, the efficiency of the boiler decreases. Another problem that influences the heat transferred to the walls is the accumulation of ash on the walls of the boiler and surfaces of the heat exchangers. This layer of deposits behaves as an additional thermal resistance, lowering the heat absorbed by the water. These deposits may corrode the tubes if the fuel contains sulphur, asphalthene, vanadium or sodium. Moreover, the emissivity of the wall changes, owing to slag accumulation, and influences the radiative heat transfer.

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A significant number of studies published in the literature aimed at the prediction of heat transfer in boilers have been based on the zonal method (Hottel and Cohen, 1958) or on Monte Carlo techniques. These models rely upon data or assumptions regarding the flow and combustion patterns (see for example, Steward and Gürüz (1974), Bueters *et al.* (1975) and Xü (1981). Other researchers have developed models that simulate the flow, combustion and heat transfer, but where these phenomena are partially or totally decoupled (Gibb, 1986; De Michelle *et al.*, 1989; Steward and Trivic, 1989). However, the simultaneous treatment of all these phenomena using full three-dimensional models is a more powerful technique. Several models of this kind have been used recently (Robinson, 1985; Abbas and Lockwood, 1986; Boyd and Kent, 1986; Görner and Zinser, 1988; Fiveland and Wessel, 1988; Carvalho and Coelho, 1990).

In the present study a full 3-D model is applied to a power station boiler of the Portuguese Electricity Utility. Preliminary studies of this boiler have already been presented (Carvalho *et al.*, 1990, 1991; Coelho and Carvalho, 1992) but no data was available to assess the predictions. Recently, an experimental study was carried out in this boiler (Cassiano *et al.*, 1994). Gas temperature, species concentrations and heat fluxes to the walls were measured at standard operating conditions, at partial load operation and for reduced air/fuel ratio. The available data is used here to assess the mathematical model with emphasis on the heat transfer phenomena.

The mathematical model is described in the next section. The relevant characteristics of the boiler and some numerical details are provided in the third section where the results are also presented and discussed. The main conclusions are summarized in the last section.

MATHEMATICAL MODEL

Turbulence and combustion models

The mathematical model is based on the numerical solution of the density-weighted conservation equations for mass, momentum and energy and transport equations for scalar quantities. Closure of the governing equations requires physical models of turbulence, combustion and heat transfer.

The $k-\varepsilon$ eddy viscosity/diffusivity turbulence model is employed. It is based on the assumption of a linear relationship between the Reynolds stresses and the rate of strain. The turbulent fluxes of scalar quantities are calculated via a gradient diffusion hypothesis. The turbulent viscosity is calculated from the turbulent kinetic energy and its dissipation rate. Transport equations are solved for these two quantities.

The combustion model is based on the assumption that the reaction rates are very fast compared with the mixing rates. Hence, combustion is controlled by diffusion rather than kinetic mechanisms. It is also supposed that the mass diffusion coefficients of all the chemical species and the thermal diffusion coefficient are equal. If the heat released during combustion is significantly higher than the heat transferred to the surroundings, the instantaneous thermochemical state of the mixture may be related with a single strictly conserved scalar variable. The mixture fraction is often the scalar variable chosen for this purpose. The relations between the conserved scalar variables depend upon the model employed.

In this work two different combustion models were employed. In one of them, the simple chemically reacting system (SCRS), combustion is described as a single-step global reaction between the oxidant and the fuel yielding combustion products. In the other one, the instantaneous thermochemical state of the mixture is calculated assuming chemical equilibrium and minimizing the free Gibbs energy (Gordon and McBride, 1971). This second model was used in most of the calculations presented later. In both cases, the instantaneous mass fractions of chemical species are related to the mixture fraction. These relations are not sufficient to calculate the mean values of mass fractions due to the turbulent fluctuations. These fluctuations were taken into account by prescribing the mixture fraction probability density function as a clipped Gaussian (Lockwood and Naguib, 1975).

The gas temperature may be calculated from the enthalpy using well-known thermodynamic concepts. However, the enthalpy does not change linearly with the mixture fraction because a significant part of the

heat released during combustion is transferred by radiation to the walls. Hence, an additional assumption about the relationship between instantaneous values of enthalpy and mixture fraction is needed. Here, the piecewise linear relation suggested by Abou-Ellail *et al.* (1978) was used. The gas density is calculated from the temperature using the equation of state for perfect gases.

Radiation model

In the energy conservation equation a source term due to radiative heat transfer appears. This term must be conveniently modelled. There are several methods available and in this work the discrete transfer method (Lockwood and Shah, 1981) was chosen. It is based on the solution of the radiative heat transfer equation along selected directions. The scattering coefficient was neglected. This is a good assumption for gas or fuel-oil fired boilers.

The physical domain is divided into control volumes. The gas temperature and the absorption coefficient are assumed to be constant in each one of them. Then, the central point P of each cell on the boundary is determined and a semihemisphere centred in P is considered and subdivided into a given number of solid angles. Each solid angle defines a direction along which the radiative transfer equation is solved. Hence, given a point P at the centre of a cell face on the boundary, a radiation beam is fired from P for each one the directions selected above. The path of a radiation beam is followed until it hits another boundary. Let Q_i be the impingement point. Although, in general, Q_i is not the central point of a boundary cell, it is assumed that the radiation intensity at Q_i and at the central point of the cell face containing Q_i are equal. Then, starting from Q_i, the path of the beam is followed back to the origin P and the radiative heat transfer equation is integrated analytically along this path.

The incident radiative heat flux at point P is calculated by adding up the contributions due to all the radiation beams that reach point P. The calculation procedure is iterative because the radiation intensities at the starting points Q_i are not known *a priori*.

The solution of the radiative heat transfer equation requires as a boundary condition knowledge of the wall temperature distribution or heat fluxes at the walls. In a previous study (Coelho and Carvalho, 1992) a coupling algorithm between the physical phenomena occurring in the combustion chamber and in the vaporization panels was proposed. This algorithm allows the simultaneous calculation of the wall temperature distribution and heat fluxes to the walls. It was found that the assumption of a uniform wall temperature is a good one. The heat fluxes at the walls and the gas temperature and composition are almost independent of the wall temperature. Therefore, in this work a constant wall temperature of 700 K was assumed.

The source term of the energy equation due to radiative heat transfer is computed for each control volume by means of summation of the change of radiation intensity of the rays that cross the control volume.

The absorption coefficient of the medium is required to solve the radiative heat transfer equation. The absorption coefficient was calculated from the emissivity of the medium using the mixed grey and clear gas formulation (Hottell and Sarofim, 1967) extended to account for soot. The constants and weighting coefficients determined by Truelove (1976) were employed. A transport equation for soot mass fraction was solved to calculate the contribution of soot to the absorption coefficient. Soot formation (Khan and Greeves, 1974) and soot oxidation (Magnussen and Hjertager, 1977) models available in the literature were used to compute the source term of the transport equation.

Boundary conditions and solution procedure

Close to the walls the Reynolds number is low and the $k-\varepsilon$ turbulence model is not valid. The laws of the wall (Launder and Spalding, 1974) were used to treat this region. Although this approach may be questionable in respect of heat transfer in complex recirculating flows, it does not constitute a serious problem for the present application. In fact, heat transfer to the walls in utility boilers is mainly due to radiation and the convective heat transfer has only a minor contribution.

The numerical solution was accomplished using a finite volume technique and a staggered grid. The governing equations are integrated over each control volume and discretized using finite differences. The central differences discretization scheme was employed except for the convective terms which were discretized using the hybrid central differences/upwind scheme. Coupling between the pressure and velocity fields was handled by a pressure correction method. The SIMPLE solution algorithm was used and the sets of discretized algebraic linear equations were solved by the Gauss–Seidel line-by-line iterative procedure. Solution of the radiative heat transfer equation is incorporated in the SIMPLE algorithm.

RESULTS AND DISCUSSION

The modelled boiler

A power station boiler of the Portuguese Electricity Utility was modelled in this work. It is a natural circulation drum boiler fired from 12 fuel-oil burners placed at the front wall. The burners are arranged in three levels of four burners each. For modelling purposes the fuel is assumed to vaporize instantaneously. At maximum capacity (250 MWe) the boiler produces 771 ton/h of steam at 167 bar and 545 °C. The air and fuel mass flow rates at standard operating conditions are 805 ton/h and 52·7 ton/h, respectively, and O_2 volumetric concentration in flue gases is 1·5%. The results presented here include two additional operating conditions. In one of them the excess air was reduced (250 MWe, 769 ton/h of air, 52·7 ton/h of fuel, 0·75% O_2 in flue gases) and in the other one the boiler was operating at partial load (125 MWe, 459·3 ton/h of air, 28·8 ton/h of fuel, 3·0% O_2 in flue gases).

The model presented here is restricted to the combustion chamber whose simplified geometry is depicted in Figure 1. The inspection ports which provided physical access to the probes used in the experimental work are also shown. Since the geometry is symmetric, only one half of the geometry was considered for modelling purposes with symmetry boundary conditions prescribed at the symmetry plane.

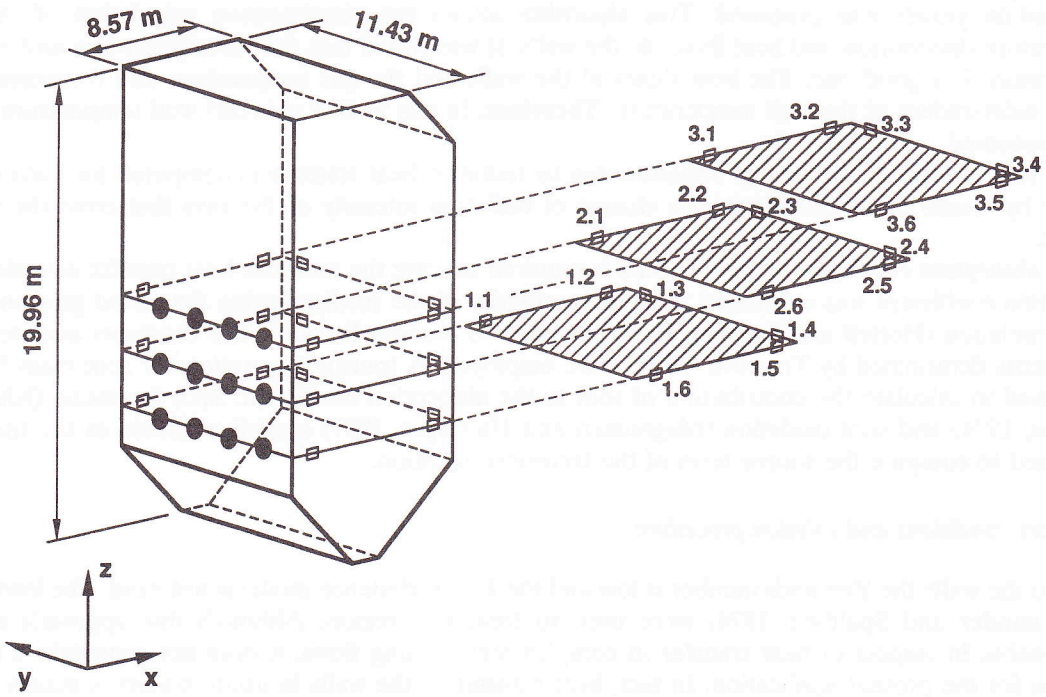

Figure 1. Sketch of the boiler simplified geometry

The experimental work carried out in this boiler is described by Cassiano *et al.* (1994). Gas temperature was measured using a 3·5 m long, double-shielded, water-cooled suction pyrometer which consists of a Pt/Pt−13% Rh thermocouple of 350 μm diameter protected by radiation shields from heat losses to the combustion chamber walls. Major gas species concentrations were obtained by a gas sampling probe connected to dedicated gas analysers by a sampling system. The probe consists of a sampling tube of 2 mm diameter mounted in a water-cooled jacket of 50 mm diameter and 3·5 m length. The total heat flux meter consists of a water-cooled probe which houses a cylindrical block whose front surface is serrated and blackened to give high radiation absorptivity. It is claimed (Cassiano *et al.*, 1994) that the measured species concentrations are close to density weighted averages with 10% of the maximum measured value and heat fluxes have an accuracy of $\pm 5\%$.

Numerical details

The results presented below were obtained using a mesh with $16 \times 34 \times 60$ grid nodes along x, y and z directions, respectively. Two finer grids with $16 \times 54 \times 98$ and $32 \times 34 \times 60$ grid nodes were also used. However, it was found that the differences between the numerical solutions computed using these grids are generally small compared with differences between measurements and predictions. Therefore, the coarser grid was preferred since it allows faster computations.

The calculation of the radiative heat transfer was performed on a coarser mesh with $10 \times 9 \times 20$ control volumes along directions x, y and z, respectively. This allows savings in CPU time with no adverse effect on the solution accuracy (Abbas and Lockwood, 1986). To ensure that this grid is actually sufficient for accurate radiative heat transfer predictions the calculations were repeated using the same grid ($16 \times 34 \times 60$) for the solution of the reactive flow governing equations and for the heat transfer calculations. As it will be shown later, the predicted heat fluxes are very similar to those computed using the coarser grid. Further savings in CPU time were achieved by solving the radiative heat transfer equation only every 10 iterations. The convergence criterion demanded the sum of the normalized absolute residuals over the whole domain to be less than 2×10^{-3} for each dependent variable. Convergence was achieved in about 1000 iterations and 6 hours CPU time (VAX 9000-400).

Gas temperature and species concentrations

Figure 2 shows the comparison between measured and predicted profiles of temperature, CO_2, CO and O_2 mole fractions along the axis of several inspection ports. The profiles computed for different values of the wall emissivity are plotted. The wall emissivity may change along the time due to slag deposition on the walls and its value when the measurements were carried out is not known. Therefore, calculations were performed for three different values of wall emissivity. It can be seen that the influence of the wall emissivity on the predictions is relatively small. The gas temperature increases slightly with the decrease of the wall emissivity, as it had already been concluded in a previous study (Coelho and Carvalho, 1992). For the profiles displayed in Figure 2 it can also be seen that the CO_2 and CO mole fractions increase with the wall emissivity, but the O_2 mole fraction decreases. In general, the results obtained assuming $\epsilon_w = 0{\cdot}65$ are in better agreement with the data than the other ones.

Along the axis of inspection port 1.5, located close to the back wall and just above the ash pit, the gas temperature is underpredicted (see Figure 2(a)). This suggests that the radiation loss was overestimated and this may be due to an overprediction of soot concentration yielding a too high absorption coefficient. Soot formation models at present available still present recognized shortcomings and more research is needed to improve such models. The gas absorption coefficient should be correctly calculated since the predicted gas species volumetric concentrations through this port are in close agreement with the data (see Figure 2(c)).

Inspection ports 2.6 and 3.6 are located at the side wall but very close to the front wall. Figure 2(b) shows that the predicted temperature profile through port 3.6 is underpredicted except in front of the combustion air inlet. The temperature differences observed in the flame region may be due to the simplified burner geometry adopted and to modelling assumptions. However, it should be noticed that

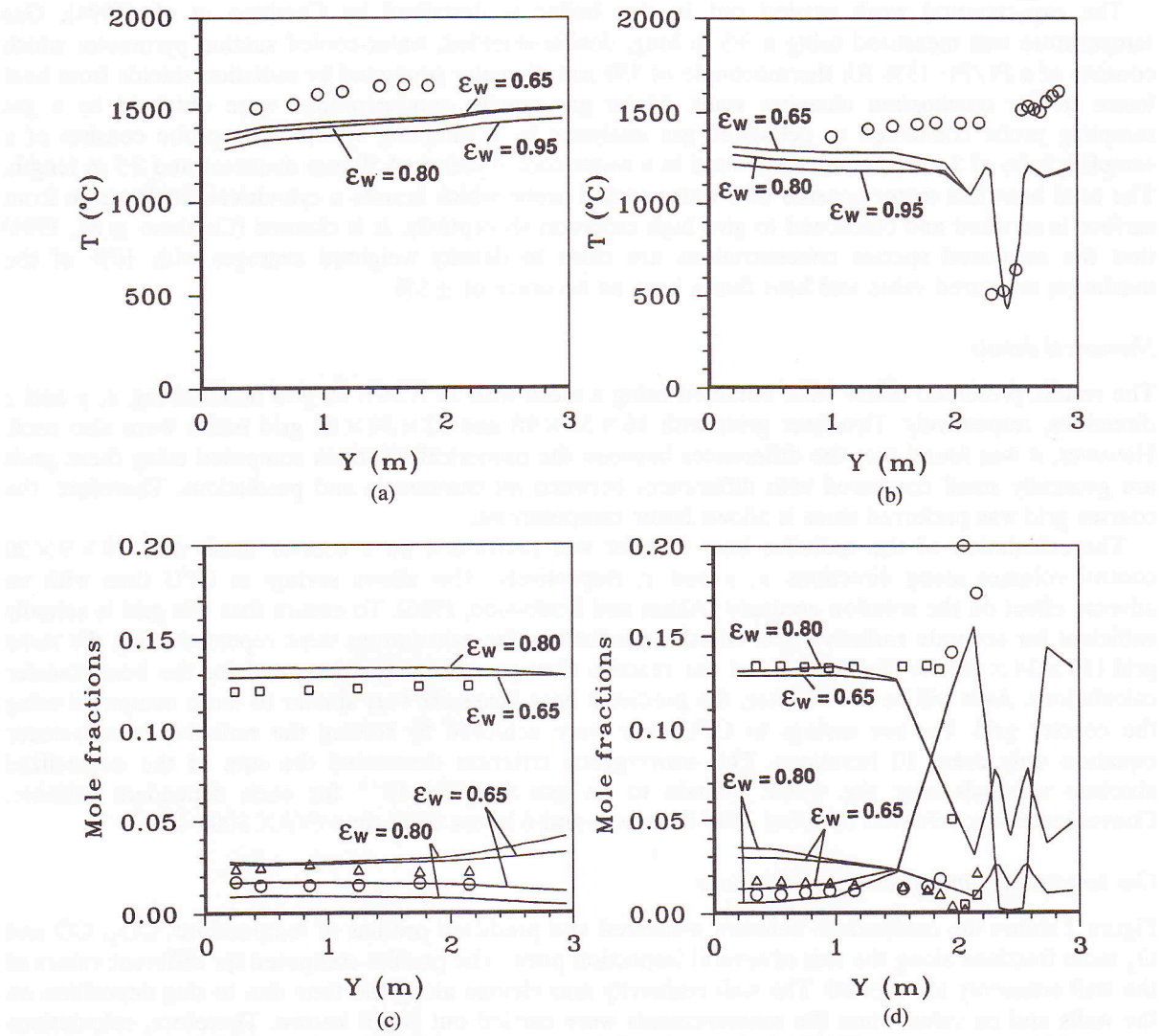

Figure 2. Comparison between predicted (solid lines) and measured (symbols) profiles for different values of the wall emissivity. (a) Temperature profile through port 1.5. (b) Temperature profile through port 3.6. (c) Mole fraction profiles through port 1.5 (□: CO_2; Δ: CO; ○: O_2). (d) Mole fraction profiles through port 2.6 (□: CO_2; Δ: CO; ○: O_2)

this profile crosses the burner axis in a region where the temperature gradients along the axis are very steep. Hence, it is difficult to draw conclusions about the solution accuracy only from the analysis of the temperature profile through port 3.6. It would be extremely useful to have experimental data along the burner axis. However, this is not possible because of limitations in physical access to the boiler.

The computed species concentrations along the axis of inspection port 2.6 are in reasonable agreement with the data (Figure 2). The carbon monoxide mole fraction is overpredicted. The predicted CO_2 and O_2 mole fractions out of the flame region are close to the data and the gradients in front of the combustion air inlet are also simulated well. The predicted gradients, however, are smoother than the experimental ones.

Incident heat fluxes to the walls

The incident heat fluxes calculated using a wall emissivity of 0·65 are shown in Figure 3 together with the available measurements (shown in italic). It can be seen that the measurements reveal an asymmetric

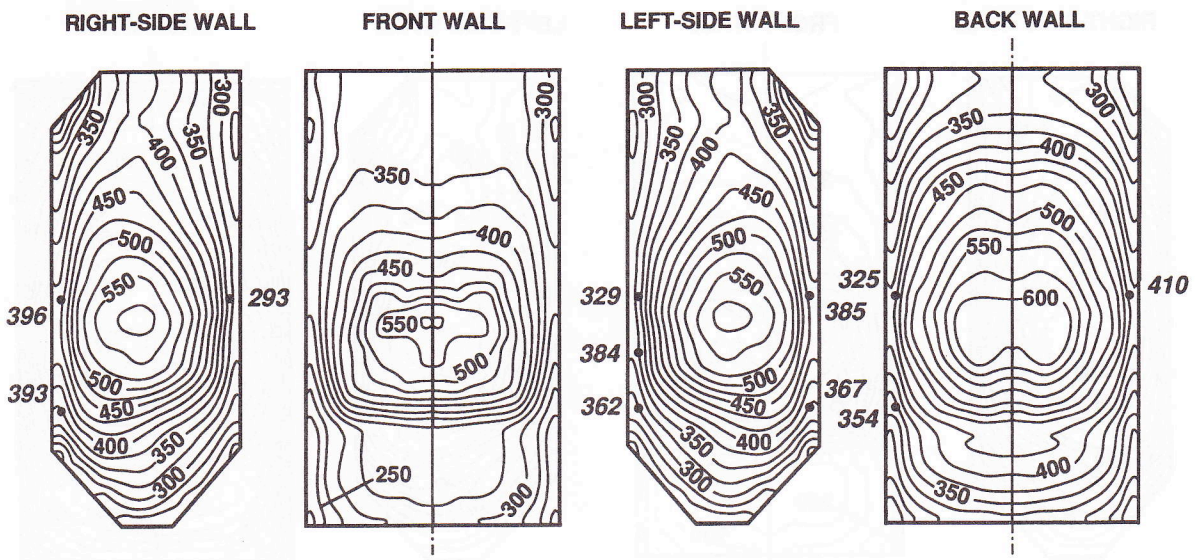

Figure 3. Predicted and measured incident heat fluxes (kW m^{-2}) for $\epsilon_w = 0{\cdot}65$

distribution of heat fluxes. This has been attributed to variations of the air/fuel ratio between the various burners caused by the impossibility of controlling precisely the air and fuel flow rates through each burner (Cassiano *et al.*, 1994). This phenomenon cannot be simulated by the mathematical model, unless the air and fuel mass flow rates were measured individually for each burner. Therefore, the predicted heat fluxes distribution is always symmetric. Regarding this limitation, the predicted results illustrated in Figure 3 are in good agreement with the data. The calculated heat flux at inspection port 1.1 is lower than the measured one and the value computed at port 3.3 is too high. However, owing to symmetry, the calculated heat fluxes for ports 3.3 and 3.4 are equal and the value computed for port 3.4 is close to the measured heat flux. The remaining predictions are close to the experimental values and in the range of uncertainty of the measurements.

The predictions also show that the measured values do not give direct information about the average heat fluxes at the planes of measurement. In fact, since the inspection ports are located very close to the corners and the maxima heat fluxes occur near the centre of the walls, average heat fluxes tend to be higher than the measured values.

To evaluate the influence of grid refinement on radiative heat transfer calculations, a second set of calculations was made using the same grid for the solution of the partial differential conservation equations for mass, momentum, energy and chemical species and for the solution of the radiative heat transfer equation. The predicted incident heat fluxes obtained in this case are displayed in Figure 4. It shows that the predictions are very similar to the previous ones, proving that the coarser grid can be used for radiative heat transfer calculations yielding a grid independent solution. This coarser grid was employed in all the following calculations.

Further calculations were performed assuming wall emissivities of 0·80 and 0·95. The predicted heat fluxes are plotted in Figures 5 and 6, respectively. Figure 5 also includes the measurements given in Figure 3. Comparing Figures 3 and 5 it can be seen that the increase of the wall emissivity from 0·65 to 0·80 has an important influence on the incident heat fluxes. In fact, the incident heat fluxes decrease 50 kW m^{-2} to 75 kW m^{-2} when the emissivity increases. Consequently, there is a much better agreement between the measurements and the predictions for $\epsilon_w = 0{\cdot}65$ than for $\epsilon_w = 0{\cdot}80$. When the wall emissivity further increases to 0·95 the predicted incident heat fluxes continue to decrease, but the difference between the results obtained for $\epsilon_w = 0{\cdot}65$ and $\epsilon_w = 0{\cdot}80$ is much greater than the difference observed for the cases $\epsilon_w = 0{\cdot}80$ and $\epsilon_w = 0{\cdot}95$.

The predictions obtained assuming $\epsilon_w = 0{\cdot}95$ are compared in Figure 6 with the measurements

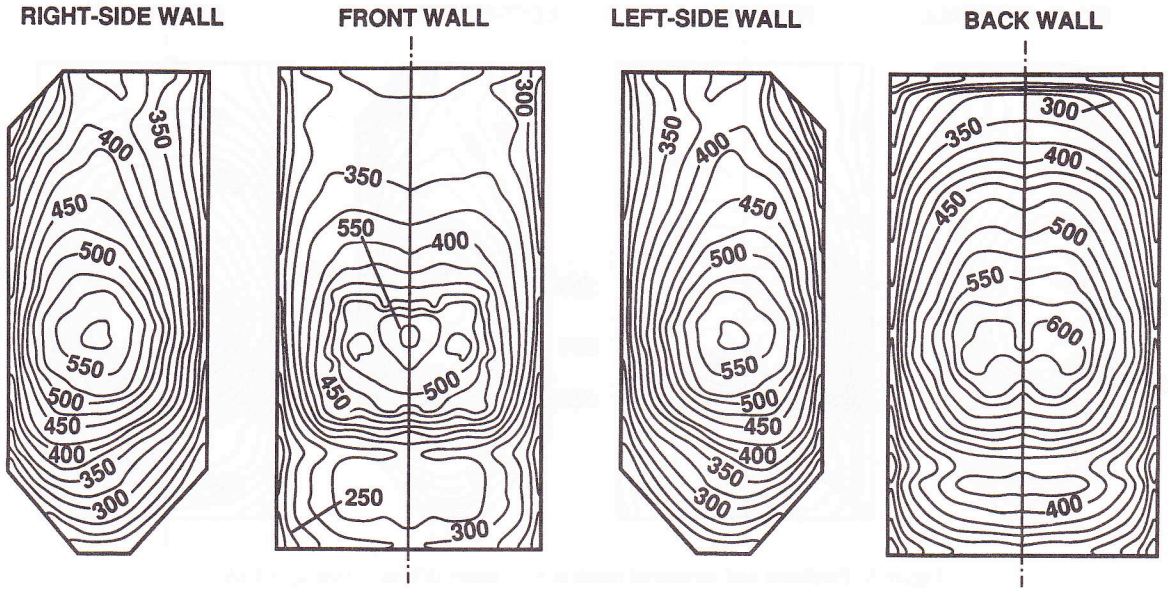

Figure 4. Predicted incident heat fluxes (kW m^{-2}) for $\epsilon_w = 0{\cdot}65$ using the same grid for flow and heat transfer calculations

reported by Cassiano *et al.* (1994) for clean walls. Although the emissivity of the walls when these measurements were made is not known, it is reasonable to assume that it should be close to 1, since in this case the heat flux absorbed by the walls is a maximum. Therefore, the value $\epsilon_w = 0{\cdot}95$ seems a good guess and since the difference between the fluxes computed for $\epsilon_w = 0{\cdot}80$ and $\epsilon_w = 0{\cdot}95$ is relatively small, it is expected that the error involved in this guess is small. The predictions are in good agreement with the data for all but port 3.2, where the heat flux was overestimated by about 30%.

Figure 7 shows the predicted incident heat fluxes to the walls for $\epsilon_w = 0{\cdot}65$ when the simple chemically reacting system (SCRS) combustion model was employed. Using this combustion model the heat fluxes

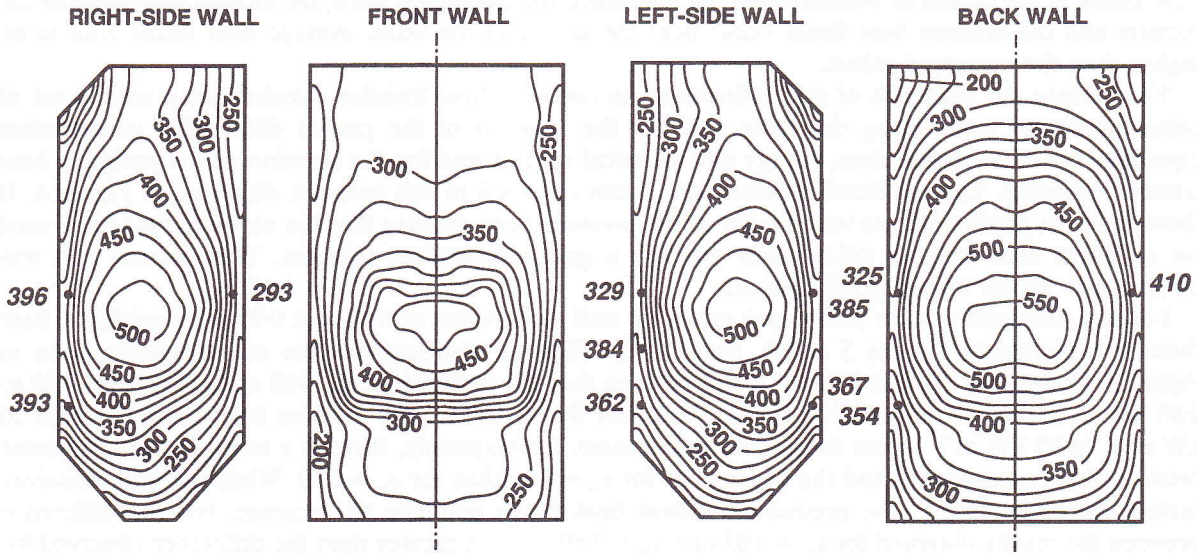

Figure 5. Predicted and measured incident heat fluxes (kW m^{-2}) for $\epsilon_w = 0{\cdot}80$

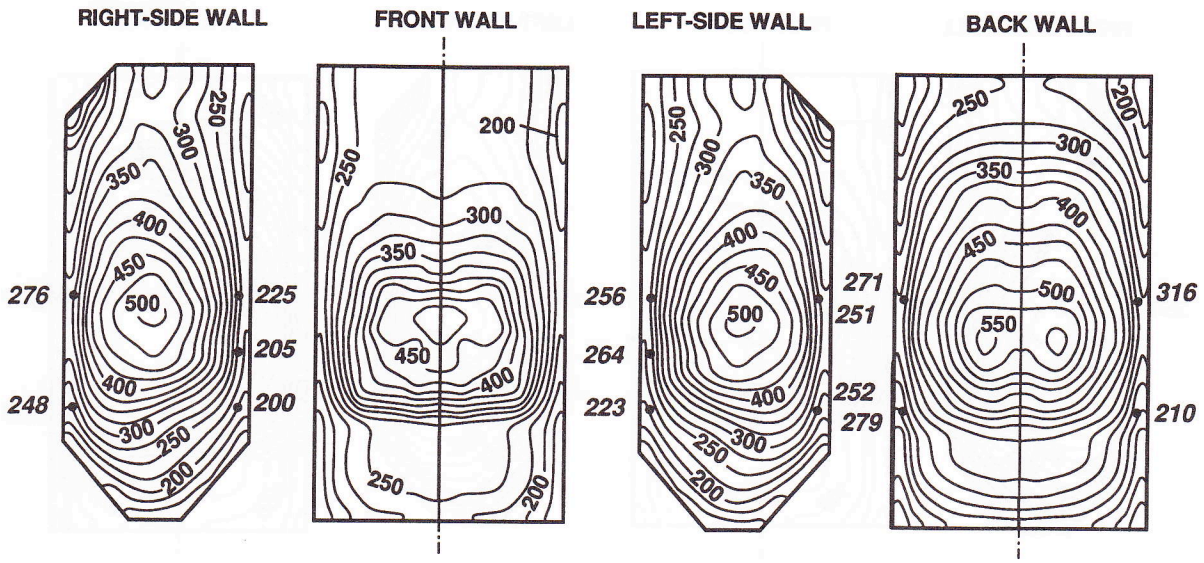

Figure 6. Predicted and measured incident heat fluxes (kW m^{-2}) for $\epsilon_w = 0{\cdot}95$

are significantly higher than before, when the chemical equilibrium model was employed (see Figure 3). This is related to the higher gas temperature predicted using the SCRS model. The calculated heat fluxes are generally much higher than the experimental values except for ports 1.1 and 2.1. Therefore, the chemical equilibrium model should be preferred and it was used for all the following cases.

The influence of decreasing the air/fuel ratio is illustrated in Figure 8 which shows the predicted heat fluxes to the walls when the oxygen volumetric concentration in the flue gases is 0·75% O_2. The computed heat fluxes are very similar to those presented before for standard operating conditions (1·5% O_2 in flue gases). They suggest that the small decrease of oxygen concentration in the flue gases has a negligible influence on the heat fluxes to the walls. Comparison with the available data is not as

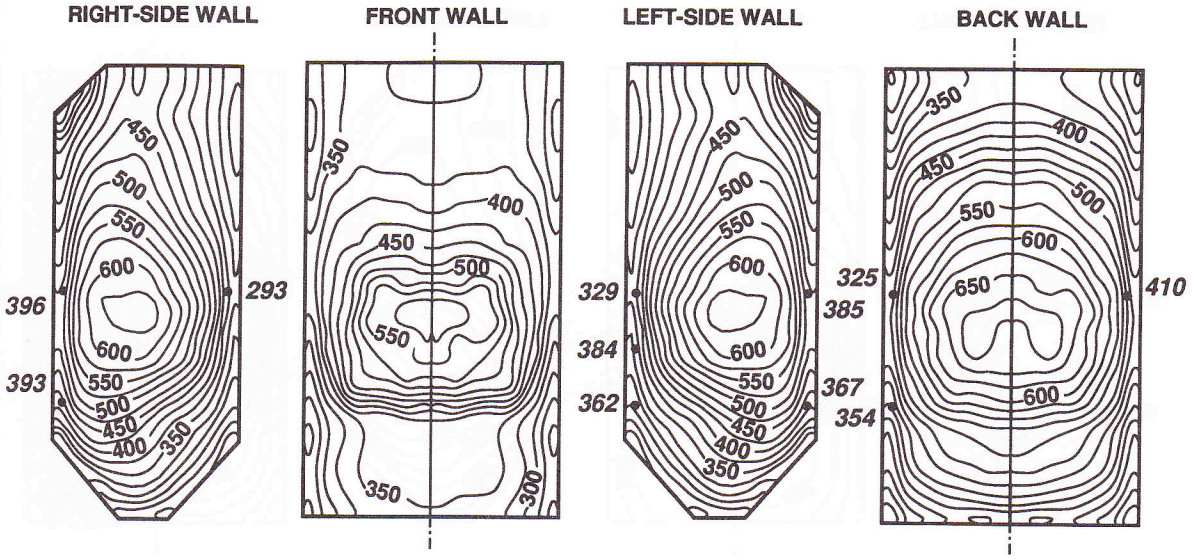

Figure 7. Predicted and measured incident heat fluxes (kW m^{-2}) using the simple chemical reacting system combustion model

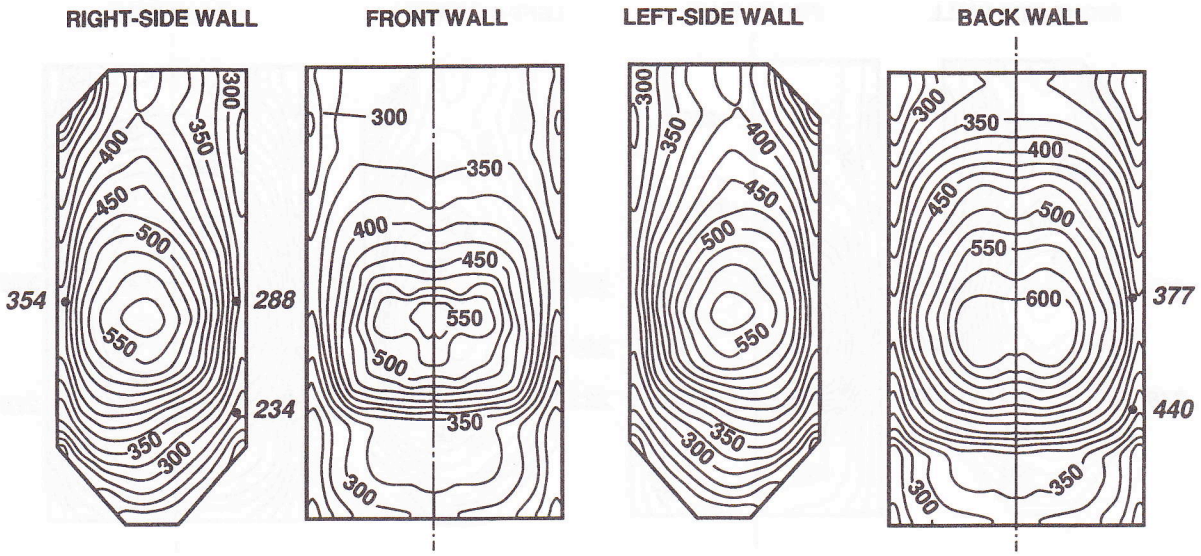

Figure 8. Predicted and measured incident heat fluxes (kW m^{-2}) for 0·75% O_2 in flue gases

satisfactory as in previous cases. The heat fluxes are overpredicted by 30 to 70 kW m^{-2} except at port 1.4 where the measured heat flux is higher than the calculated value.

The predicted heat fluxes distribution at a boiler load of 125 MWe are displayed in Figure 9 together with the available data. Comparison with Figure 3 shows that the reduction of the boiler load yields a significant decrease of the maximum incident heat flux. At partial load, the burners at the top level are out of service. This explains why the maxima heat fluxes occur at a lower level. As a result, the heat fluxes at the top burners level are much smaller than at full load. This is correctly predicted by the model and there is a very good agreement with the data. At the lower level of burners the heat fluxes were underpredicted. However, the heat flux measured at port 1.5 is higher at 125 MWe than at 250 MWe.

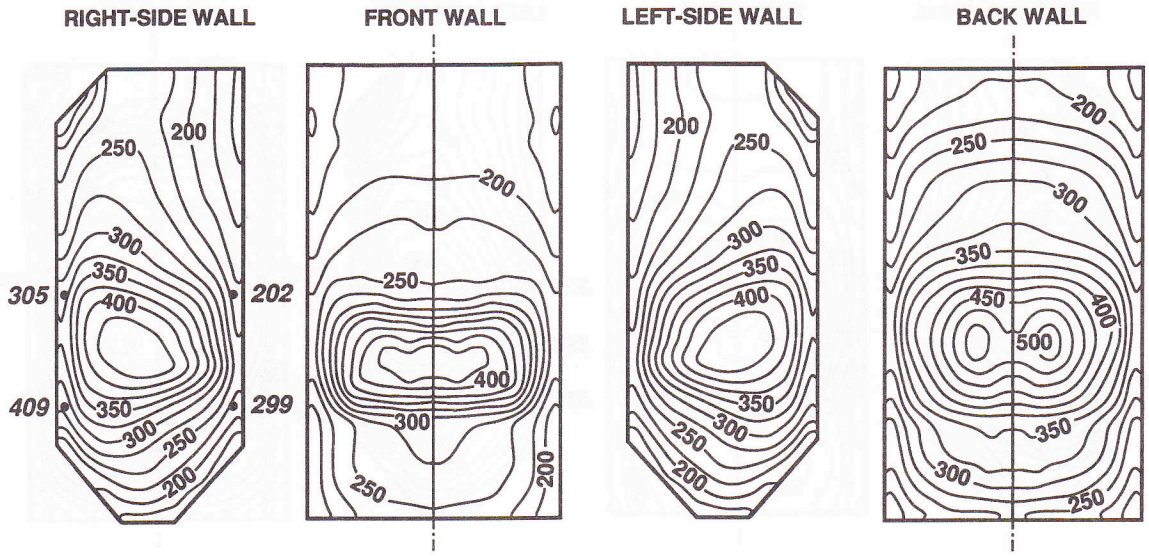

Figure 9. Predicted and measured incident heat fluxes (kW m^{-2}) at partial operation load

This is unlikely to actually occur and, therefore, the high heat flux measured at this port at 125 MWe should be considered doubtful.

Heat fluxes absorbed by the walls

The predicted heat fluxes absorbed by the walls are shown in Figures 10, 11 and 12 for $\epsilon_w = 0{\cdot}65$, $0{\cdot}80$ and $0{\cdot}95$, respectively. It can be seen that the absorbed heat fluxes increase with the wall emissivity. The results support the conclusion drawn in a previous study (Coelho and Carvalho, 1992) where it was found

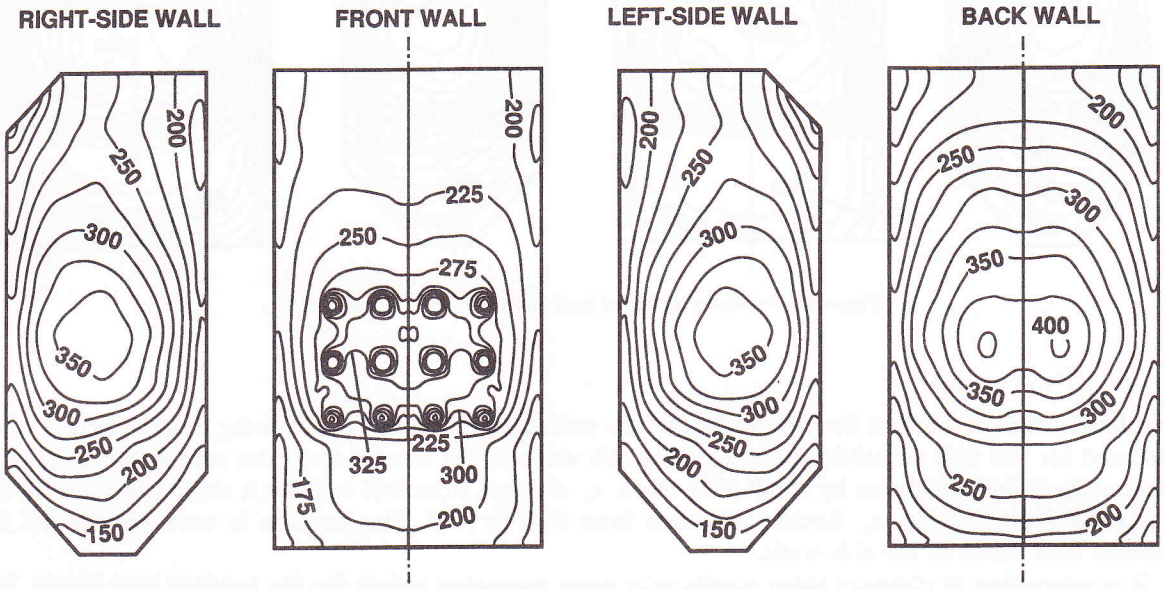

Figure 10. Predicted absorbed heat fluxes (kW m^{-2}) for $\epsilon_w = 0{\cdot}65$

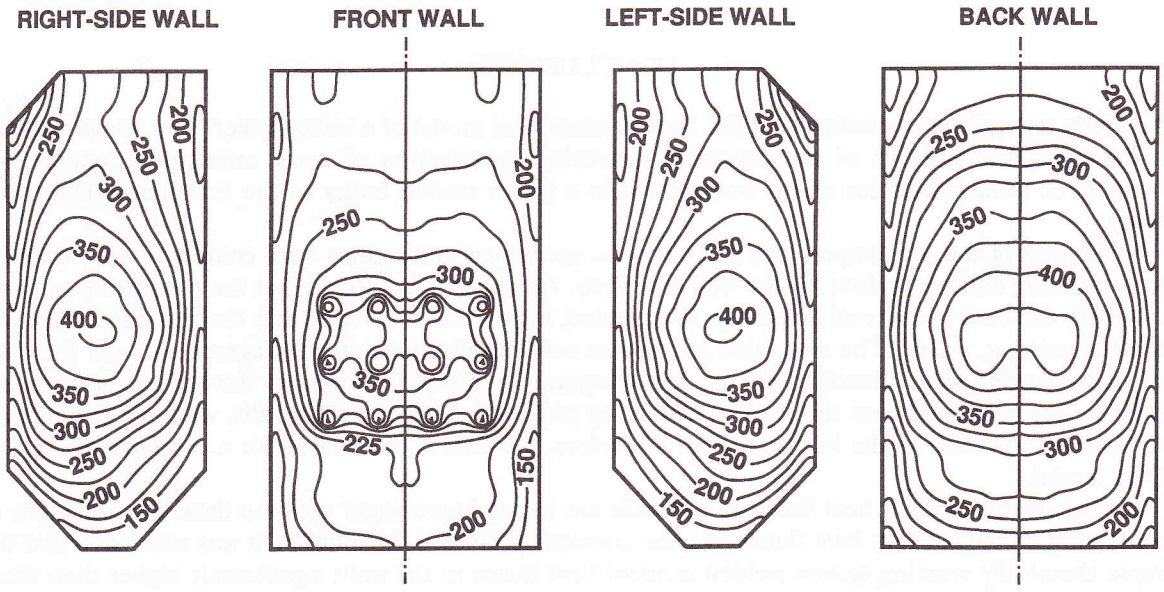

Figure 11. Predicted absorbed heat fluxes (kW m^{-2}) for $\epsilon_w = 0{\cdot}80$

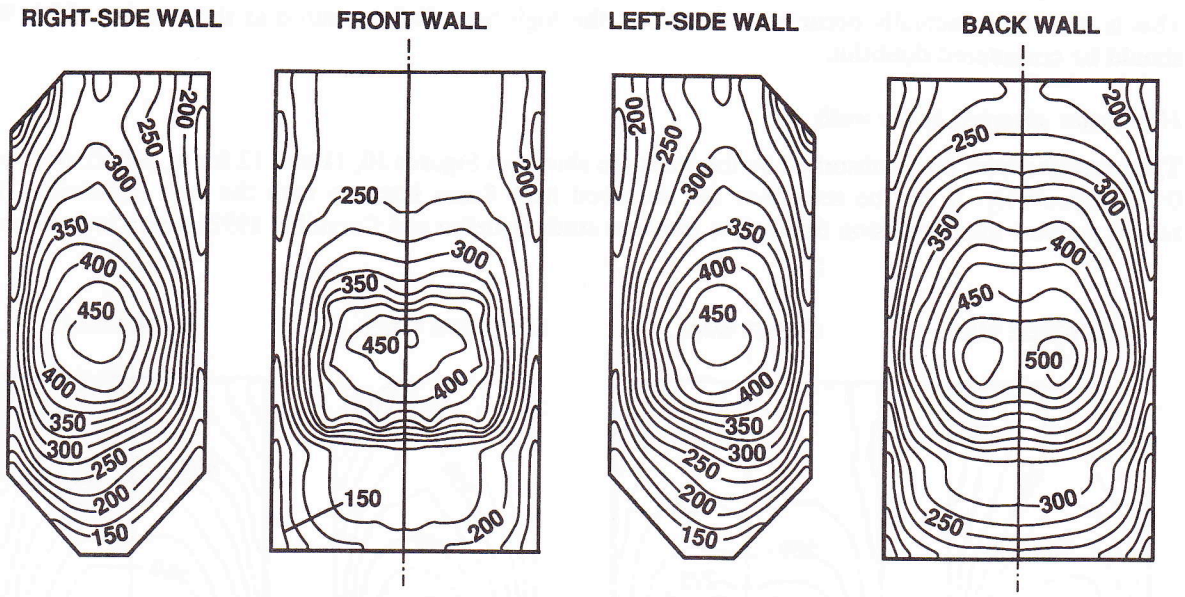

Figure 12. Predicted absorbed heat fluxes (kW m^{-2}) for $\epsilon_w = 0{\cdot}95$

that local errors in the heat fluxes absorbed by the walls up to 10% may occur owing to an incorrect value assumed for the wall emissivity, provided that this value is not too far from the actual wall emissivity. Maxima heat fluxes increase by about 10% when ϵ_w changes from 0·65 to 0·80. A similar increase in the fluxes takes place when ϵ_w further increases from 0·80 to 0·95. The increase is even greater for the maxima heat fluxes at the side walls.

It is interesting to compare these results with those presented before for the incident heat fluxes. The absorbed heat fluxes increase, but the incident heat fluxes decrease with the increase of the wall emissivity.

CONCLUSIONS

This study reports the evaluation of a full three-dimensional model of a utility boiler. The model is based on the numerical solution of the equations governing conservation of mass, momentum, energy and transport of scalar quantities and it was applied to a power station boiler of the Portuguese Electricity Utility.

Predictions of the gas temperature and chemical species concentrations were compared with available data, assuming different values for the wall emissivity. Although it was found that the predictions are only marginally affected by the wall emissivity, as expected, it could be concluded that the best agreement was obtained using $\epsilon_w = 0{\cdot}65$. The computed gas species concentrations are in good agreement with the data, but discrepancies were identified in the gas temperature profiles. However, the only measurements available were obtained near the corners and along profiles very close to the walls, where high gradients are present, especially in the burner region. Therefore, the data is too limited for a complete evaluation of the model.

The calculated incident heat fluxes to the walls are in good agreement with the data. The influence of wall fouling on the incident heat fluxes was also correctly predicted. In addition, it was concluded that the simple chemically reacting system yielded incident heat fluxes to the walls significantly higher than those

calculated using the chemical equilibrium model, which were close to the measurements. Therefore the chemical equilibrium model was preferred.

The influence of the air/fuel ratio and power load on the heat fluxes to the walls were investigated. The correct trends were predicted but in the former case the heat fluxes were overestimated.

On the whole, comparison of the results with the measurements has shown that in general a good agreement was achieved. However, in some cases, for example, decreasing the air/fuel ratio, only qualitative agreement was found. Therefore, there is room for improvement of the model but, owing to the complexity and multiplicity of physical phenomena involved, it is essential to have much more data to identify the sources of discrepancy. Nevertheless, the model proved to be a useful tool for the analysis of heat transfer in utility boilers.

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