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THE APPLICATION OF THE ZONE AND MONTE CARLO METHODS TO PREDICT HEAT TRANSFER IN PULVERIZED COAL-FIRED UTILITY BOILERS

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ABSTRACT

A two-dimensional, multi-zone heat transfer model has been developed for pulverized coal-fired furnaces. A zone method of analysis is used to divide the furnace into a series of horizontal slices. Each slice is divided into two concentric rectangular zones. Radiant heat transfer in the combustion chamber is simulated by a reducing energy beam Monte Carlo technique. The radiation model includes the effects of particulate matter, such as soot, char, and ash particles, as well as nonluminous gases. The furnace walls are considered grey Lambert surfaces, and multiple wall reflections are simulated.

The model has been applied to the Riley Stoker TURBO® Furnace design. The TURBO geometry includes upper and lower rectangular furnace chambers separated by a venturi-shaped restriction. The model includes the effects of both forward and recirculating gas flow within the lower furnace.

The furnace model has been verified using detailed field measurements from a large coal-fired utility boiler. Local variations in gas temperature and wall heat flux are predicted. Model predictions of the sensitivity of radiant furnace performance to variations in burner zone stoichiometry, combustion heat release distribution, and ash accumulations on the furnace walls are presented.

INTRODUCTION

Radiation is the dominant mode of heat transfer in coal-fired combustion chambers. Understanding this process in pulverized coal-fired industrial and utility boilers is important both in furnace design and in evaluating the impact of operational changes on performance.

Predicting gas temperatures at a furnace exit is the starting point for designing superheaters, reheaters, and convective heat exchange sections. Local waterwall heat transfer rates within the radiant furnace are used to evaluate the potential for tube overheating and burnout. In natural circulation boilers, the vertical heat absorption profile is important in establishing circulation rates. Knowledge of the gas temperature distribution within the furnace volume along with ash properties and combustion aerodynamics can also be used to evaluate the potential for furnace slagging and ash deposition.

Today there are additional incentives for predicting thermal performance of the radiant furnace section of boilers. Emerging combustion technology offers the potential of in-furnace control of pollutants. Advanced staged combustion techniques involving the redistribution of fuel and air within the furnace are being developed for the reduction of nitrogen oxides, NO_X. Also, limestone injection in multi-stage burners (LIMB) for the control of sulfur dioxide appears promising. Both of these technologies require a detailed knowledge of furnace temperatures and heat absorption profiles. In addition, there is considerable interest in retrofit technologies for converting gas- and oil-fired boilers to pulverized coal. These technologies include advanced slagging add-on combustors and the use of coal mixtures such as coal water slurries. All of these technologies will significantly affect combustion and flow patterns as well as temperatures within the furnace.

A number of detailed heat transfer models have been developed for research furnaces as well as tangential and wall-fired boilers.[1-5] In this paper we present a multi-zone heat transfer model for a pulverized coal-fired TURBO Furnace. As shown by the cross-section in Figure 1, the dry bottom TURBO Furnace design is characterized by upper and lower rectangular furnace chambers separated by a venturi-shaped restriction. Axial flow burners are mounted in the lower furnace on opposite downward facing arches. This unique burner/furnace combination produces long diffusion controlled flames, creating both forward and recirculating gas flow within the lower furnace.

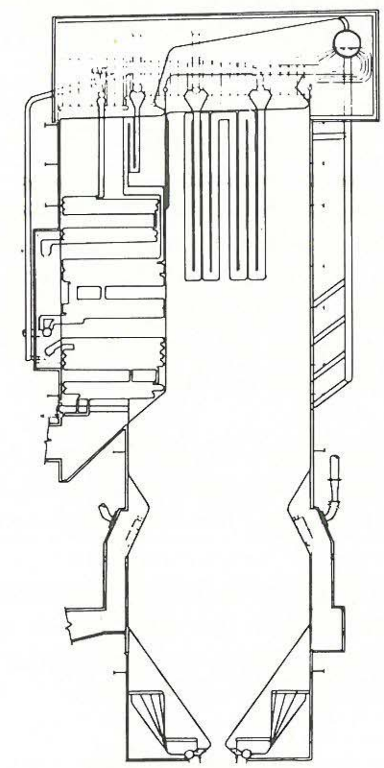

Figure 1 Riley Stoker Dry Bottom TURBO Furnace

Rather than develop a completely rigorous model of all the physical and chemical processes in the system, we have taken a phenomenological or engineering approach. Our model focuses on the heat transfer aspects of the combustion system. The model is based on a mathematical simulation with input from both laboratory experiments and full-scale field measurements. This partial modeling approach has both advantages and disadvantages.[6] It offers perhaps the quickest and most realistic method of predicting the performance of utility scale coal-fired furnaces. However, it also requires knowledge of the flow patterns and heat release profile within the furnace volume.

This paper provides a description of the model, and background on its development. The emphasis is on comparing model results with measurements from operating utility furnaces. Predictions of furnace gas temperature and wall heat flux are presented for various furnace design cases. The sensitivity of these predictions to different operating conditions is also discussed.

MODEL DESCRIPTION

The current TURBO Furnace model is a two-dimensional, multi-zone heat transfer model. This model was developed by combining the zone method of analysis with Monte Carlo techniques. In the model, the furnace is divided vertically into a series of horizontal slices. Each slice is further divided into two concentric rectangular zones. The result is the two-dimensional zone system shown in Figure 2. Gas temperature or wall temperature, and the wall heat absorption rate are predicted by the mathematical model for each zone.

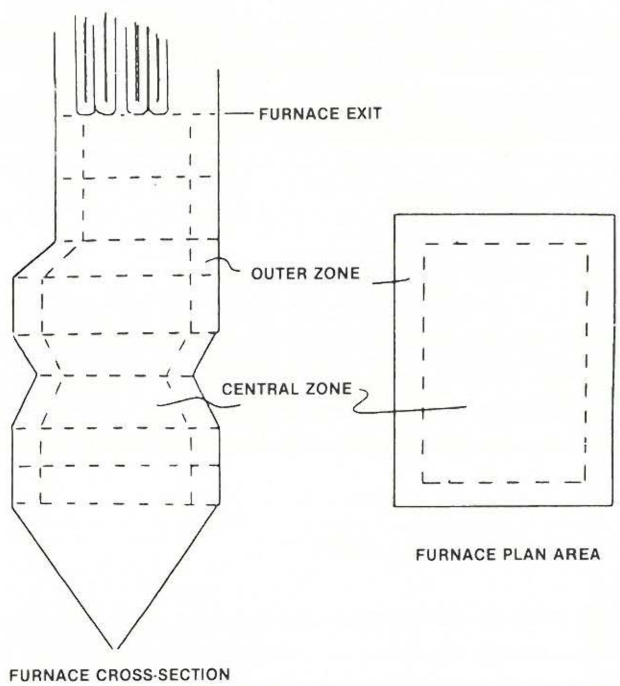

Figure 2   Typical Two-dimensional Furnace Zone Model

The appliation of the zone method requires that gas temperatures, gas composition, and radiative properties be uniform in each zone defined in the system. This requirement led to the development of a two-dimensional furnace model, which represents the combustion chamber as a hot flame core surrounded by cooler absorbing gases near the furnace walls. This type of two-dimensional model has been applied successfully to tangentially-fired boilers by Johnson.[5] We believe that this two-dimensional zone system represents a minimum level of complexity for applying zone methods to utility scale furnaces.

In the furnace zone methods developed by Hottel and Cohen[7] and Hottel and Sarofim[8], direct radiation exchange areas between zones are obtained for various geometrical arrangements from tabular and graphic data, or by solving closed integrals. Becker[9] has formulated an exchange area calculation technique for an enclosure zoned as two concentric rectangles. In all of these methods, total exchange areas, which account for multiple reflections from grey walls, require a matrix solution.[8] We have chosen to calculate both the direct and total radiative heat exchange areas by a reducing energy beam (REB) Monte Carlo technique.[10]

Monte Carlo methods treat radiative exchange between gases or surfaces in a probabilistic manner.[11] Once a furnace zone system is established, the total energy emitted by each zone is divided into a finite number of energy beams. For example, for gas volume i, the total emitted energy is defined as $4\ K_iV_i\sigma T_i^4$. Assuming the total energy can be distributed equally among N energy beams, and also normalizing with respect to σT_i^4, the initial energy of each beam can be expressed as $4\ K_iV_i/N$.

Each of the N energy beams is emitted in a random direction from a random location in the zone. The three location coordinates are selected from their cumulative distribution functions. For example, the x coordinate distribution function is given by:

$$R_x = \frac{x_o - x_{min}}{x_{max} - x_{min}} \tag{1}$$

where x_o is the emission coordinate and R_x is a random number between 0 and 1.

Emission directions are chosen by selecting a cone angle, η, and a circumferential angle, Θ. The distribution function for emissions from a volume for each of these angles can be derived[12] as follows:

$$R_\eta = (1-\cos\eta)/2 \tag{2}$$

and

$$R_\Theta = \Theta/2\pi \tag{3}$$

where R_η and R_Θ are random variables in the interval of (0, 1).

As each beam is emitted, its energy dissipation as it travels through the furnace is described. In the reducing energy beam Monte Carlo method, Beer's Law is used to define the attenuation of a beam traveling through a gas volume. For example, consider an energy beam emitted by gas volume i. Let P_i^o represent the beam's incoming energy as it enters any furnace zone. As the beam travels a distance L_j in a gas volume V_j, its energy is attenuated according to:

$$P_i^j = P_i^o\ (1-e^{k_j l_j}) \tag{4}$$

A similar process is followed when an energy beam reaches a furnace wall. If surface m has an absorptivity α_m, then the fraction of incident energy absorbed by the surface is $\alpha_m P_i^o$. The remaining energy is reflected back into the surrounding furnace volume along a random direction.

An example path of an energy beam traveling through the furnace is shown in Figure 3.

Each energy beam is followed until the energy of the beam entering some Zone b is less than or equal to 1% of its initial energy. When this occurs, Zone b absorbs all of the energy remaining in the beam, and the emitting zone emits another beam at random until all beams have been emitted. As multiple energy beams are tracked, a matrix whose terms represent the total energy each zone absorbs from every other zone in the system is generated from the summations $G_iG_j = \Sigma P_i^j$ and $G_iS_m = \Sigma\alpha_m P_i^o$. This matrix of radiation exchange areas is the final product of the Monte Carlo subroutines.

The reducing energy beam technique was used in the TURBO Furnace model for several reasons. The venturi shape of the TURBO Furnace requires modeling zones with irregular geometries. Because energy absorption by the furnace gases is calculated continuously as energy beams traverse the furnace, radiative exchange areas can be easily calculated for arbitrary zone shapes and for a medium with nonuniform characteristics. Increased accuracy in the radiative exchange calculations is achieved easily by increasing the number of emitted energy beams.

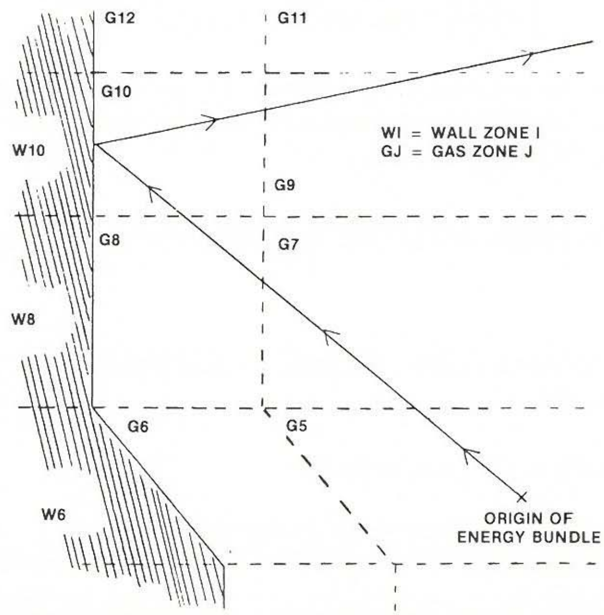

Figure 3 Example Energy Beam Path Using the Reducing Energy Beam Monte Carlo Model

Our convergence criteria for the Monte Carlo subroutines is calculating less than 5% change in the radiative exchange areas for increasing values of N. Running our model for a variety of both rectangular and TURBO Furnace geometries showed that this convergence criteria is satisfied when an average of one energy beam/m^3 of furnace volume is emitted. If the convergence criteria is relaxed to less than a 7% change in the calculated exchange areas, the emitted energy beam density can be reduced to 0.25 beams/m^3.

RADIATIVE PROPERTIES

Before the energy beam tracking calculations can proceed, the radiative properties of each zone in the furnace must be specified. The TURBO Furnace model assumes that the furnace gases are gray emitters. The computer code does not account for scattering within gas volumes or the spectral nature of emitted radiation. Rather, the model assumes that a single value for the gas absorption coefficient characterizes the combustion products in each zone.

Radiative transfer in pulverized coal flames is complicated by the presence of particulate matter. Soot, char, and ash particles, as well as nonluminous radiating gases, are important contributors to total flame emissivity. The emissivity model developed by Wall[13] for these particles is used in the TURBO Furnace model. Since the radiative properties of these pulverized coal flame emitters are based on their particle size distribution, the absorption characteristics are also a function of the local unburnt fuel fraction. If the particulate emission is considered gray, the composite absorption coefficient of the flame is simply:

$$K_f = \Sigma K_m \tag{5}$$

where K_m represents the absorption coefficient for each component, i.e. char, ash, soot, and triatomic gases (water vapor and carbon dioxide).

The furnace walls are modeled as gray Lambert surfaces. The model accounts for multiple wall reflections and for both clean and ash-covered walls. The influence of deposits on the furnace walls is accounted for in two ways: the wall absorptivity can be varied in each zone, and an ash layer of varying thickness can be added to any wall zone.

ENERGY BALANCE CALCULATIONS

Once the radiative exchange areas have been determined, energy balance equations must be solved for each zone in the furnace to predict local furnace temperatures. The heat transfer rate from a gas zone to all other gas and surface zones in the furance is given by:

$$Q_{net_i} = \sum_j \overline{G_iG_j}\ (T_i^4 - T_j^4) + \sum_k \overline{G_iS_k}\ (T_i^4 - T_k^4) \tag{6}$$

For all gas zones adjacent to furnace walls, the convective heat transfer term $A_kH_k\ (T_i - T_k)$ is also added to the right hand side of Eq. (6).

The energy input in each gas zone is also given by:

$$Q_{net_i} = m_i\ c_{pi}\ (T_m - T_{ref}) + Q_{chem_i}, \tag{7}$$

where T_m is the temperature of all gases entering zone i, and T_{ref} is some reference temperature.

At this point, the Monte Carlo subroutines have already calculated the radiative exchange area terms $\overline{G_iG_j}$ and $\overline{G_iS_k}$. Input information characterizing the furnace recirculation and heat release distribution patterns is required to complete the energy balance calculations.

A clipped Gaussian distribution function is used to represent the heat release due to combustion in the furnace. This distribution function is used in place of a detailed char combustion kinetics model. The clipped Gaussian model was selected since it provides for varying the shape of the heat release profile as well as the combustion zone size and location in the furnace. It also allows burnout to decay in both upward and downward furnace directions, which is characteristic of this type of arch firing.

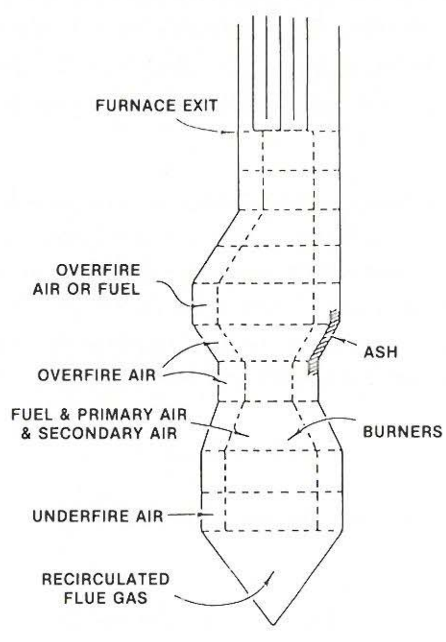

Figure 4 Fuel and Gas Streams Represented in the Furnace Model

The fuel and gas streams included in the flow recirculation model are shown in Figure 4. We assume that the coal, primary air, and secondary air are perfectly mixed as they enter the burner zones, and that these three streams follow the same recirculation pattern. The model also accounts for the effects of adding overfire air, underfire air, or recirculated flue gas. These gas streams are presumed to enter the furnace in the locations shown in Figure 4.

The degree of gas recirculation below the burners is an input to the computer code. Experience from cold flow modeling studies[14], [15] is used to specify this input data.

In the burner zones and zones below the burners, the gases are assumed to be perfectly mixed. Above the burner zones, the correction for the departure from perfectly stirred gases suggested by Hottel and Sarofim[16] is used, so that the radiating gas temperature in each zone is a weighted average of the gas exit temperature and the adiabatic flame temperature of the zone.

Once the gas recirculation and combustion heat release patterns are specified, a surface temperature for each wall zone in the furnace is assumed. Equations (6) and (7) can now be combined to write an energy balance equation of the form:

$$aT_i^4 + bT_i = c \tag{8}$$

where a and b depend weakly on T_i, and c depends on all the other gas and wall zone temperatures in the system. In order to solve Equation (8) for all gas zones i in the system, an initial guess is made for all gas and wall surface temperatures. Newton-Raphson and Gauss-Seidel techniques[17] are then used to iteratively solve for the gas temperatures T_i of each zone. Once all gas temperatures are known, the wall heat flux can be calculated.

Another iteration is then performed to update the assumed wall surface temperatures. Surface temperatures are now computed according to Equation (9) based upon the calculated wall heat absorption, the thickness and thermal conductivity of the ash layer on the wall, and the temperature of the fluid in the boiler tubes.

$$\frac{Q}{A} = \frac{T_{wall} - T_{sat}}{\dfrac{1}{H_B} + R_F + \dfrac{x_{wall}}{K_{wall}} + \dfrac{x_{ash}}{K_{ash}}} \tag{9}$$

If the differences between the computed and assumed wall temperatures exceeds an error tolerance, the iteration on gas temperature is repeated, using the revised wall surface temperatures. Convergence on both gas and wall zone temperatures is usually achieved in three to five iterations on surface temperature.

CASE STUDIES OF UTILITY BOILERS

Before embarking on a series of parametric studies using the computer code, this model was verified using operating data from two utility boilers. The furnace geometry of the boilers is shown in Figure 5, while Table I presents their full load operating conditions.

The Case I unit was used to verify the computer model since it had been instrumented extensively as part of a special boiler performance evaluation program. The Case II unit was selected to test the computer code on an alternate TURBO Furnace geometry. The Case I unit is a 490 MWe TURBO Furnace firing midwestern bituminous coal. Test data is available for this unit at several loads, with the boiler operating unstaged in all cases. This data includes local measurements of wall heat flux in the radiant furnace and of gas temperatures at the entrance to the superheater cavity.

All furnace gas temperatures were measured using suction pyrometers. A temperature traverse from one to eighteen feet of the furnace wall was performed at the radiant furnace exit.

Wall heat flux measurements were made using Northover-Hitchcock style heat flux meters[18] supplied by the Central Electric Generating Board of Surrey, England. These heat flux meters employ the conducting disc principle to measure heat absorption. The meters are 18 mm diameter, 8 mm thick discs that are welded onto the boiler waterwall tubes. Each meter was calibrated before being installed in the furnace. Readings from all meters were recorded by an automatic data logger. Eight readings from each meter were recorded during a half-hour period and then averaged to obtain the final heat flux measurement at each location.

The average predicted vs. measured wall heat flux rates for the test boiler agreed within ± 10%. Two data sets of the measured vs. predicted wall heat absorption are presented in Figure 6. The measured data points plotted in this figure are averages of the reading of the twenty heat flux meters installed at the indicated elevations, since the furnace model considers the furnace enclosure at each elevation as one wall surface.

The gas temperature traverse was used to specify the width of the outer gas zone used in the model, as well as to check the predicted exit gas temperatures. We found that the model's heat flux and gas temperature predictions correlated with the field measurements when the width of the outer gas zone was 12% of the furnace depth. Predicted temperature profiles are presented in Figure 7, together with the average measured exit temperatures.

	CASE I	CASE II
MWe	490	270
FURNACE VOLUME (m^3)	10,374	4,000
FURNACE HEIGHT (m)	33.2	29.5
FUEL	BIT. COAL	SUB-BIT COAL
HIGHER HEATING VALUE (KJ/kg)	24,411	19,390
TOTAL AIR INPUT (kg/sec)	610	296
TOTAL FUEL INPUT (kg/sec)	57	39

Table I Full Load Design and Operating Data on Model Study Boilers

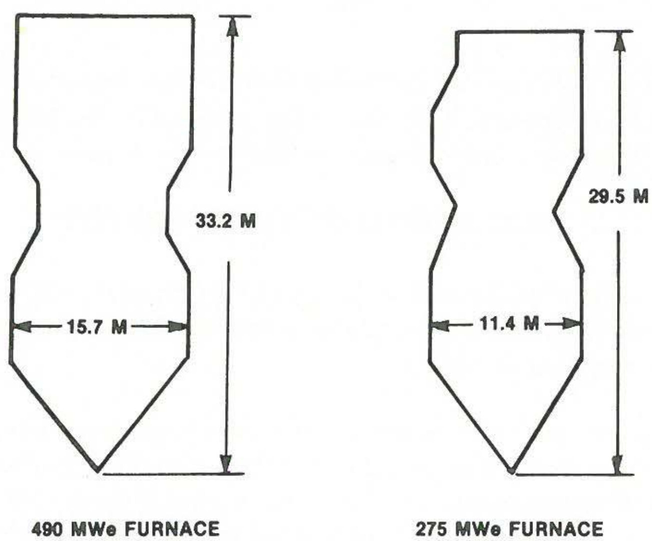

Figure 5 Furnace Geometries Modeled in the Field Data Comparison Studies

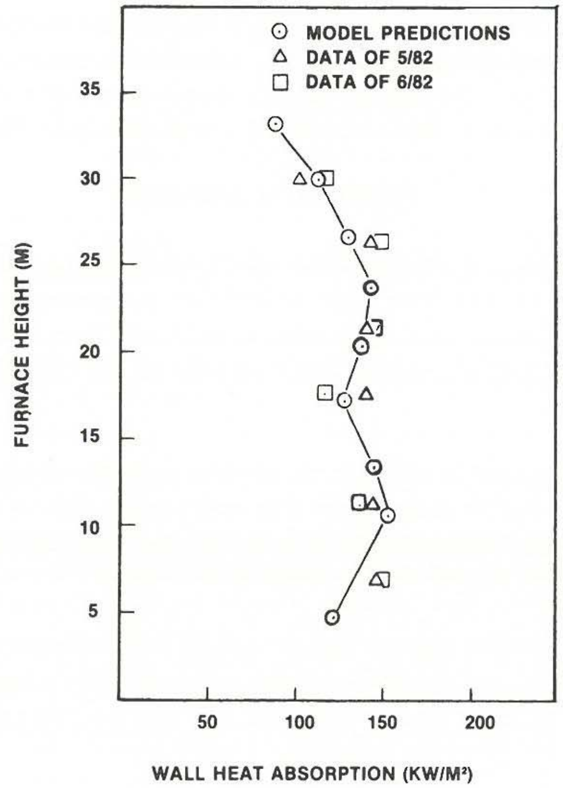

Figure 6 Model Predictions vs. Field Data, 490 MWe Furnace Study (80% load firing bituminous coal)

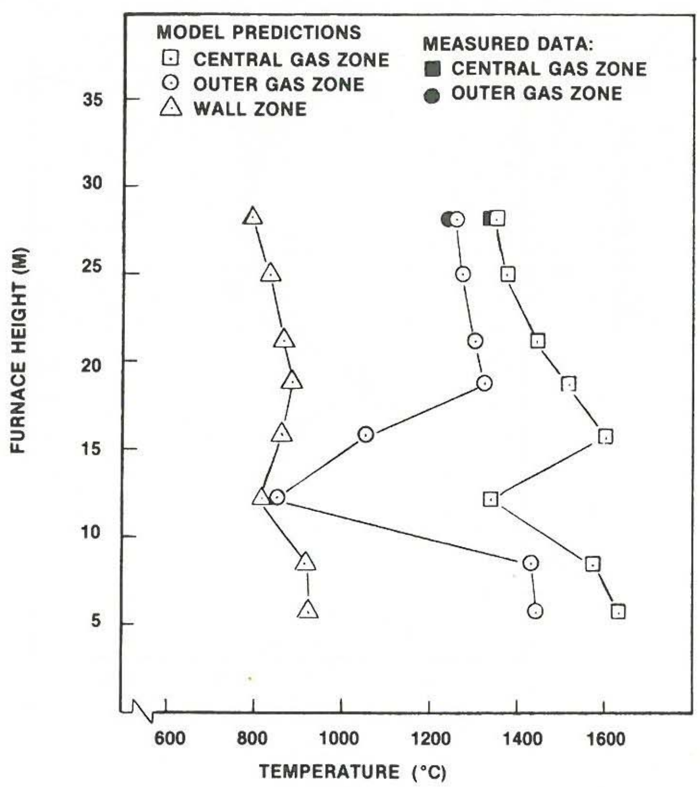

Figure 7 Predicted Furnace Temperature Profiles, 490 MWe Furnace Study (60% load firing bituminous coal)

The Case II unit is a 270 MWe utility boiler fired by a western sub-bituminous coal. Although this unit was not instrumented as extensively as the previous boiler, wall heat flux measurements were made using chordal thermocouples at 30 locations near the furnace venturi. Comparisons of the model predicitons with these measurements are presented in Figure 8. Again, the model's predictions agree with the field data within ± 10%.

SENSITIVITY STUDIES

A major objective for developing the furnace heat transfer model was to assess the impact of design modifications on furnace performance. Therefore, once the wall heat absorption simulation was confirmed, a series of parametric studies was performed. In these studies, variations in burner zone stoichiometry, combustion heat release distribution, flow pattern, and ash accumulations on the furnace walls were modeled to study their impact on performance.

Since the furnace recirculation and heat release distribution patterns are inputs to the computer model, we performed several studies to assess the sensitivity of the model's predictions to these parameters. The chemical heat release profile input stongly influenced the program results, while the model was relatively insensitive to the flow pattern used. Figure 9 presents the results of this study.

Another important parameter affecting the furnace wall heat absorption is whether the furnace walls are clean or covered with an ash layer. Ash influences furnace performance by increasing both the wall temperatures and the wall absorptivity. The net effect is a decrease in the local wall heat absorption rate, as shown in Figure 10.

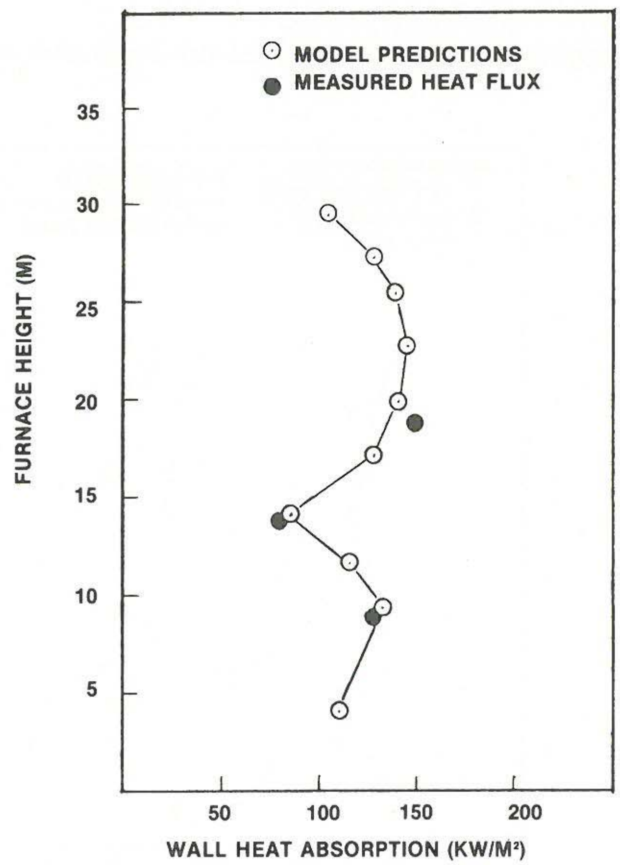

Figure 8 Model Predictions vs. Field Data, 275 MWe Furnace Study (100% load firing sub-bituminous coal)

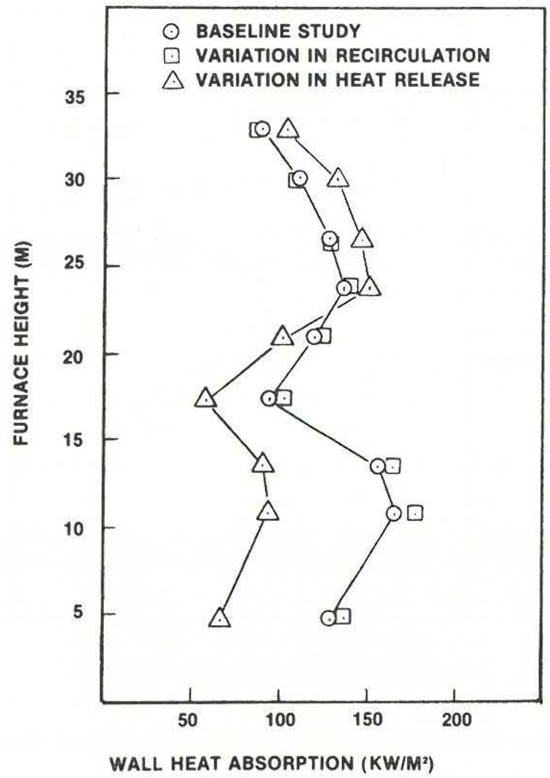

Figure 9 Influence of Recirculation and Heat Release Distributin on Wall Heat Absorption

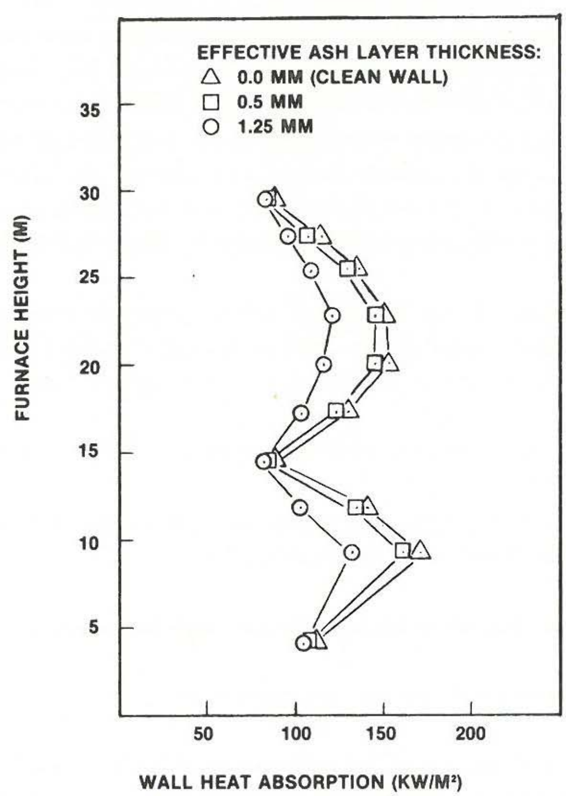

Figure 10 Influence of Ash Deposits on Wall Heat Absorption

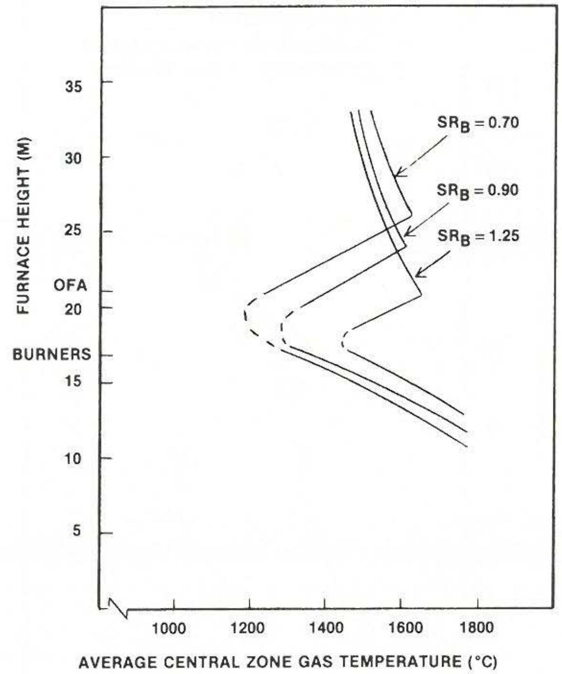

Figure 11 Turbo Furnace Temperature Profiles, Staged Combustion Study

STAGED COMBUSTION STUDIES

Staged combustion, achieved by introducing a portion of the secondary combustion air downstream of the burners, is a promising NO_X control technology. Staging lowers NO_X emissions by modifying the mixing, temperature, and heat release distribution in the furnace. In general, NO_X formation is promoted by high burning zone heat release rates, and also by rapid mixing and high oxygen concentrations during the early stages of combustion. In advanced or deeply staged systems, as much as one-half the secondary combustion air flow may be introduced as staged air, thereby creating two distinct combustion zones. Combustion in the first stage therefore occurs in a fuel-rich atmosphere, and lower furnace temperatures are produced through the combined effects of delayed combustion and quenching by the overfire air jets.

The model has been used to simulate the thermal effects of advanced staging in TURBO Furnaces. Since no units are operating in this mode, results from pilot scale staged combustion studies by Beer, et al[19] and Yang, et al[20] were used to check the trends predicted by the computer model.

These references predict that as the burner zone stoichiometry, SR_B, decreases:

(1) Heat release in the first stage combustion zone decreases, with corresponding decreases in gas temperatures and wall heat absorption rates.

(2) Heat release in the second stage combustion zone increases.

(3) There is a small change in the exit gas temperature.

Reference 20 also includes a correlation of the percentage of heat released in the first stage combustion zone with the first stage stoichiometry. When this correlation is applied to the TURBO Furnace, the gas temperature profiles shown in Figure 11 are predicted by the computer model. These profiles are consistent with the results of the MIT studies.[19]

CONCLUSIONS

A flexible model of a complex furnace geometry has been developed by combining zone and Monte Carlo methods. Comparisons between the model's predictions and measurements from utility boilers have demonstrated that the model successfully simulates radiant furnace performance. We have also shown that knowledge of the combustion heat release distribution and of local furnace wall conditions are important parameters in predicting local operating conditions.

ACKNOWLEDGEMENTS

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NOMENCLATURE

A	Wall surface area
C_p	Specific heat at constant pressure
G_iG_j	Gas-to-gas radiative exchange area
G_iS_k	Gas-to-surface radiative exchange area
h	Convective heat transfer coefficient
H_B	Boiling heat transfer coefficient
K	Gas absorption coefficient
L	Length
m	Gas product mass flow
N	Number of energy bundles emitted in a zone
P	Probability function describing a beam's energy in a zone
Q	Heat release rate
R	Random number in the interval between (0,1)
R_f	Steam side fouling resistance
T	Temperature
V	Zone volume
x	Coordinate
α	Wall absorption coefficient
η	Cone angle
σ	Stephan-Boltzman constant
Θ	Circumferential angle

SUBSCRIPTS

chem	chemical
f	flame
1,j,k,m	zones
max	maximum
min	minimum
η	cone angle
Θ	circumferential angle

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