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Scale-Up and Advanced Performance Analysis of Boiler Combustion Chambers

WOLFGANG RICHTER
Senior Research Scientist
Energy & Environmental Research Corporation (EER)
Irvine, California

ABSTRACT

This paper discusses methods for evaluation of thermal performance of large boiler furnaces. Merits and limitations of pilot-scale testing and mathematical modeling are pointed out. Available computer models for furnace performance predictions are reviewed according to their classification into finite-difference methods and zone methods. Current state of the art models for industrial application are predominantly zone methods based on advanced Monte-Carlo type techniques for calculation of radiation heat transfer. A representation of this model type is described in more detail together with examples of its practical application. It is also shown, how pilot-scale results can be scaled-up with help of the model to predict full-scale performance of particular boiler furnaces.

NOMENCLATURE

- A surface area, m^2
- A frequency factor, kg/m^2s bar
- C_p specific heat capacity, kJ/kg K
- E activation energy, kJ/kmole
- f factor in Eq. 7
- K_a absorption coefficient, 1/m
- k thermal conductivity, kW/mK
- L furnace dimension, m
- $\dot{M}_o$ mass flux, kg/s
- $\dot{Q}$ ratio of actual release of volatiles and release of volatiles according to prox. analysis
- $\dot{Q}_o$ thermal input, kW
- $\dot{q}_s$ heat release per heat sink surface area, kW/m^2
- $\dot{q}_v$ volumetric heat release, kW/m^3
- s path length, m
- s thickness of ash deposits, m
- T temperature, K
- T_o reference temperature, K
- V volume, m^3
- α absorptivity
- α convective heat transfer coefficient, kW/m^2K
- Δ finite increment
- η_f furnace efficiency
- ε emissivity
- Ω solid angle, sr
- ρ density, kg/m^3
- σ Stefan-Boltzmann constant, kW/m^2T^4
- τ_{res} residence time, s

SUBSCRIPTS

- a absorption
- ad adiabatic
- eff effective
- ex exit
- f full-scale
- I,J,K zone indices
- o initial
- p pilot scale
- ref reference
- s heat sink surface
- t total
- v volume

INTRODUCTION

Despite the considerable progress in the development of analytical methods in many fields of engineering science and despite an increasing understanding of fundamental combustion processes the design or performance predictions of existing boiler furnaces may still be considered as an art based primarily on the empirical knowledge and ingenuity of the combustion engineer. Two facts are especially responsible for this phenomenon:

- the extremely complicated boundary conditions encountered in almost all practical furnaces.
- a lack of confidence into existing analytical methods.

The former fact cannot be changed and it is up to those who develop the analytical methods to generalize

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their models so they can more easily be applied to practical systems. The latter fact originates from historical reasons. Fifteen years ago, when the first comprehensive furnace models were published (1) the expectations from the new analytical tools were high. The models which were at that time two-dimensional were soon used by several researchers for axisymmetric flow and flame predictions (2-4). Good agreement was found by comparing the predictions with experimental data in pilot scale furnaces, especially those operated by the IFRF (5-6). However, this was achieved in many cases, by adjusting a number of model parameters, especially those of the turbulence and reaction models. Encouraged by the early good results, combustion engineers were interested in practical application but they soon became somewhat disappointed because these models, in spite of their complexity, sometimes failed to predict simple furnace performance features within a reasonable range of accuracy (7). The restriction to simple furnace geometries and often encountered stability problems had also an adverse effect on a wide spread of the models into the combustion practice. And finally, practical combustion engineers not so familiar with the complex details of the new methods, required answers beyond the limit of applicability of the models which were still under development. Unfortunately, the initial distrust of practical furnace designers into analytical methods, in general, can still be observed today despite major improvement in many of the methods (8, 9).

On the other hand, design requirements necessary for trouble-free operation of boiler furnaces are increasing for several reasons:

- Necessity of improvements of combustion efficiency
- Deterioration of fuel quality
- Fuel conversion
- Restrictive regulations for emission of pollutants
- Larger units
- Operation with flexible load

Demands for successful design of coal fired furnaces are especially challenging. Errors in furnace design are costly and fuel conversions can have a strong impact on furnace performance, emission of pollutants and the cost of energy. Therefore, the need for reliable analytical methods which allow performance predictions for a wide range of operating conditions and fuel types is more urgent than ever.

The purpose of this paper is to demonstrate that advanced analytical methods developed for practical application can indeed successfully support the solution to a variety of furnace operating problems. These methods bridge the gap between comprehensive combustor models developed by the scientific community and current design practice based on empirical knowledge. They are especially suitable to scale-up results obtained in pilot-scale furnaces to predict performance of particular full-scale boiler furnaces.

The paper is divided into five sections. In a first section, the major factors influencing boiler furnace performance are analyzed. In the second section, merits and limitations of experimental pilot-scale studies and of mathematical models for full scale performance predictions will be discussed. This section also reviews available models with respect to practical applicability. A particular "applicable" furnace model developed by Energy and Environmental Research Corporation (EER) over the last five years and representing the current state of art of practical furnace modeling will be described in more detail. In the third section, this model will be applied to analyze experimental results obtained in a pilot scale furnace. For a coal water slurry it will be shown, how the pilot furnace results were scaled-up with help of the model to predict the performance of the same fuel in a large scale boiler furnace. The fourth section summarizes a variety of successful model applications to various coal fired boiler combustion chambers. The fifth section, finally, provides an outlook on a model of formation of ash deposits in coal fired boilers.

GENERAL ASPECT OF BOILER FURNACE PERFORMANCE

Major dependent variables which the combustion engineer would like to obtain from a quantitative furnace analysis are listed in Figure 1. Three groups of these variables are distinguished which refer to the furnace exit or beyond, to the furnace heat sink surfaces and to the furnace volume, respectively. The quantities of interest at or beyond the furnace exit are: First and most important furnace efficiency η_f and furnace exit temperature T_{ex} which are related to each other by Eq. (1), heat flux from the furnace to the radiant superheaters to assess metal temperatures, the amount of fouling at the radiant heat exchangers, the amount of unburnt combustibles and finally emission of gaseous and solid pollutants.

$$\eta_f = 1 - \frac{\dot{M}_o c_p \big|_{T_o}^{T_{ex}} (T_{ex} - T_o)}{\dot{Q}_o} \qquad (1)$$

Quantities of interest at furnace walls are: heat flux distribution, peak heat fluxes and amount of slagging and corrosion. The interesting quantities in the furnace volume itself are local temperatures and concentrations of particles and gaseous chemical species, and more general flame extension and flame stability especially at lower loads and finally the formation of pollutants in the flame itself. Independent variables of a performance analysis listed on the left hand side of Figure 1 are furnace geometry, fuel type and quality, load and boundary conditions like for instance the amount of refractory coverage.

Factor Influencing Furnace Performance

The key process influencing furnace performance is heat transfer which affects almost every other physical or chemical process occurring in the combustion chamber. Due to the high temperature levels in fossil-fuel-fired combustion chambers the dominant mode of heat transfer to heat sinks is by thermal radiation. This is especially true when pulverized fuel is fired since here the emissivity of the furnace volume is considerably increased by the presence of solids such as coal, char fly ash, and soot particles. Therefore, the special characteristics of radiative heat transfer have to be carefully taken into account in any furnace analysis. Major factors contributing to heat transfer in boiler furnaces are summarized in Figure 2. These are: the adiabatic flame temperature T_{ad}, the firing density $\dot{q}_s$ the total emissivity ε_t, the temperatures T_s of the heat sink surfaces exposed to the furnace gases and the flow, mixing and heat release pattern. The adiabatic flame temperature depends on the fuel type i.e. on the heating value and moisture and ash content, on the amount of excess air and on fuel and air preheat. The firing density is the ratio of thermal load and heat sink area defined by Eq. (2).

$$\dot{q}_s = \frac{\dot{Q}_o}{A_s} \tag{2}$$

The total emissivity of the furnace is related to the emissivity ε_v of the combustion products and the emissivity ε_s and absorptivity α_s of the surfaces. The latter properties, namely ε_s and α_s as well as the surface temperatures depend on the characteristics of ash deposits which, consequently, have a strong impact on the performance of p.f. fired furnaces. The flow, mixing and heat release patterns are dictated by many factors, most important are geometrical configuration of furnace and burners and burner operating conditions. In case of pulverized fuel firing the heat release pattern is particularly influenced by fuel-burn-out characteristics.

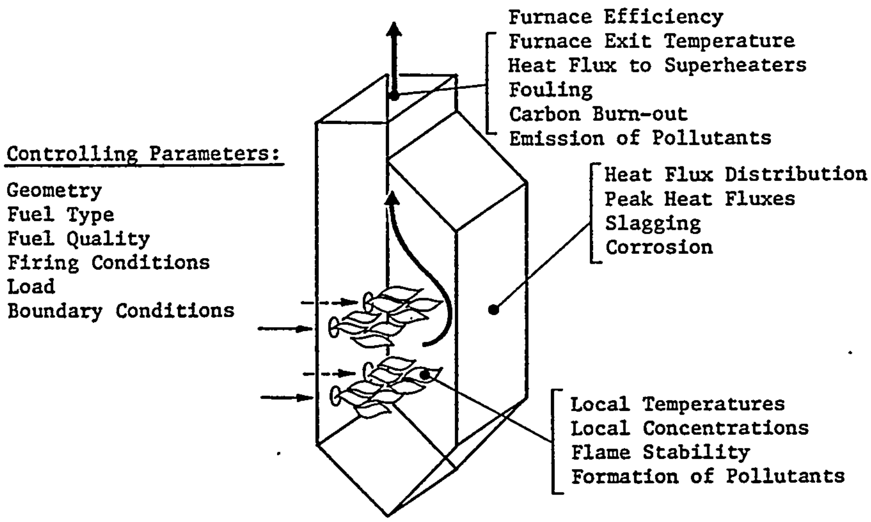


Fig. 1 Performance analysis of boiler furnaces.

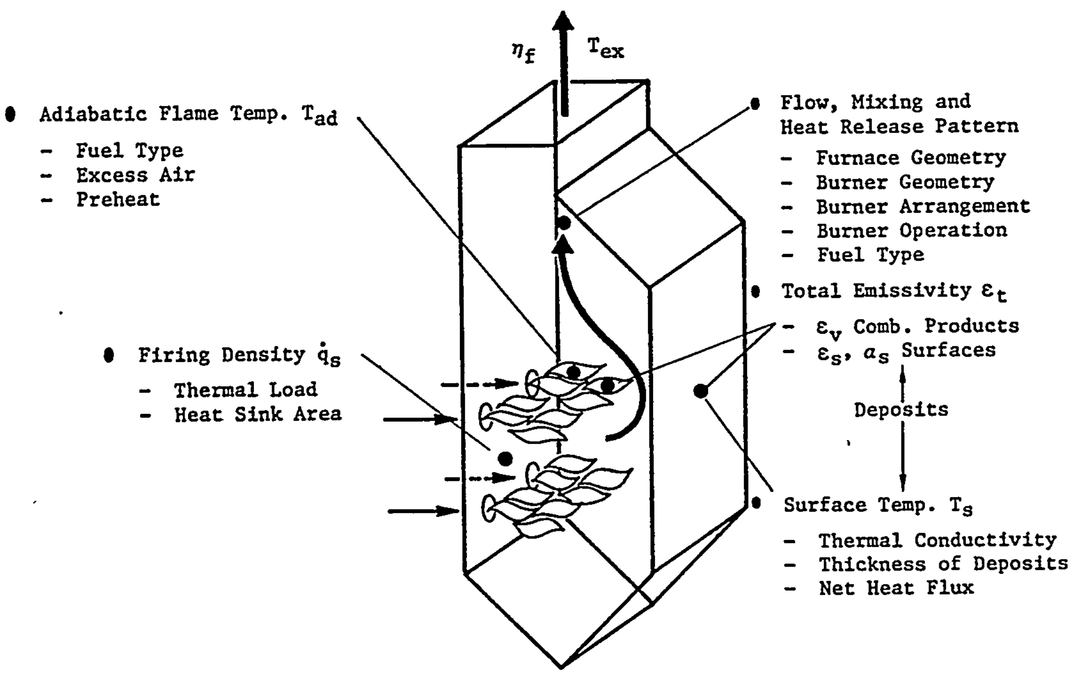


Fig. 2. Factors influencing thermal performance of boiler furnaces.

METHODS FOR FULL SCALE BOILER PERFORMANCE PREDICTIONS

A quantitative prediction of large-scale boiler furnace performance for a given fuel can currently be carried out in three different ways, namely

- Empirically by extrapolation from known performance of furnaces of similar design fired by the same or a similar fuel
- Experimentally, by scaling up results from pilot scale trials using the same fuel
- Analytically, by applying mathematical models directly to operating conditions of the large scale furnace.

The first method is still common practice in boiler industry. The method often involves a trial and error process and occasionally leads to design errors which are especially expensive for larger units. Merits and limitations of the two other methods are discussed in the following.

Pilot Scale Experiments for Full Scale Performance Predictions

Due to the occasional failure of the empirical methods and due to lack of confidence in mathematical models for reasons mentioned already before, experimental pilot scale trials play an important role in evaluating fuel performance for particular boilers. The ideal pilot scale trial would be one in which all important physical and chemical processes are similar to those occurring in the full-scale boiler combustion chamber. However, there are numerous physical and technical reasons why a complete similarity can never be achieved (10). Therefore, besides the use of same stoichiometries and the attempt to obtain a similar mixing pattern, pilot scale testing for furnace performance is usually carried out with the minimum requirement that the time-temperature histories in the pilot-scale furnace and the full scale boiler furnace are approximately the same (11). This allows then conclusions about thermal performance, fuel burn-out characteristics, slagging and fouling potential, effectiveness of measures of emissions control like dry sorbent injection, etc. in the full-scale furnace. However, even the achievement of equal time-temperature histories in pilot full scale furnace is not always guaranteed as shown in the following by use of results of a simple well-stirred analysis (12). In order to obtain the same residence times

$$\tau_{res,p} = \frac{\rho_p V_p}{\dot{M}_{o,f}} = \frac{\rho_f V_f}{\dot{M}_{o,f}} = \tau_{res,f} \qquad (3)$$

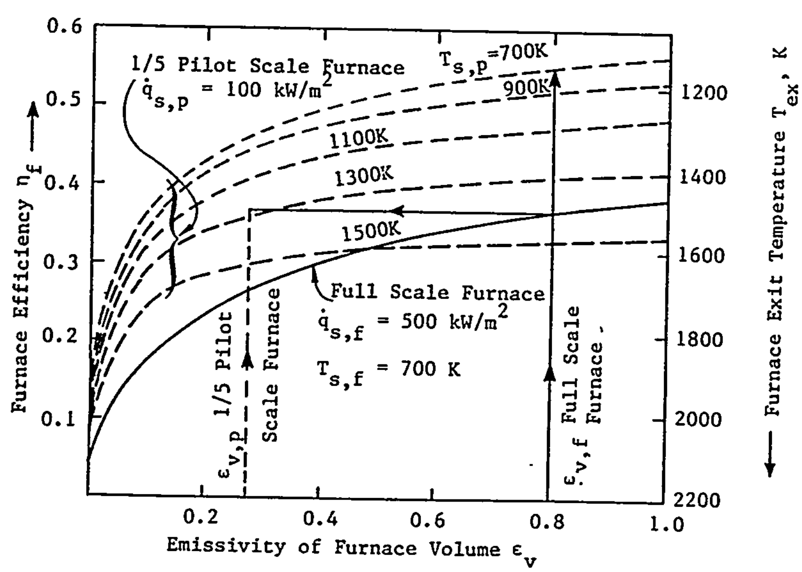
the volumetric firing densities $\dot{q}_v$ defined in Eq. (4) and mean furnace temperatures $\overline{T}$ (= T_{ex} due to well-stirred assumption) of pilot and boiler furnace must be the same:

$$\dot{q}_{v,p} = \frac{\dot{Q}_{o,p}}{V_p} = \frac{\dot{Q}_{o,f}}{V_f} = \dot{q}_{v,f} \qquad (4)$$

$$\overline{T}_p = \overline{T}_f \qquad (5)$$

However, the well-stirred analysis (Figure 3) shows that $\overline{T}_p = \overline{T}_f$ can usually not be obtained for same surface temperatures of the heat sinks in pilot and full scale furnace because the thermal performance of a furnace is, instead by $\dot{q}_v$ rather characterized by $\dot{q}_s$, the firing rate per heat sink surface area. For similar shapes and equal volumetric firing densities, $\dot{q}_{s,p}$ of the pilot scale furnace is related to $\dot{q}_{s,f}$ of the full scale furnace by Eq. (6)

$$\dot{q}_{s,p} = \frac{L_p}{L_f}\dot{q}_{s,f} \qquad (6)$$



T_{ad}=2200K; ε_s=0.7; α=0.0116 kW/m²K

Fig. 3 Scaling of thermal performance of furnaces.

in which L_p and L_f are characteristic furnace dimensions (e.g. width) of pilot and full-scale furnace. Figure 3 shows as an example that temperatures in a 1/5 scale pilot furnace would decrease by 170K assuming a surface firing density of $\dot{q}_{s,f}$ = 500 kW/m^2 in the full-scale furnace, equal heat sink surface temperatures and equal emissivities ε_v = 0.8 of the furnace volumes. The latter assumption is of course not true, since ε_v is a function of the scale itself according to Eq. (7)

$$\varepsilon_{v,p} = 1 - (1-\varepsilon_{v,f})^{f \frac{L_p}{L_f} \frac{K_{a,p}}{K_{a,f}}} \qquad (7)$$

in which $K_{a,p}$ and $K_{a,f}$ are absorption coefficients in pilot and full scale furnace, respectively. The factor f in Eq. (7) depends slightly on the ratio of optical thickness in pilot and full scale furnace and has usually a value somewhat larger than one. Simplifying $f \approx 1$ and assuming gray radiation i.e. $K_{a,p} = K_{a,f}$, the example for $\dot{q}_{s,f}$ = 500 kW/m^2 and $\varepsilon_{v,f}$ = 0.8 shown in Fig. 3 yields according to Eq. (7) an emissivity $\varepsilon_{v,f}$ of the 1/5 pilot scale furnace of 0.275. Therefore, in order to obtain the same temperature level, the surface temperatures of the heat sinks of the pilot scale furnace have to be increased from 700K to 1270K, which can be achieved by insulating the walls of the pilot-scale furnace with refractory material.

The foregoing analysis was carried out assuming idealized conditions. Under practical circumstances it is almost impossible to achieve exactly the same time-temperature histories in pilot and full scale furnace. Major reasons for this fact are:

- No geometrical similarity
- Differences in local flow, mixing and heat release pattern
- Inhomogeneous distribution of wall deposits in full-scale furnace
- Different ignition characteristics in pilot-scale furnace due to radiation from high temperature surfaces
- Different dependency of furnace temperature on load in refractory covered pilot-scale furnace and full-scale furnace
- Unknown radiative properties in the full scale-furnace
- Non-grayness of furnace radiation
- Impact of scattering of radiation on full scale furnace performance

The possible differences in performance and time-temperature histories due to the reasons mentioned above can cause considerable uncertainties in scaling up pilot furnace results to particular full scale furnaces. It cannot always be excluded that these uncertainties are of the same order of magnitude than the range of accuracy necessary for useful full scale performance predictions of particular furnaces. Occasionally, hot geometrically scaled physical models of particular boiler furnaces have been built (12a, 12b) and operated with some success. It was shown that velocity and temperature profiles measured in the models were in good agreement with corresponding field measurements and observations. In (12a) it was also shown, that velocity profiles obtained in the model under hot and isothermal conditions, respectively, agreed when the burner jet flows in the isothermally operated furnace model were distorted by gauzes to take into account the expansion in the flame fronts.

Hot geometrically scaled pilot models of particular furnaces are very expensive to build and to operate and still suffer from some of the uncertainties listed above. Therefore, direct mathematical modeling of the full-scale process must be considered as serious alternative to pilot scale testing. However, as it will be demonstrated later, mathematical models can also be used to support the scale-up of experimental pilot scale data.

Mathematical Furnace Models

Characteristics of Mathematical Models. A mathematical model is a set of equations which describe the physical and chemical phenomena which take place during the process. If the process involves combustion (particularly of a solid fuel) these phenomena are complex and interactive; as such, modeling requires:

- The development of a series of submodels which describe all the relevant phenomena and their interactions.
- Restriction to the process phenomena which are of interest for a given application.

Consequently, a model is not perfect. A perfect model which reflects every detail of the true physical process would no longer be a model.

Questions are often raised regarding the need for complexity in mathematical models. Complexity is quite often equated with correctness. The processes occurring in boiler combustion chambers are complicated and yet a successful model need not be complex. The most appropriate model is one which uses the simplest concepts to predict the process features of interest with the desired accuracy. It might be necessary to involve more sophisticated concepts if higher accuracy or more complicated process details must be predicted. However, these more complicated models do not contradict the simple models provided the latter are used in the limited domain for which they were developed. Consequently, the eventual purpose of a model will dictate its complexity.

As mentioned before, the key process influencing performance of boiler furnaces is heat transfer. Therefore, the main efforts in boiler furnace modeling have been directed in the past towards development of more reliable methods for furnace heat transfer predictions. A model which is capable of predicting correctly the boiler furnace performance must incorporate:

- A description of radiant and convective heat transfer between the flame, the furnace gases and the walls, including a description of the properties of the flame, combustion products, and wall deposits which affect heat transfer.
- A description of the furnace flow field.

- A description of energy release process (combustion).

Available Furnace Models

Depending on the number of fundamental physical lows considered, the degree of simplification and the chosen solution algorithm, a wide spectrum of mathematical combustor models has been developed in the past (8, 13-14). However, excluding pure empirical models, all furnace models are based, in some way, on balances of physical properties transported continuously within the combustion chamber. These properties are either conserved in the form in which they were supplied to the combustion chamber through burners and other inlets or they are converted into other forms by chemical reactions, dissipation, emission/absorption, etc., depending on spatial location and time. The balances are numerically solved for a more or less large numbers of control elements in which the furnace divided, and for a more or less number of dependent furnace variables like temperatures, concentration of chemical species, velocities, pressure etc. representative for each control element.

The fineness of subdivision into control elements is frequently used to classify available furnace models into two groups, namely:

- the zone models and
- the finite - difference models.

The zone models use with respect to the whole furnace a macroscopic subdivision whereas the finite-difference models utilize almost microscopic control volumes. Zone models are usually so-called decoupled heat transfer models whereas in finite-difference models, fluid flow, combustion and heat transfer is modeled in a coupled manner. The advantages and disadvantages of both approaches are extensively discussed in (9).

Finite-difference models allow a finer resolution of flame temperatures and other furnace variables. They also need a much smaller amount of input data than zone methods since velocity distribution in the furnace is calculated simultaneously with heat transfer. However, because of the small size of control volumes, sophisticated turbulence models are necessary to predict turbulent exchange. Finite difference furnace models which are usually based on computation techniques for turbulent flows developed by Spalding and co-workers (15-17) have been improved in many details over the last decade (8-9). The most advanced models for p.f. fired combustors are probably those developed by Smoot (18) and by Lockwood (19). Both models are two-dimensional and only applicable to cylindrical furnaces. Dynamics and thermal histories (i.e. devolatilization, char burn-out) of the particulate phase are calculated with a detailed Lagrangian approach in which reacting coal particles representing different size classes and different inlet positions are tracked through the Eulerian grid used for the gas phase calculations. Turbulent particle dispersion is only approximately taken into account.

Representative for advanced 3-D finite-difference furnace models are the models of Lockwood for a gas-fired refinery heater (20) and the model of Srivatsa for gas turbine combustors distributed by NASA (21). The 3-D model of Lockwood is interesting because it uses a combination of two computational grids one of which is curvilinear orthogonal to take the 3-D geometry of burner and combustor into account. The model of Srivatsa is remarkable because it includes a four-step hydrocarbon oxidation scheme, a two-equation model for soot formation and an elementary kinetic scheme for NO_x emissions. 3-D applications of coupled finite-difference methods to boiler furnaces require still an enormous computational effort, yet a stable converged solution of the balance equations is not always guaranteed. The first (and up to date only) operational fully coupled 3-D mathematical model of boiler combustion chambers was developed by Zuber (22). This model can be applied to coal fired boilers (23). The model was also applied to study the dry sorbent injection process for SO_x control in boiler furnaces (24). Due to the large computational effort required in his model, Zuber was forced to make a considerable number of simplifications i.e. constant viscosity hypothesis, neglection of two-phase flow effects, one overall fuel fraction, premixing of air and fuel in near burner cells etc. The major disadvantage of available finite-difference models for boiler furnace performance predictions lies in the fact that it is notably the important radiative heat transfer which is approximated in a most approximate manner. In almost all of the finite-difference models reviewed, the geometrical and physical description of radiative transfer is simplified by use of so-called flux methods (25-27) in such a way that levels and profiles of predicted temperatures can be doubtful. Thus, more sophisticated radiation models have to be coupled to finite-difference furnace models to make them a reliable tool for boiler performance predictions. This would increase computing times which are already large because of the fully coupled numerical solution procedure. An interesting new development is the discrete transfer method which promises a fast and fairly accurate computation in combination with finite-difference modeling (28).

Zone models allow a more realistic simulation of directional radiative heat exchange in the furnace enclosure especially for complex 3-D boiler furnace geometries. This, of course, is achieved on the expense of resolution of the temperature distribution because of use of coarser control volumes. In most of zone methods the flow pattern necessary to solve total energy balance has to be prescribed. However, it has been shown (29) that overall thermal performance predictions of boiler furnaces are not very sensitive to details of velocity field due to the integral nature of radiative transport. Thus, it is appropriate in more cases to base the flow field description on isothermal physical modeling, sometimes a pure engineering guess can be sufficient. An interesting alternative to this approach is decoupled computational flow modeling (30). This approach is pursued by at least three boiler manufacturers or utility organizations (31-33). Figure 4 shows an impressive example of such a decoupled 3-D flow computation carried out for a tangentially fired boiler (33). In this example ca 7200 control elements of flexible shape were used. However, in order to decrease the computational effort and to stabilize the solution, turbulent exchange was modeled with a simple constant eddy viscosity concept. The velocity field obtained by decoupled modeling such as that shown in Figure 4 are integrated to provide automatically mass fluxes for zone methods with coarser control volume arrangement.

The zone models themselves can vary in their degree of complexity. The number of volume and

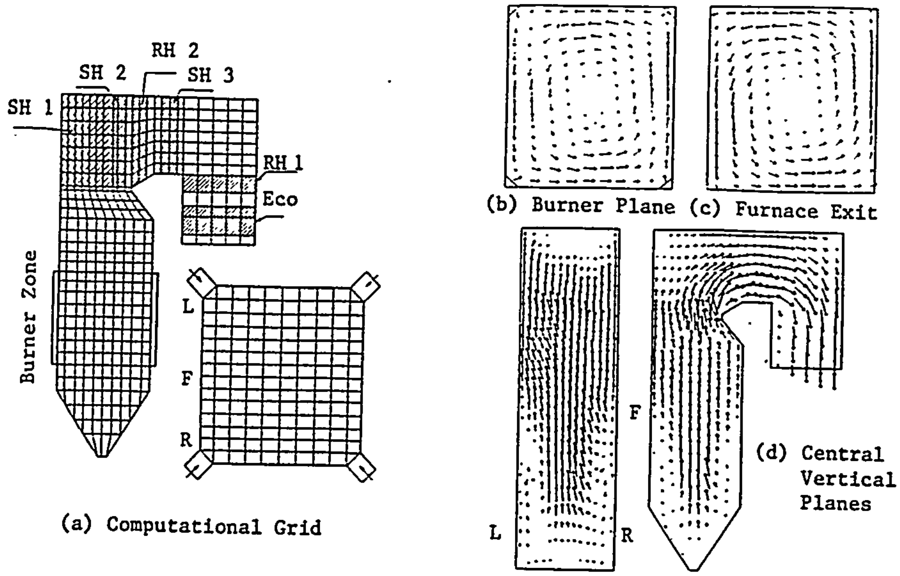


Fig. 4. Computational flow modeling of 500 MW_e tangentially coal fired boiler (Figure taken from Ref. 33).

surface zones for which heat balances are solved may vary between one and several hundred. The zone models also differ considerably in the way the heat release pattern and radiative properties are evaluated. Advanced methods (see also next section) are based on models for fuel burn-out rather than on pure prescriptions.

According to the way of handling radiative heat transfer, zone methods can be loosely grouped into classical methods and so-called Monte Carlo calculation techniques. The classical methods which were first published by Hottel and co-workers (34,35), use precalculated or tabulated radiative heat exchange coefficients for total energy balances. This saves computing time, however, these methods are relative inflexible with respect to furnace and zone geometries and handling of local dependencies of radiative properties. Direct exchange area for cubes and for concentric cylindrical zones are published in (34-35), and for concentric rectangular zones in (36). Monte Carlo methods (37-38) allow direct evaluation of zonal radiative interchange and direct coupling with furnace total energy balance. They are extremely versatile with respect to zone shape, arrangement and fineness of zoning. They can account for dependency of radiative properties on local temperature, and for spectral and directional dependencies of these properties. They can also easily include anisotropic scattering effects.

The advantage of zone models to provide a realistic prediction of radiative heat transfer is the reason, why, with exemption of Zuber's classical finite-difference boiler model (22-24) these methods are the only ones which have been actually used by boiler manufacturers and utility organizations to support design and analysis of boiler furnaces. Table I lists the most important zone models of boiler furnaces published in literature together with some characteristics features of the individual approaches (39-52). The table also includes the boiler models of Lockwood (50) and Babcock & Wilcox (51). Computation of radiative heat transfer in these models is based on the "Discrete Transfer" Method which allows use of finer control volumes but the models have otherwise all the characteristics of zone methods. Most of the recent applications of the zone models made use of the Monte-Carlo calculation techniques because of their extreme flexibility mentioned above. A representative example for heat transfer predictions obtained with these methods is shown in Figure 5. This figure compares net heat flux densities at the furnace walls measured and predicted with the model of Ref. (41) for a 500 MW_e oil fired boiler (43). The agreement is surprisingly good. Some local discrepancies are attributed to use of a symmetrical flow field whereas flow in the actual boiler was obviously assymetric. Each model listed in Table I deserves a more thorough description, but because of space limitations, only EER's zone model, developed by Richter et al. (29, 52-53), which is believed to represent the current state of art of practical furnace modeling is described in more detail.

EER's Furnace Model

EER's general 3-D furnace model allows predictions of local and overall heat transfer, temperature profiles and burnout of solid fuel particles in boiler combustion chambers and industrial furnaces dependent on actual furnace geometry and operating conditions, fuel characteristics and characteristics of wall deposits. All major fuel types can be considered. Additionally, unconventional fuels like methanol, coal/oil mixtures and coal/water slurries can also be studied. The code was recently extended to handle heat transfer in radiant superheater and reheater sections simultaneously with furnace heat transfer. A special 2-D version of the otherwise identical furnace code is also available. This version can be applied to cylindrical furnaces fired with an axisymmetrical flame.

The model is basically a zone method. The flow pattern in the furnace necessary to solve the total heat balance of volume zones is prescribed in decoupled manner. The distribution of heat release from gaseous fuel components or volatile matter is calculated automatically from the prescribed flow

Table I Zones models for boiler furnaces.

AUTHORS	REFS.	YEAR	MAJOR FUELS	TYPICAL ZONING	ZONE SHAPE	RADIATION MODEL	FLOW PATTERN	HEAT RELEASE PATTERN
Hottel & Sarofim	39	1978	All	1	Rectangular	Mean Beam Length	Well-Stirred	Well-Stirred
Johnson et. al.	40	1978	Lignite, Coal	2 x 10	Concentric Rectangular	Exchange Coefficients	Engineering Guess	Prescribed for Vol., Balance for Char
Arscott, Gibb et. al.	41-43	1973	Oil, Coal	4 x 6 x 11	Arbitrary Prismatic	M-Carlo, NE Diffusion	Iso. Phys. Models, Simpl. Comp. Flow Modeling	Vol: Full Scale Burner Testing Coal: Simple Model
Steward & Guruz	44	1974	Oil	2 x 3 x 5	Cubes	Exchange Coeff. from M-Carlo	Engineering Guess	Prescribed
Rueters et. al.	45	1975	Gas, Oil, Coal	1 x 25	Rectangular	Radial: Mean Beam Length Axial: Flux Method	Engineering Guess	Prescribed
Lowe et. al.	46	1975	Coal	3 x 15	Concentric Cylindric	Exchange Coefficients	Engineering Guess	Prescribed for Vol. Balances for Char
Xu	47	1981	Coal	10 x 10 x 10	Parallelepipeds	M-Carlo	Iso. Phys. Model	Prescribed
Fiveland et. al.	48	1982	Coal	13 x 6 x 24	Arbitrary Prismatic	Discrete Transfer Method	Computational Flow Modeling	From Overall Fuel Balance
Lockwood et. al.	49	1983	Coal	5 x 5 x 12	Arbitrary Prismatic	Discrete Transfer Method	Engineering Guess	Prescribed
McHale et. al.	50	1983	Coal	5 x 3 x 3	Arbitrary Prismatic	Exchange Coeff. from M-Carlo	Iso. Physical Model	Empirical Distribution Function
Sakai et. al.	51	1984	Coal	up to 10 x 10 x 20	Cubes	Exchange Coeff.	Iso. Phys. Model, Computational Flow Modeling	Empirical Distribution Function
Richter et. al.	29,52 53	1980	All incl. Slurries	5 x 5 x 16 3 x 11	Arbitrary, Prismatic, Concentric, Cylindrical	Semistochastic Method, Radiant Heat Exchangers Included	Iso. Phys. Models, Engineering Guess	Statistical Distribution Model for Vol., Balances for Char

pattern and from estimated mixing (burning) times. Burnout of char particles and corresponding heat release is calculated from mass balances of fixed carbon which are also based on the prescribed flow pattern. The zone arrangement can have a variable fineness. The volume zones themselves may have a general prismatic shape. Thus, almost any furnace geometry and firing pattern encountered in practice may be investigated. Typical zone arrangements used in the application of the model to a pilot-scale and two boiler furnaces are shown in Fig. 6. This figure also shows how the geometry of actual radiant superheater and reheater sections in the upper furnace of a boiler can be simulated in the furnace model.

The radiation model in the furnace heat transfer code is derived from Monte Carlo calculation techniques. Some features of the radiation model are indicated in Fig. 7. The emissive power of each furnace zone is distributed among a discrete number of beams of unit radiation, which are traced through the arrangement of furnace zones. The energy fluxes of the beams are gradually attenuated and redirected due to wall reflection and eventual scattering until final absorption occurs. At the end of the radiative exchange calculation a balance is set up for each zone. The difference between the sum of energy fluxes emitted by a zone and the accumulated absorbed energy fluxes is the net radiative heat flux necessary to solve the total heat balance of this zone. In pure Monte Carlo methods all steps in the history of a radiation beam are determined with help of random numbers. However, in order to reduce the statistical error for comparative calculation times, some random steps of such a pure Monte Carlo approach were replaced in our model by deterministic decisions, resulting in a so-called semistochastic model for furnace heat transfer predictions (52). The accuracy of the semi-stochastic method is increased as more beams are traced. It is possible to obtain an acceptable accuracy of overall heat transfer with less than 10000 beams (29). Radiating species considered in our current model are H_2O, CO_2, soot, char, and ash particles. Nongray radiation of the gaseous species H_2O and CO_2 and of soot is simulated with weighted gray gas approach following a suggestion of Johnson and Beer (54). However, Johnson's "one clear two gray" gases approach for real gas radiation derived from the emissivity charts of Hottel (35) was replaced by a "one clear three gases" approach suggested by Smith et. al. (55) which is based on Edwards experimental white band model (56-57). This was done because the performance calculations for large gas fired boilers yielded better agreement with measured performance data using Smith's instead of Johnson's approach. Radiation of char and ash particles is assumed to be gray. The absorption coefficient of char particles is calculated from local size distributions predicted by the char combustion model.

The present furnace model can take anisotropic scattering of radiation at ash particles optionally into account. In this case, effective scattering and absorption efficiencies as well as phase functions of ash particle clouds are calculated from measured or assumed ash particle size distributions and from complex refractive indices from literature (58) using Mie theory (59). The model of radiative exchange is directly coupled with total heat balance for all gas and surface zones of the furnace of unknown

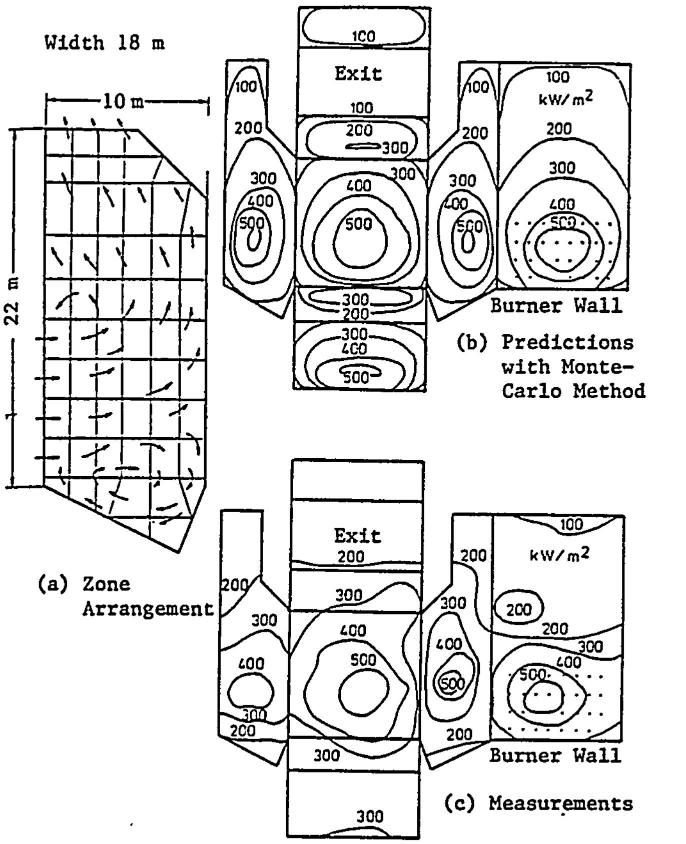


Fig. 5. Distribution of net heat flux densities (kW/m^2) in a 500 MW$_e$ oil-fired boiler (Figure taken from Ref. 43).

temperatures, and, if required, for all radiant superheater and reheater surfaces. Radiative and total heat balances are based on locally prescribed thickness of ash deposits and values of emissivity and thermal conductivity available from literature (60-62). The output of the total heat balance of the surface zones is the distribution of surface temperatures of the deposits. Any wet slag spot due to local attainment of ash fusion temperature can be identified. This is also possible for every superheater and reheater surface area.

The total heat balance for volume zones is based on decoupled prescription of mass flux vectors at each zone boundary obtained indirectly or directly from physical flow modeling (30). An "intelligent' automatic program coupled to the furnace model corrects the flow field for continuity. The correction program is based on a stochastic tracking procedure of fluid lumps and yields, for a given zone arrangement, a flow field free of mass sources which is as similar as possible to the uncorrected specified flow field. The automatic continuity correction subroutine is also used to generate, from a given flow pattern, approximate mass flow fields for furnace operating conditions in which burners are unbalanced or taken out of service in order to reduce load.

It was found that, even for a coarse zone arrangement, turbulent transport of energy between zone pairs must be accounted for. Therefore, a turbulent field is superimposed on the mean mass flow field by assigning additional mutual mass flow vectors at non-wall boundary surfaces with help of a simple model of turbulence. The resulting turbulent flow field, can be verified with isothermal physical flow models by comparing measured and predicted tracer concentrations and, specially, by comparing the dispersion of photographed and numerically generated steaklines (30) (see also Fig. 18).

The heat release distribution for a given zone arrangement is evaluated in the following way. Heat release due to burning of gaseous fuels or volatile matter is either prescribed a priori or, alternatively, calculated from a simple transport model. It is assumed that lumps of gaseous fuel or instantaneously released volatile matter are transported with the main turbulent flow. These lumps are tracked within the furnace zones and the lifetime and associated heat release from the individual lumps is calculated statistically from weighted random numbers taking into account the fact that life expectancies follow certain exponential decay functions. Heat release and burnout of residual char particles is calculated from mass balances. It is assumed that the particles will follow the main flow without slip and that devolatilization is completed in the first downstream burner zones. The mass balances are set up for 10 different particle size classes and O_2 concentration. The particles burn with decreasing diameter under combined kinetic and diffusion control. In order to compute more accurately the influence of water evaporation during CWF combustion the burn-out model was recently extended to account for droplet tracking and evaporation. The burn-out model which is directly coupled to the iterative solution of the total heat balance allows also the calculation of zonal concentrations of gaseous and solid species from which local radiative properties of the combustion products are determined. In particular, local soot concentrations are assumed to be proportional to the local amount of unburnt volatile matter.

The constant of proportionality is related to the carbon content of volatile matter obtained from the proximate analysis and to an empirical factor.

In conceptualizing the model described above care was taken to model the individual processes affecting thermal performance of boiler furnaces within a comparative degree of complexity. Nevertheless, due to simplifications necessary and due to insufficient physical knowledge, several model uncertainties still exist, especially with respect to

- Reactivity constants in the burn-out model for individual fuels
- Radiative properties like soot concentration fly ash size distribution, surface emissivities
- Formation of wall deposits

However, as shown in the following, some of the uncertainties can be minimized by "calibrating" the model with results obtained from pilot-scale testing for individual fuels and firing conditions. The calibrated model can then provide a more reliable scale-up for particular furnaces than any empirical extrapolation of the pilot-scale data.

EXAMPLES FOR PILOT SCALE STUDIES AND SCALE UP

The scale-up procedure suggested relies on almost

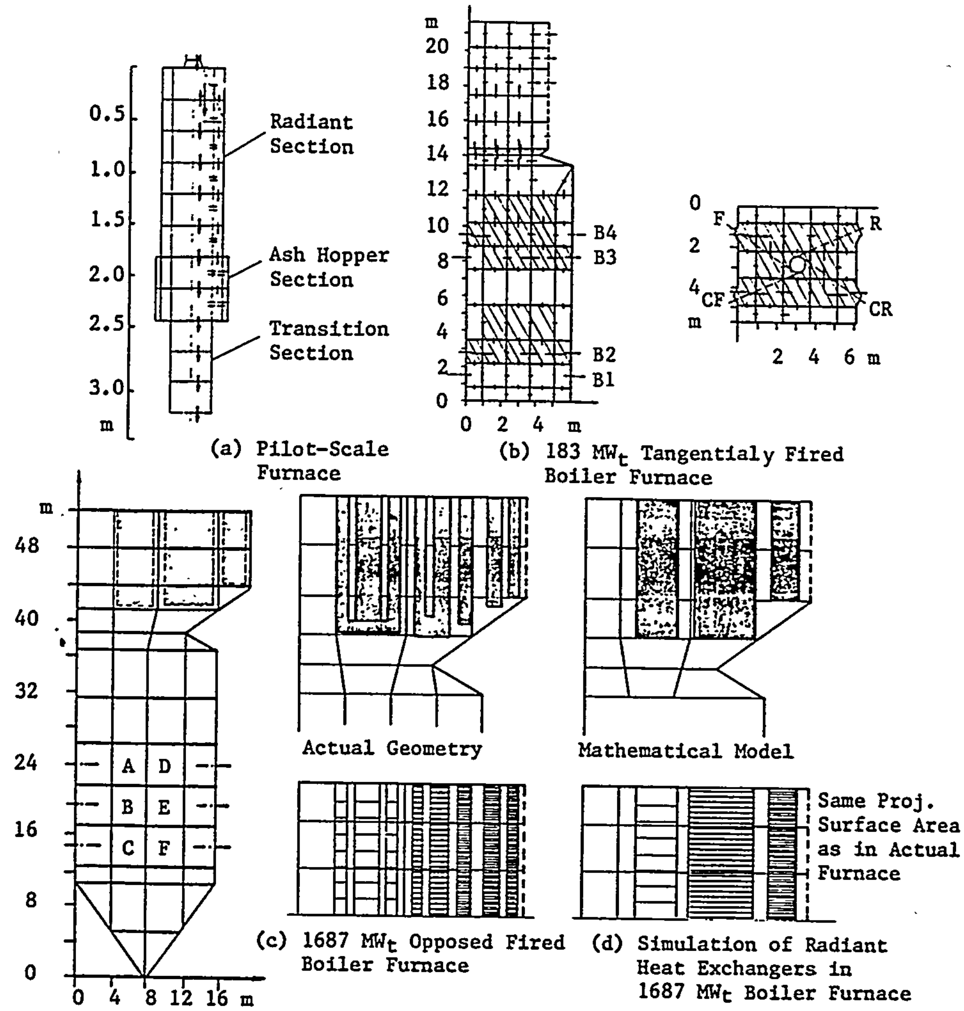


Fig. 6 Typical zone arrangements in EER's furnace heat transfer model.

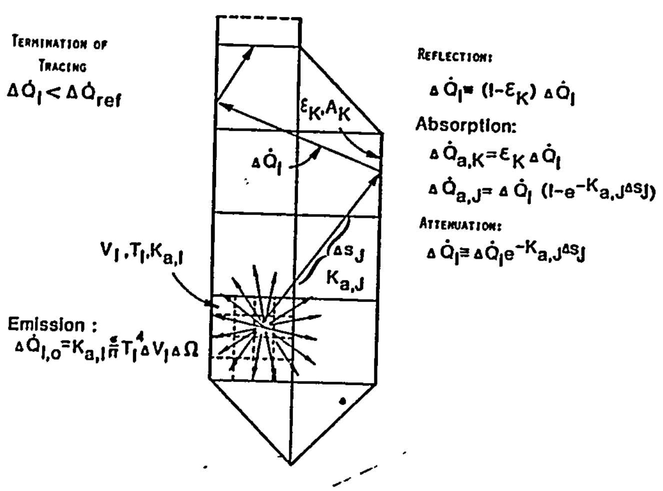
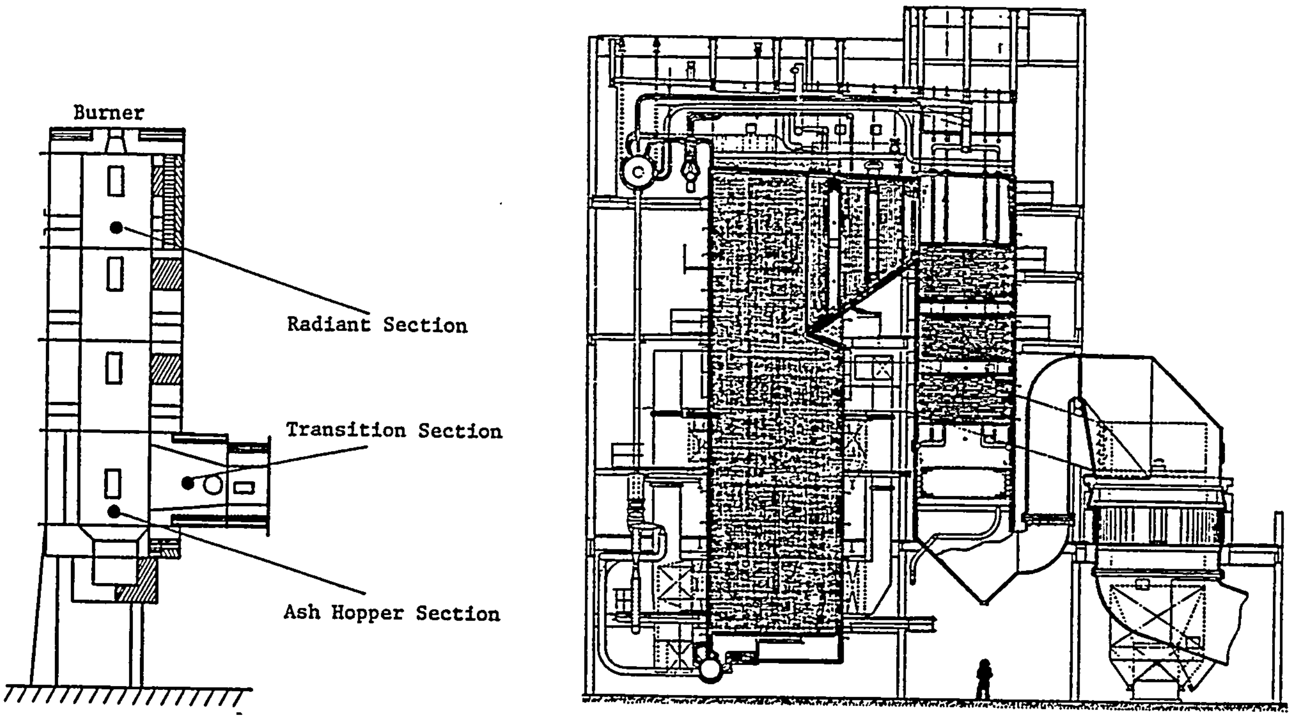


Fig. 7. Some features of the semi-stochastic radiation model.

identical versions of the mathematical model described above for a pilot and boiler furnace. The only differences is that the procedure used for the pilot scale furnace assumes axisymmetry and is two-dimensional whereas the boiler model does not, is three-dimensional and can handle any complex furnace geometry. The following examples show results for application of the 2-D model to a pilot scale furnace fired with a high-volatile coal and a CWM, respectively. The furnace shown in Figure 8a has a nominal heat input of 260 kW. It can be operated at volumetric heat release rates typical of boilers and is specifically designed to determine flame stability, burnout characteristics and slagging fouling and erosion potential of fuels. The pilot scale furnace was modeled using the axisymmetric zone arrangement and flow distribution shown in Fig. 6a. The combustion is divided into 11 axial zones and up to 3 radial zones.

Figure 9a compares gas and wall temperatures measured and predicted along the furnace for coal and slurry fining, respectively. In both cases, good agreement is achieved for the axial profiles. Figure 9b compares axial burn-out of combustibles measured and predicted in the pilot scale furnace. The burn-out is plotted on a sensitive logarithmic scale. The predictions for the coal were carried out with a



(a) Pilot Scale Furnace
260 kW Nominal Load

(b) Tangentially Fired Boiler Furnace
183 MW_t Load for Oil Firing per Half of Twin Furnace

Fig. 8. Furnace investigated in scale-up study.

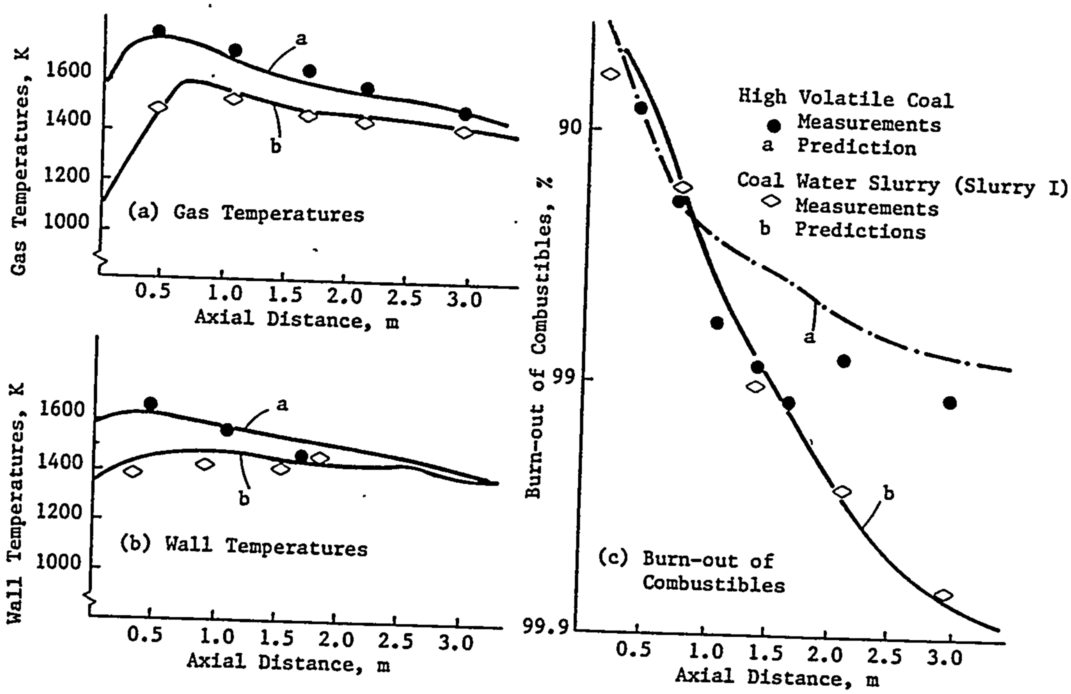


Fig. 9 Axial profiles of furnace variables in pilot-scale furnace.

standard set of reactivity constants i.e. with a Q-factor[1] of 1, a frequency factor A for char combustion of 1200 kg/m^2s bar and an apparent activation energy E of 90000 kJ/kmole. No adjustment was made in this case to achieve a close agreement with the measurement. In case of the slurry, the Q-factor was increased from 1 to 1.5 in order to match measured burn-out at the end of the transition section and this value was used for subsequent full scale performance predictions for that slurry.

The pilot-scale combustion experiments for the slurries were carried out with the aim to evaluate its performance in the particular boiler shown in Fig. 8b. The 132 MW_e boiler has twin furnaces divided by a center wall. Each furnace has a tangential firing system with four burners at each of four levels. The boiler is currently fired with natural gas or residual oil.

The 3-D version of the heat transfer model calibrated as described above was used to scale-up the slurry combustion tests. Zoning and flow pattern used for the boiler furnace predictions are shown in Fig. 6b. Figure 10 compares mean-cross-sectional time-temperature histories predicted with the 2-D and 3-D model for pilot-scale and boiler furnace, respectively, fired with the same slurry. The residence times of boiler and the furnace are about 2 sec; however, in the boiler, the total upward mass flow varies between burner levels and between furnace exit elevations. The temperature peaks in the boiler furnace are about 70K lower than in the pilot scale furnace. The peak temperature difference is reduced to only 15K when formation of an ash deposit layer with 2 mm overall thickness is assumed. Since the time-temperature histories in pilot and boiler furnace are in fair agreement it is believed that the boiler performance predictions obtained with the model calibrated for the pilot-scale furnace are quite reliable. The confidence is supported by the fact that the 3-D model could accurately predict temperature profiles independently measured in full-scale trials in the same boiler fired with natural gas (see Fig. 15a).

Overall performance predictions for the boiler furnace fired alternatively with gas, oil or various coal slurries are compared in Table II. Burn-out values obtained for the slurries in the pilot scale furnace are also listed in this table. Although the same net fuel heat input is assumed boiler furnace efficiencies for slurry firing are considerably reduced compared to oil firing. This is primarily caused by the decrease of the furnace temperature level due to evaporation of fuel moisture (Figure 11). For slurries with bad burnout behavior furnace efficiency is also decreased because of ignition delay of the solid particles.

Predicted combustion efficiencies for the boiler show, in general, the same trend as those predicted and measured in the pilot scale furnace. However, at same excess air level they can be lower up to 4 percentage points in the boiler furnace than in the pilot scale furnace. This underlines again the usefulness of the mathematical model for scaling up the pilot furnace experiments.

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[1]The Q-factor relates the amount of volatile matter actually released from a coal particle to the value of the proximate analysis.

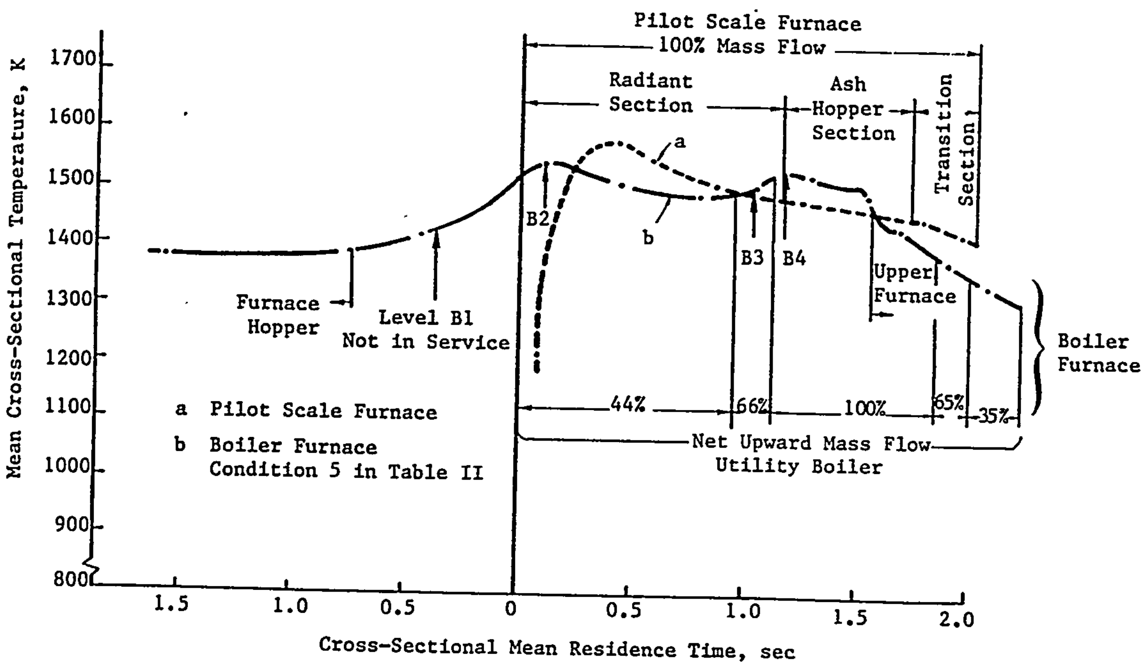


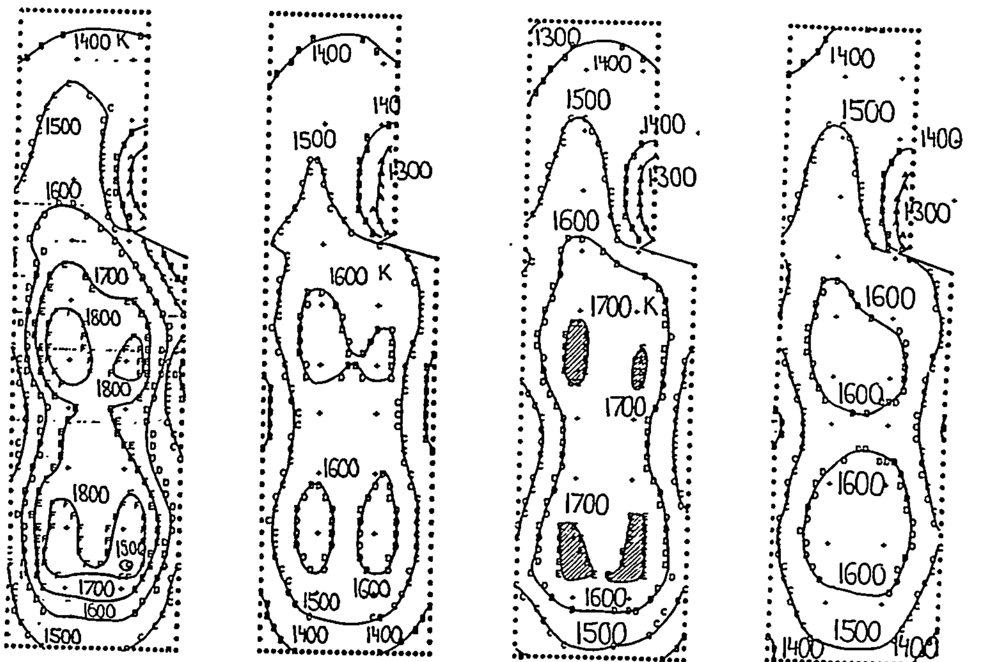
Fig. 10. Comparison of mean time temperature profiles predicted for pilot scale and boiler furnace fired with same coal water slurry (Slurry I).

Table II Overall performance predictions for 183 MW_t boiler furnace tangentially fired with natural gas, oil and various coal-water slurries.

NO	CASE	EXCESS AIR %	η_f %	T_{ex} K	MAX. NET HEAT FLUX DENSITY kW/m²	MAX. FLAME TEMP. K	BURN-OUT OF COMBUSTIBLES BOILER PREDICTED %	PILOT SCALE PREDICTED %	PILOT SCALE MEASURED %
1	Natural Gas	11.5	40.8	1529	292	2150	100	—	—
2	Oil No. 6	11.5	50.1	1388	389	1996	100	—	—
3	Oil No. 6	30	43.6	1392	392	1936	100	—	—
4	Slurry I	30	32.7	1381	240	1661	99.4	—	—
5	Slurry I	24	33.4	1368	253	1674	99.1	99.9	99.8
6	Slurry II Fine	30	41.2	1377	276	1765	99.3	99.7	99.1
7	Slurry II coarse	30	36.8	1401	234	1642	92.4	96.1	96.7
8	Slurry I 2 mm Deposit	24	32.1	1407	195	1706	99.4	—	—

evaluate the time-temperature distribution in the upper furnace is nearly impossible. The following example demonstrates the ability of the model for temperature and heat transfer predictions in this section utilizing a detailed model of the upper furnace which is directly coupled to the main furnace program. The example was obtained for a 1687 MW_t opposed fired boiler furnace. The zoning of the furnace was shown in Fig. 6c together with the detailed geometrical simulation in the model of the actual radiant superheater and reheater sections. The flow pattern in this case was based on observations in a 1/30 isothermal physical flow model of the same boiler.

Figure 12 shows the distributions of gas temperatures predicted for 100% load in a central vertical plane of the boiler. The isotherms in Figure 12a were obtained neglecting heat extraction by superheaters and reheater whereas this effect was included in Figure 12b. The temperature predictions in the upper furnace were verified by full-scale measurements with a multi-shielded water cooled



(a) Oil No. 6 (b) Slurry I (c) Slurry II, Fine (d) Slurry II, Coarse

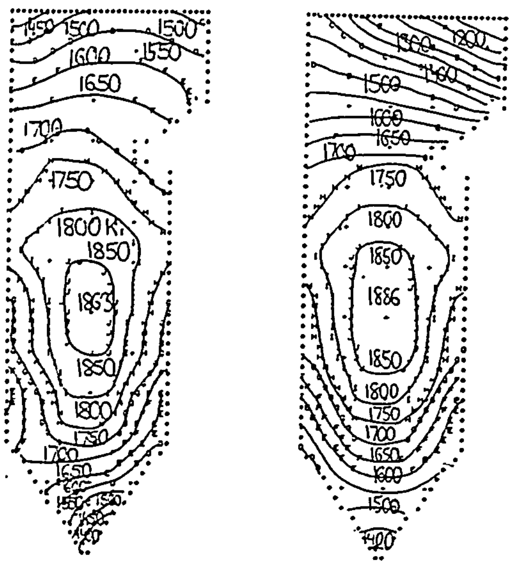
Fig. 11. Temperatures (K) predicted in center plane of full scale boiler furnace fired with various coal-water slurries and comparison to oil firing.

EXAMPLE FOR PERFORMANCE PREDICTIONS OF COAL FIRED BOILER FURNACES

1687 MW_t Opposed Fired Boiler

The assessment of many of the problems associated with coal-fired boilers requires a knowledge of both furnace temperatures, and the gas temperature range encountered in radiant superheater and reheater sections. An accurate assessment of time-temperature histories in the upper furnace is especially important for efficient application of the limestone injection process for SO_x control. Pilot scale testing to

suction pyrometer. The boiler load during these measurements differed somewhat from the loads for which the predictions were made at an earlier stage. However, taking the load effect into account the agreement between the temperatures measured and predicted just behind the inlet plane into the radiant superheater is good (Figure 13). The impact of the superheater and reheater on heat transfer can also be seen in Figure 14 which shows the complete distribution of net heat flux densities predicted at the walls of the lower and upper furnace. The minimum of net heat fluxes above the upper burner rows is due to the insulating effect of a thick layer of ash



(a) Heat Extraction by Radiant Heat Exchangers Not Included

(b) Heat Extraction by Radiant Heat Exchangers Included

Fig. 12. Temperature distribution (K) predicted in central plane of 1687 MW_t opposed coal fired boiler furnace.

Table III Overall performance predictions of 1687 MW_t opposed fired boiler furnace for various operating conditions.

NO	CASE	η_f*) %	T_{ex}*) K	MAX NET HEAT FLUX DENSITY kW/m^2	MAX. FLAME TEMP K	CARBON CONTENT OF FLY ASH %
1	684 MW_e	56.2	1255	260	1880	14.9
2	556 MW_e, All Burners	59.2	1208	278	1858	13.2
3	556 MW_e Top Burners Only Air	59.2	1202	320	1949	19.8
4	684 MW_e Clean Furnace	60.5	1182	329	1840	17.1

*)Heat Extraction by Radiant Heat Exchangers Included

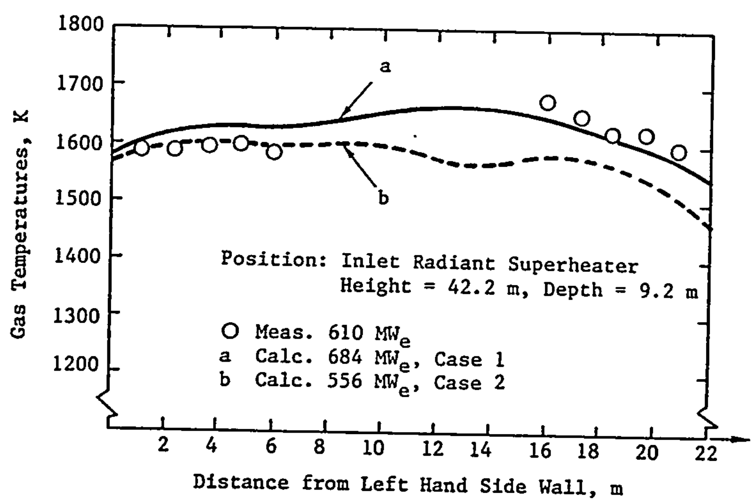


Fig. 13 Comparison between temperature profiles measured and predicted in upper furnace of 1687 MW_t P.F. fired boiler.

deposits assumed at this elevation. Table III lists the overall performance predictions of the boiler furnace for various operating conditions. Besides the load furnace exit temperatures T_{ex} and efficiency η_f are predominantly influenced by the cleanliness of the furnace. Assuming an effective heat conductance of $k = 0.0008$ kW/mK and an overall ash deposit layer on the furnace walls of $\Delta s = 0.5$ mm instead of a layer with $\Delta s = 2$ mm as well as thick slag layers above the burner belt of $\Delta s = 10$ mm increases furnace efficiency η_f by 4.3 percentage points corresponding to a decrease of T_{ex} of 73 K.

Further Boiler Furnace Predictions

The furnace heat transfer model described in the foregoing sections has been successfully applied in cooperation with boiler manufacturers and utilities for many performance predictions of gas-oil-and coal fired boilers located all over the world (53). It has been verified several times by full-scale measurements as shown in Figure 15 which compares temperature profiles measured and predicted near lower furnace exit planes of boilers of different design and rating and fired with various fuels.

One study carried out in cooperation with a boiler manufacturer showed that the model is able to predict accurately the effect of volatile matter

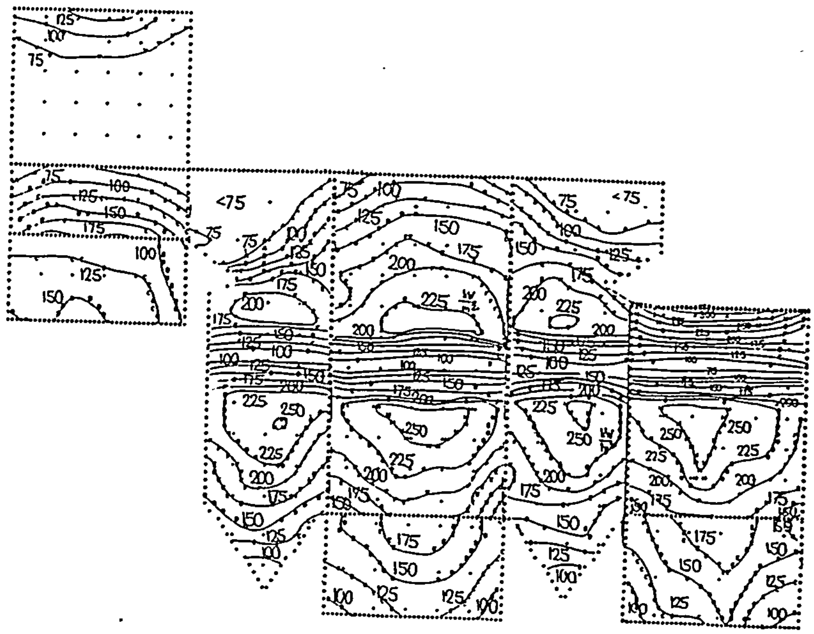


Fig. 14 Distribution of net heat flux densities (kW/m^2) predicted in 1687 MW_t P.F. fired boiler (Case 1).

content of coal and initial coal particle size distribution on carbon content in ash measured in large p.f. fired boilers (53).

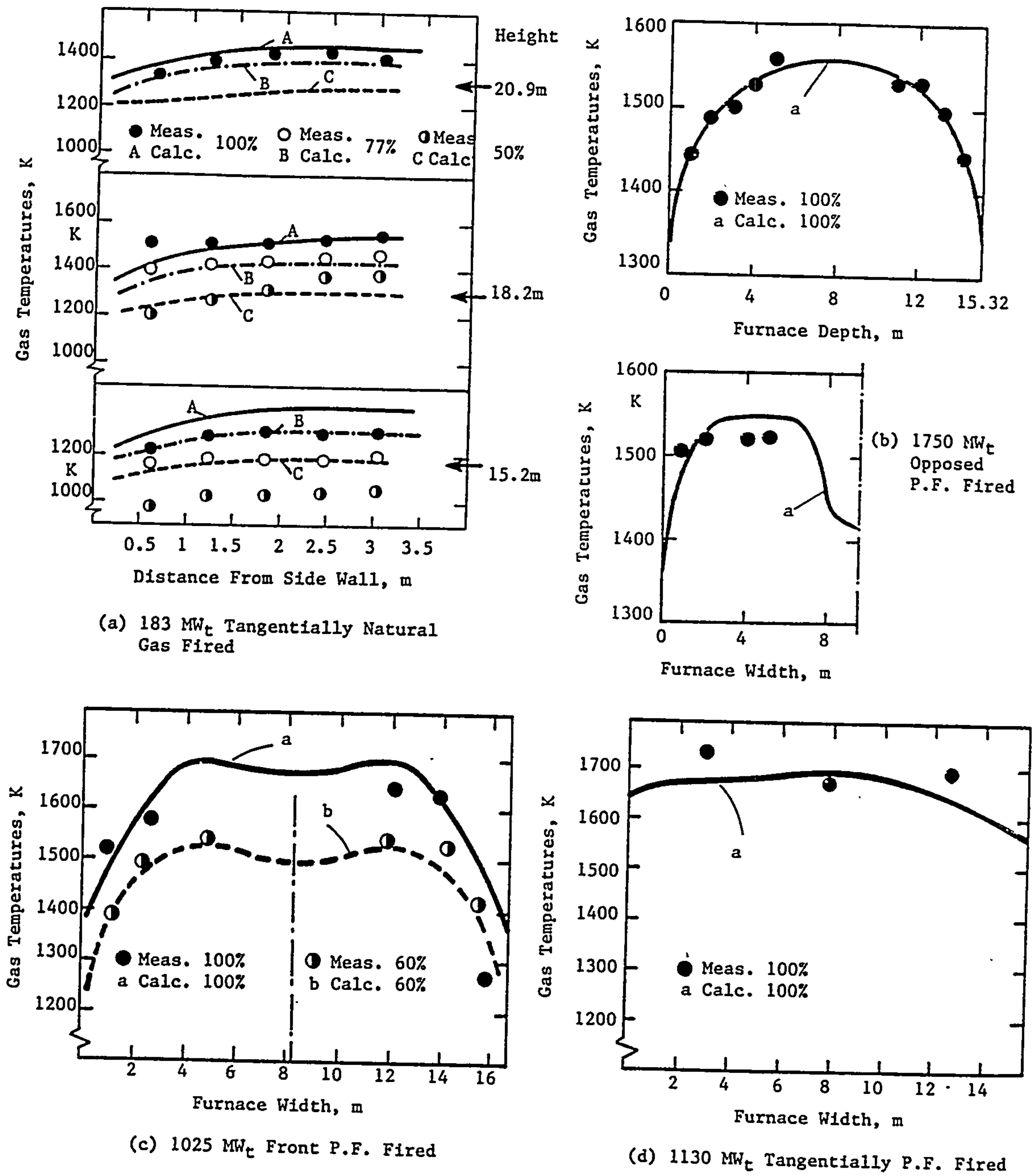
In another application, the effect of scattering of radiation at fly ash particles was investigated. It was found that for a large combustion chamber (19 x19 x 65m) fired with a 30% ash coal furnace efficiency decreased by 3.3 percentage points due to anisotropic scatter. The effect of scattering in smaller furnaces fired with low ash coals, however, was found to be much smaller (53).

Current applications of the models are especially carried out to study the effect of fuel conversion on boiler performance (63-65) and to screen time-temperature profiles of existing boilers for application of the dry sorbent injection process for SO_x control. Typical mean time-temperature profiles predicted with the model for several existing boiler furnaces are shown in Figure 16. Such predictions are used to optimize the injection location of the sorbents.

A generalization of performance predictions for a number of p.f., CWM and COM fired boiler furnaces is shown in Fig. 17. In this figure, the mean temperature predicted for horizontal cross-section through the furnace nose is plotted against the net head releasé per projected surface area $\dot{q}_S$. Different slagging and fouling conditions in the various furnaces were taken into account specifying the distribution of the effective heat conduction coefficients at the furnace walls either based on observations in actual furnaces or on engineering judgement. The temperature can clearly be correlated to the cleanliness of the furnaces. Despite the scatter in data which is largely due to different furnace design and operating conditions, and to a minor extent to model assumptions, exit temperatures of furnaces with slagging are of the order of 100 K higher than those for clean furnaces.

OUTLOOK TO AN ASH DEPOSITION FORMATION MODEL

The major uncertainty of the type of models described above and applied to p.f. fired furnaces lies in the fact that the ash deposition patterns in the furnaces are generally unknown and have to be specified prior to any performance calculations. A major improvement would be to couple a model of ash transport directly with the furnace heat transfer model. It is clear, that a model of the deposit process based on first principles is still too difficult to construct. But some useful answers are expected to be obtained from a semi-empirical model currently under development. Some features of that model are depicted in Figure 18. The left hand side of this figure shows turbulent particle streaklines computed and observed in the isothermal model of a large p.f. boiler. The computer simulation was carried out with a finite-element type approach using the same zone arrangement as in the heat transfer model. It was assumed in the computation, that the particles follow the turbulent flow field without slip like it was the case for the observed particles which were small neutrally buoyant He - bubbles. The right hand side of Figure 18 compares computed and observed slagging pattern in the upper furnace of the boiler. The computations were still carried out by prescribing the thickness and extension of the deposit layers based on observation. Molten slag deposits (indicated by the star symbols) are predicted for the lowest part of the superheater surfaces. Wet spots in this area were also observed in the field. A future task is to combine the particle transport model, known thermal characteristics of the particles and deposit layers in order to estimate deposition rates. This empirical model has to take into account boundary layer characteristics and effect of particle size on the stopping distance. Calculation of injections of larger particles will be based on particle momentum balances. Figure 19 demonstrates the size effect on large particles on wall collisions in a tangentially fired boiler. In this example, a 200 μm particle emerging from an upper level burner will impact the wall whereas a 100 μm particle can exit the furnace. In the calculation of Fig. 19, turbulence was neglected, however the current dynamic model is already capable to include the effect of gas phase turbulence on the particle path in a similar way as shown for smaller particles in Fig. 18.



(a) 183 MW_t Tangentially Natural Gas Fired

(b) 1750 MW_t Opposed P.F. Fired

(c) 1025 MW_t Front P.F. Fired

(d) 1130 MW_t Tangentially P.F. Fired

Fig. 15. Model verification of exit temperatures from lower furnace of various boilers.

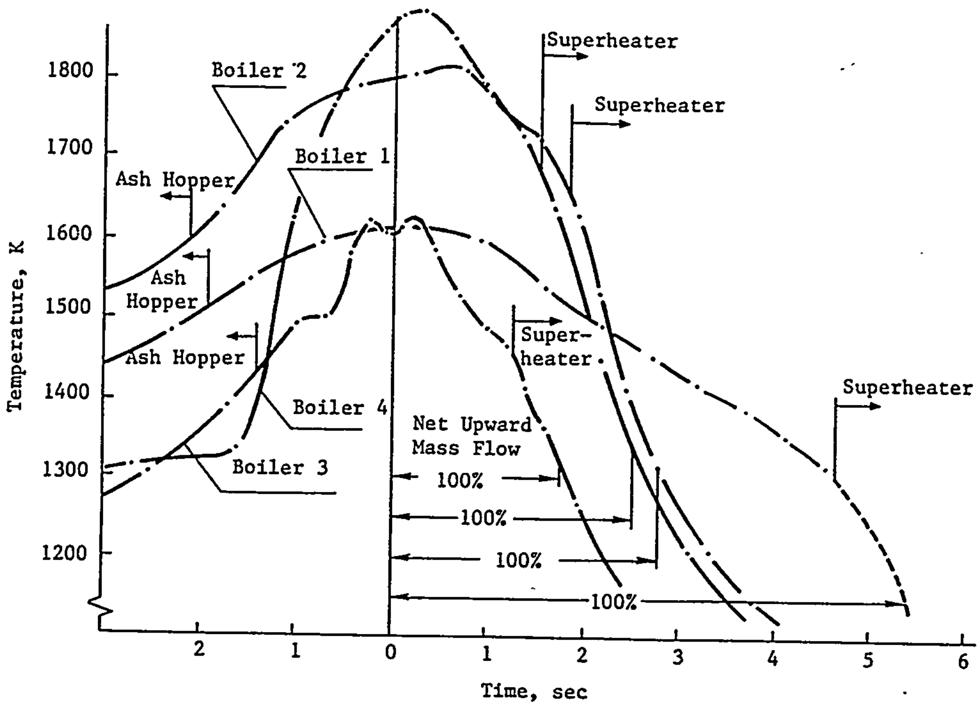


Fig. 16 Mean cross-sectional time-temperature profiles predicted for various boilers.

CONCLUSIONS

The selected examples have demonstrated that advanced zone models are well-suited as engineering tools for simulation and performance analysis of newly designed or existing boiler combustion chambers. Even, when fully coupled 3-D finite-difference models with improved handling of radiative transfer, reduced computational expense and more directed towards practical applications will become available in the near future, many furnace operating problems can still be more economically solved with zone models of the type described in this paper.

The uncertainty of absolute values of furnace variables predicted with these models is probably of the same order as the uncertainty due to unknown actual operating conditions of individual boiler furnaces. Absolute temperatures can typically be predicted within a range of $\pm$ 40 K and heat flux densities within $\pm$ 10%. For p.f. fired boilers, major uncertainties are caused by the unknown slagging and fouling pattern in the furnace. An ash deposition model could help to reduce these uncertainties.

Besides their ability for absolute predictions within the above range, the models are especially useful to study the impact of relative changes of geometrical and operational parameters on furnace heat transfer including radiant heat exchangers. They are a powerful tool for fuel conversion evaluation. As demonstrated in this paper, they allow the scale-up of pilot-scale trials carried out for individual fuels to full-scale performance predictions of furnace and combustion efficiencies, and consequently overall efficiencies of particular boilers.

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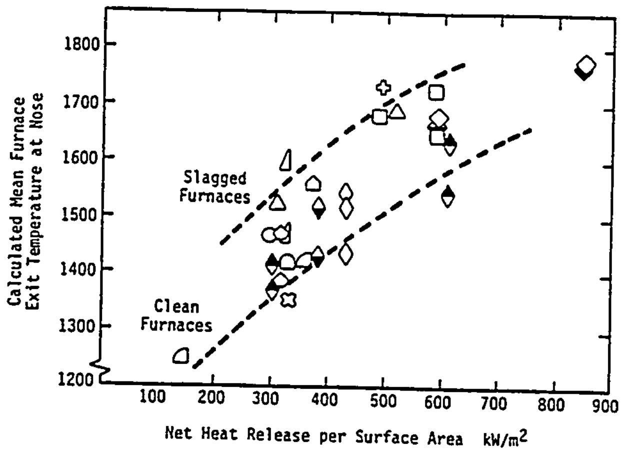

Furnace Nr.& Sym.		Firing Pattern	Fuel	Surface Area m^2	$(k/\Delta s)_{eff}$ kW/m^2K	Wet Slag Spots Pred.	Furnace Nr.& Sym.		Firing Pattern	Fuel	Surface Area m^2	$(k/\Delta s)_{eff}$ kW/m^2K	Wet Slag Spots Pred.
1	◇	Tan.	Bit.	1748	0.8 0.32 Local	No	6	⌂	Front	Bit.	538	0.4	No
	◆	Tan.	CWM	1748	0.8 0.32 Local	No	7	⌂	Opp.	Bit.	5254	1.6	No
2	□	Opp.	Bit.	3103	1.6	No	8	✧	Tan.	Lig.	5120	1.6	No
					0.4 0.08 Local	Yes	9	⌂	Opp.	Bit.	2862	0.4	No
3	◊	Tan.	Bit.	1535	∞	No	10	⌂	Opp.	Bit.	2151	0.25	Yes
					0.6	No						0.04 Local	
					0.6 0.14 Local	Yes						0.1 0.04 Local	Yes
	⧫	Tan.	COM	1535	∞	No							
					0.6	No	11	○	Opp.	Bit.	3080	1.6	No
	⧫	Tan.	CWM	1535	∞	No						0.4	No
					0.6	No	12	○	Tan.	Lig.	547	0.4	No
4	△	Front	Bit.	2109	0.4	Yes High Load							
5	✧	Tan.	Bit.	2407	0.4	Yes							

Fig. 17. Mean temperatures predicted in horizontal plane through furnace nose for various boilers (Figure taken from Ref. 62).

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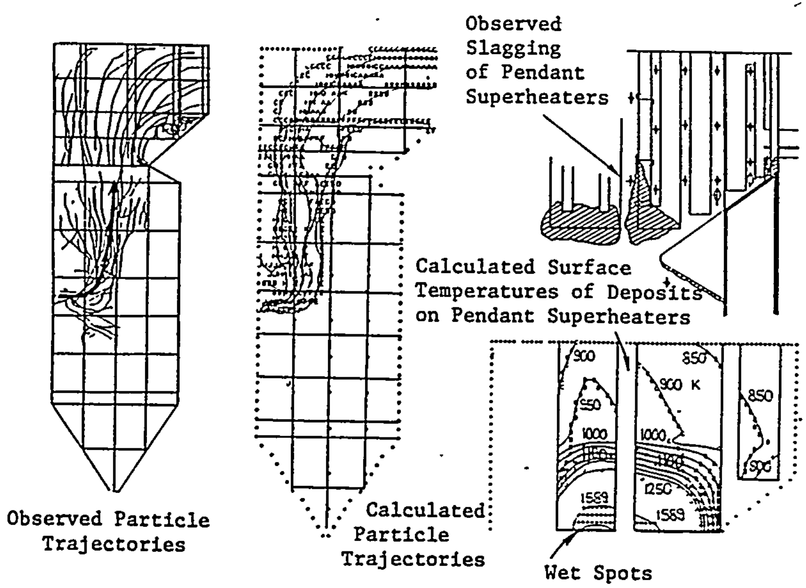

Fig. 18. Towards a model of build-up of ash deposits.

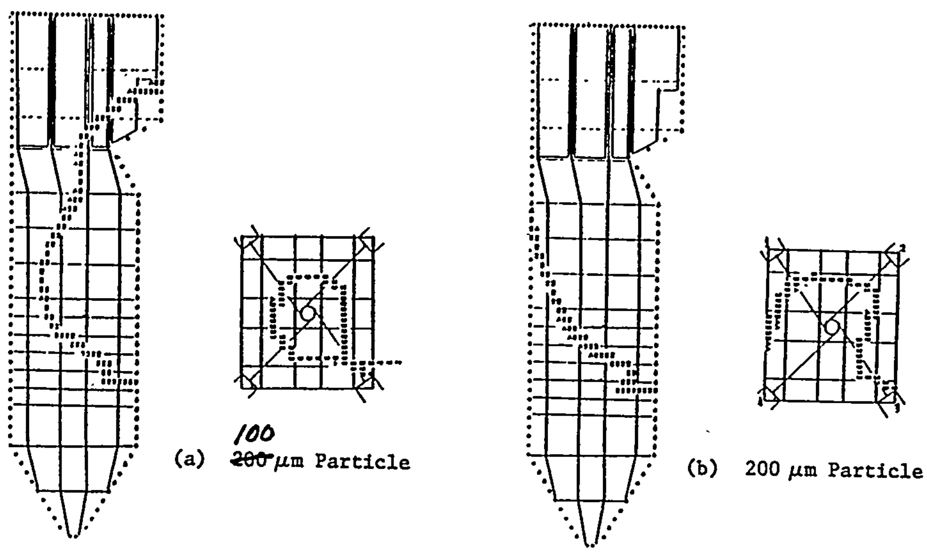

Fig. 19 Effect of particle size on wall impact in tangentially fired boiler combustion chamber.

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