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ASSESSMENT OF PULVERIZED-COAL-FIRED COMBUSTOR PERFORMANCE

PARAMETRIC SCREENING STUDIES FOR THE
CALCULATION OF HEAT TRANSFER IN COMBUSTION CHAMBERS

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1.0 INTRODUCTION

The objective of the screening studies presented in this report is to identify those parameters most influencing the thermal performance of full-scale combustion chambers, such as those of industrial furnaces or utility boilers. Special attention is given to pulverized-coal (p.f.) firing. The report consists basically of two parts. Part one deals with overall thermal behavior, expressed in terms of a furnace efficiency and a furnace exit temperature. In part two, local heat transfer, characterized by local temperature and heat flux distributions, is investigated. It is also shown how these local inhomogeneities may influence the overall performance.

The screening studies are carried out analytically.

The complexity and interaction of the various chemico physical processes in industrial combustors, especially in those fired with p.f., require that complete predictive methods of combustor performance be based on physical laws of fluid dynamics, chemical kinetics and heat transfer derived from first principles. However, since such methods require an excessive amount of computer time to obtain meaningful solutions, they are unmanageable from an engineering point of view. Thus, analytical methods allowing an engineering assessment of combustion systems are attractive. Such models use "first principle laws" (if any at all) only for certain interesting aspects of the combustion process, whereas many other aspects are simplified or even neglected. Depending on complexity, available furnace models can be classified into

- pure heat transfer models,
- coupled fluid flow, combustion, and heat transfer models.

It has been shown in ref. 1 that decoupled pure heat transfer models enable an easier access to engineering analysis and understanding of thermal performance of furnaces. Consequently, such models will be used for the following screening studies.

The pure heat transfer models utilized are based on zone methods. The number of zones; i.e., the number of control volumes or surfaces for which heat balances are solved; may vary between one and several hundred.

The "one volume zone" model, better known as the well-stirred furnace model, is the simplest approach of overall thermal behavior of furnaces. Studies of Hottel and Sarofim (refs. 2 and 3) and the present investigations have shown that the well-stirred model is a very valuable tool for qualitative predictions of overall heat transfer for almost every furnace type and size encountered in practice. Thus, such a model has been utilized to identify the key variables of overall furnace performance (section 2).

Due to the high temperature levels in fossil-fuel-fired combustion chambers (usually $>1000^{\circ}\text{C}$), the dominant mode of heat transfer to the heat sinks is by thermal radiation. This is especially true when pulverized coal is fired since here the emissivity of the furnace volume is increased by the presence of solids such as coal, char, fly ash and soot particles. In section 3, the effect of the different radiative species on furnace emissivity is quantified. The emissivity data and adequate estimations of concentration levels are then used together with the well-stirred analysis to demonstrate the relative importance of the different radiating species on overall furnace heat transfer. The well-stirred model can only empirically consider the effect of local flame zones on thermal performance. Furthermore, it cannot yield local temperatures and heat fluxes. However, such local distributions have been obtained in studies carried out with a recently developed multizone furnace model (ref. 1). The studies described in section 4 were specially conducted to show the impact of flow pattern, local heat-release and luminous-flame zones on temperature and heat flux profiles. Local and overall furnace performances for gas and coal firing have been compared. In an additional study, the influence of five different coals on local and overall heat transfer was investigated for different values of excess air.

Depending on ash properties and burning conditions, more or less severe slagging and fouling will occur in coal-fired combustion chambers; therefore, section 5 of this report shows some results of a study carried out to demonstrate the effect of ash deposits at the heat sink surfaces on local temperature and heat flux distributions as well as on the furnace efficiency of a large boiler combustion chamber.

2.0 IDENTIFICATION OF MAJOR VARIABLES INFLUENCING OVERALL THERMAL PERFORMANCE OF FURNACES

Key quantities characterizing the overall thermal performance of furnaces are the mean temperature T_{ex} of the combustion products passing through the exit and the net heat absorption $\dot{Q}_w$ by furnace walls and other heat sinks within the furnace. Provided combustion is complete and there is no mass leakage through the walls, both quantities are related to each other by eq. (2.1), which also defines the so-called furnace efficiency η_f :

$$\eta_f \equiv \frac{\dot{Q}_w}{\dot{Q}_0} = 1 - \frac{\dot{M}_0 c_p|_{T_0}^{T_{ex}} (T_{ex} - T_0)}{\dot{Q}_0} \quad (2.1)$$

$\dot{M}_0$ is the mass flow rate of fuel and oxidant and $\dot{Q}_0$ is the total thermal input; i.e., the sum of net chemical heat $\dot{Q}_{0,ch}$ and sensible heat $\dot{Q}_{0,s}$ of the input streams.* T_0 is the ambient temperature. For simplicity, T_0 is also used as the reference temperature for calculating the mean specific heat c_p of the combustion products.

In the following discussion, the most important variables derived from furnace operating and boundary conditions which influence η_f , $\dot{Q}_w$ and T_{ex} are identified. It is also intended to quantify some of these dependencies. The investigation is carried out with the aid of a simple furnace model.

2.1 Well-stirred Analysis

The simplest existing closed furnace model is that derived for a single well-stirred furnace zone with volume V completely enclosed by a single surface zone of area A . Both V and A are characterized by uniform temperatures T and T_w , respectively, and by uniform values for properties describing the radiative and convective transfer; i.e., total emissivity ϵ_t of furnace volume, convective heat transfer coefficient α , etc. A heat balance for the volume zone yields the following relation:

$$\dot{M}_0 c_p|_{T_0}^{T_{ex}} (T_{ex} - T_0) + A\alpha(T - T_w) + A\epsilon_t\sigma(T^4 - T_w^4) - \dot{Q}_0 = 0 \quad (2.2)$$

*The heat necessary to vaporize fuel and air moisture is considered to be already subtracted from $\dot{Q}_{0,ch}$ and $\dot{Q}_{0,s}$, respectively.

If it is assumed, that the furnace is perfectly stirred, then the exit temperature is the same as the characteristic furnace temperature; i.e.,

$$T_{ex} = T. \quad (2.3)$$

In this case, the balance equation (2.2) can be solved for T . Although an analytical solution of (2.2) is possible for fixed values of c_p and ϵ_t , all the results presented in the next section were iteratively obtained with the following easily programmable solution algorithm:

$$T_n = T_{n-1} - \left[\frac{f(T)}{f'(T)} \right]_{n-1}. \quad (2.4)$$

Index n denotes the n 'th iteration step; $f(T)$ and $f'(T)$ represent eq. (2.2) and its derivative, respectively, calculated for the $n-1$ 'th iteration.

After final convergence $T_n = T_{n-1} = T = T_{ex}$, the total heat absorption $\dot{Q}_w$ and the furnace efficiency η_f are determined from eq. (2.1).

Although the system of eqs. (2.1) through (2.3) is sufficient to describe the behavior of a well-stirred furnace, some of the quantities in these equations can be combined to form new variables which are more familiar to the combustion engineer. One of these quantities is the (pseudo) adiabatic flame temperature defined as

$$T_{ad}^* \equiv T_0 + \frac{\dot{Q}_0}{\dot{M}_0 c_p|_{T_0}^{T_{ex}}}. \quad (2.5)$$

T_{ad}^* is referred to as pseudoadiabatic because the mean specific heat c_p in (2.5) is calculated between T_0 and T_{ex} instead between T_0 and the true adiabatic flame temperature T_{ad} . The definition (2.5) has the advantage that parametric studies with the well-stirred model can be carried out avoiding direct computation of c_p . Instead, T_{ad}^* may be described.

In addition to a moderate dependence on the exit temperature, T_{ad}^* itself depends upon the fuel type (i.e., lower heating value, stoichiometric air/fuel ratio, moisture content) and on the operating conditions (air number, fuel and air preheat, air humidity, added moisture). Eq. (2.6) reveals some of these dependencies in more detail:

$$T_{ad}^* = T_0 + \frac{(1-W)H_\ell \left(1 + \frac{\dot{Q}_{0,s}}{\dot{Q}_{0,ch}}\right)}{\left[1 + \lambda \left(\frac{\dot{M}_a}{\dot{M}_f}\right)_{stoich}\right] c_p|_{T_0}^{T_{ex}}}. \quad (2.6)$$

H_l is the lower heating value of the dry fuel in kJ per kg dry fuel. $\dot{M}_a/\dot{M}_f$ stoich represents the dry stoichiometric air/fuel ratio. W is the total moisture content of the input mass flow in $\text{kg}_{\text{H}_2\text{O}}/\text{kg}$. All other quantities in (2.6) have been explained above.

The introduction of T_{ad}^* leads to the following simple relation between η_f and the furnace exit temperature T_{ex} :

$$\eta_f = \frac{T_{ad}^* - T_{ex}}{T_{ad}^* - T_0}, \quad (2.7)$$

in which T_{ex} may be replaced by T in the case of perfect stirring.

In this case, the combination of eq. (2.5) with eqs. (2.2) and (2.3) yields:

$$\dot{Q}_0 \frac{T - T_0}{T_{ad}^* - T_0} + A\alpha(T - T_w) + A\epsilon_t\sigma(T^4 - T_w^4) - \dot{Q}_0 = 0. \quad (2.8)$$

Another quantity familiar to furnace designers is the firing density $\dot{q}_0$ defined by

$$q_0 \equiv \frac{\dot{Q}_0}{A}. \quad (2.9)$$

The quantity will also be occasionally used in the following study. The balance equation is then further reduced to:

$$\dot{q}_0 \frac{T - T_0}{T_{ad}^* - T_0} + \alpha(T - T_w) + \epsilon_t\sigma(T^4 - T_w^4) - \dot{q}_0 = 0. \quad (2.10)$$

The most rigorous reduction of the balance equation (2.2) has been suggested by Hottel and Sarofim (ref. 2). By linearization of the convective term with respect to T^4 according to eq. (2.11)

$$\alpha(T - T_w) + \epsilon_t\sigma(T^4 - T_w^4) \approx \epsilon_t^*\sigma(T^4 - T_w^4) \quad (2.11)$$

and further transformation of eq. (2.10) they obtained:

$$\dot{q}_0^* \eta_f^* + \left(\frac{T_w}{T_{ad}^*}\right)^4 - (1 - \eta_f^*)^4 = 0. \quad (2.12)$$

$\dot{q}_0^*$ and η_f^* are called the reduced firing density and the reduced furnace efficiency, respectively. These quantities have the following definitions:

$$\dot{q}_0^* \equiv \frac{\dot{q}_0}{\epsilon_t^* \sigma T_{ad}^{*3} (T_{ad}^* - T_0)} \quad , \quad (2.13)$$

$$\eta^* \equiv \eta_f \frac{T_{ad}^* - T_0}{T_{ad}^*} = \frac{T_{ad}^* - T_0}{T_{ad}^* - T_0} \frac{T_{ad}^* - T_0}{T_{ad}^*} \quad . \quad (2.14)$$

By introducing $\dot{q}_0^*$ and η_f^* the overall thermal performance of a perfectly stirred combustor is completely described by only three quantities, namely $\dot{q}_0^*$ and the ratio T_w/T_{ad}^* as independent variables and η_f^* as dependent variable. The approach has also been found to be very useful in correlating thermal performance data of furnaces which are not perfectly stirred. However, the composite nature of the new quantities makes the physical understanding of thermal behavior of furnaces more difficult. For this reason, the following parametric study has been carried out with some of the simpler variables introduced earlier, such as η_f , $\dot{Q}_0$ or $\dot{q}_0$, T_{ad}^* , etc; furthermore, it is also worthwhile to consider the still remaining composite variable ϵ_t .

ϵ_t is the constant of proportionality of net radiative exchange between the furnace volume and furnace heat sinks and is also called total furnace emissivity. In order to allow a simple assessment of ϵ_t , it will be assumed, for the present, that radiation from the medium in the furnace and from the furnace walls is gray. Thus, ϵ_t is expressed as

$$\epsilon_t = \frac{1}{\frac{1}{\epsilon_f} + \frac{1}{\epsilon_w} - 1} \quad , \quad (2.15)$$

where ϵ_f represents the emissivity of the combustion products in the furnace volume and ϵ_w is the wall emissivity.

The assumed gray radiation of the combustion products allows the use of a single absorption coefficient K_a to determine ϵ_f :

$$\epsilon_f = 1 - e^{-K_a L_m} \quad . \quad (2.16)$$

L_m in eq. (2.16) is the mean radiation beam length. In a furnace of arbitrary shape, L_m is most suitably approximated by

$$L_m = 3.6 \frac{V}{A} \quad (2.17)$$

A rectangular furnace shape with a height of L and square cross-section of width $L/3$ was assumed in some of the following examples, in which case

$$L_m = 3.6 \frac{L}{14} \quad (2.18)$$

and for the surface area required in eqs. (2.2) or 2.8)

$$A = \frac{14}{9} L^2 \quad (2.19)$$

2.2 Results and Discussion; Parametric Study of a Perfectly Stirred Combustor

The results of this parametric study, which is based on the assumption of gray radiation, are presented in terms of the following independent variables (parameters):

- T_{ad}^* or λ , c_p , etc. (eq. 2.6)
- $\dot{q}_0$, ϵ_f , or $\dot{Q}_0$, K_a , L
- T_w , ϵ_w
- α

Dependent variables are:

- η_f or T_{ex} ($=T$) .

η_f and T_{ex} are linked via the adiabatic furnace temperature T_{ad}^* by eq. (2.7). Figure 2-1 shows how, according to this equation, a change ΔT_{ex} of the exit temperature is related to a change $\Delta \eta_f$ of furnace efficiency. for a typical adiabatic flame temperature of 2200°K, a decrease of 1 percent in the exit temperature corresponds to an increase in furnace heat absorption of 1.15 percent of the total heat input. The dependency shown in Figure 2-1 is also valid in the case of departure from perfect stirring $T \neq T_{ex}$, when T_{ex} is replaced by the temperature $\bar{T}_{ex}$ averaged over the furnace exit.

Figure 2-1. Increase in Furnace Efficiency Depending on Decrease of Furnace Exit Temperature.

Values of the independent variables considered for the baseline case are listed in Table 2-1. Computed furnace efficiencies or exit temperatures obtained by varying these values with respect to the baseline case are displayed in Figures 2-2 through 2-19.

2.2.1 Studies With Prescribed Emissivity of the Furnace Volume

Figure 2-2 shows the impact of the adiabatic flame temperature on overall furnace performance for different firing densities. The dependence is almost linear. An increase of T_{ad}^* by 1 percent yields approximately 0.46 percentage points in furnace efficiency.

As described earlier, the adiabatic flame temperature itself is a function of various quantities, such as excess air ratio, stoichiometric air/fuel ratio, lower heating value, specific heat, etc, resulting in corresponding relations between η_f and these quantities. Variations of η_f with excess air ratio are shown in Figures 2-3 and 2-4. The computations in these figures were carried out for different air/fuel ratios and different heating values, respectively. No preheat and no moisture content of the input streams were considered. The value of mean c_p at exit temperature was chosen to be constant at 1.25 kJ/kg·K, and is an approximation only.

Although there is a certain change in the slope of the $\eta_f = f(\lambda)$ relations, a coarse approximation is that a 10 percent increase in excess air lowers η_f by three percentage points; at least for the baseline data and usual operating conditions ($1.05 < \lambda < 1.30$). In the case of perfect stirring considered here, higher excess air also causes the exit temperature to drop. If the condition of perfect stirring is not fulfilled, it may occur that an increase in excess air is accompanied by an increase in the exit temperature. This depends very much on the distance between the flame zone and the furnace exit (see also section 4.5, Figure 4-12a).

In addition to the adiabatic furnace or flame temperature, the two next important quantities in the thermal behavior of combustors are the firing density and the emissivity of the furnace volume. This is documented in Figures 2-5 and 2-6 which show the $\eta_f(\dot{q}_0)$ functions for ϵ_f as parameter and similarly, the $\eta_f = f(\epsilon_f)$ functions with $\dot{q}_0$ being the parameter. Beyond a firing intensity of 1000 kW/m², a larger gain in η_f can only be obtained by increasing the emissivity of the furnace volume drastically. The finite values of furnace efficiencies at zero emissivities of the furnace volume are due to pure convective heat transfer with the specified α . A general

Table 2-1. Input Data of Basic Case for the Study of Thermal Performance of Perfectly Stirred Combustors

Independent Variables	Symbol	Dimension	Value
Pseudo-Adiabatic Flame Temperature	T_{ad}^*	K	2200
Mean Spec. Heat at Furnace Exit	$c_p _{T_0}^{T_{ex}}$	kJ/kg K	1.25
Lower Heating Value	H_L	kJ/kg	40,000
Stoichiometric Air/Fuel Ratio	$\dot{M}_a/\dot{M}_f$	-	12
Firing Density	$\dot{q}_0$	kW/m ²	300
Emissivity of Furnace Volume	ϵ_f	-	0.3
Thermal Input	$\dot{Q}_0$	MW	500
Gray Absorption Coefficient	K_a	1/m	0.1
Ratio Furnace Height/Width or Depth	$L/(L/3)$	-	3
Temperature of Furnace Walls	T_w	K	298.15
Emissivity of Furnace Walls	ϵ_w	-	1.0
Convective Heat Transfer Coefficient	α	kW/m ² K	0.0116
Ambient Temperature	T_0	K	298.15

Figure 2-2. Dependency of Furnace Efficiency on Adiabatic Flame Temperature for Different Firing Densities.

Figure 2-3. Dependency of Furnace Efficiency on Excess Air for Various Air/Fuel Ratios.

Figure 2-4. Dependency of Furnace Efficiency on Excess Air With Lower Heating Value as Parameter.

Figure 2-5. Dependency of Furnace Efficiency on Firing Density With Emissivity of Furnace Volume as Parameter.

Figure 2-6. Dependency of Furnace Efficiency on Emissivity of Furnace Volume With Firing Density as Parameter.

Figure 2-7. Furnace Efficiency Dependent on Emissivity of Furnace Volume With Wall Temperature as Parameter.

Figure 2-8. Dependency of Furnace Efficiency on Emissivity of Furnace Volume With Wall Emissivity as Parameter and Cold Walls.

Figure 2-9. Dependency of Furnace Efficiency on Emissivity of Furnace Volume With Wall Emissivity as Parameter and Hot Walls.

Figure 2-10. Dependency of Furnace Efficiency on Emissivity of Furnace Walls With Wall Temperature as Parameter and $\epsilon_f = 0.3$.

Figure 2-11. Dependency of Furnace Efficiency on Emissivity of Furnace Volume With Convective Heat Transfer Coefficient as Parameter.

Figure 2-12. Dependency of Furnace Efficiency on Furnace Dimensions With Thermal Input as Parameter.

Figure 2-13. Dependency of Furnace Exit Temperature on Furnace Dimensions for Various Thermal Inputs.

Figure 2-14. Dependency of Furnace Efficiency on Furnace Dimensions for Various Wall Temperatures.

Figure 2-15. Dependency of Furnace Efficiency on Furnace Height With Wall Emissivity as Parameter.

Figure 2-16. Dependency of Necessary Furnace Height on Wall Emissivity With Furnace Efficiency as Parameter.

Figure 2-17. Dependency of Gray Absorption Coefficient on Furnace Emissivity for Various Furnace Dimensions.

Figure 2-18. Dependency of Furnace Efficiency on Furnace Dimension With Gray Absorption Coefficient as Parameter.

Figure 2-19. Dependency of Furnace Dimensions on Gray Absorption Coefficient With Furnace Efficiency as Parameter.

value for the increase of $\Delta\eta_f$ with increase of $\dot{q}_0$ by $\Delta\dot{q}_0$ or ϵ_f by $\Delta\epsilon_f$ cannot be given because of the strong nonlinearity of the functions $\eta_f = f(\dot{q}_0)$ and $\eta_f = f(\epsilon_f)$, especially for moderate values of $\dot{q}_0$ and ϵ_f at which the majority of furnaces are operated.

Wall Temperature

As might be expected from the proportionality of radiative heat transfer with T^4 , the influence of the heat sink wall temperatures of the furnace is negligible for surface temperatures lower than 700°K (Figure 2-7). Above this value, the influence of T_w on η_f increases with increasing emissivity of the furnace volume. This is due to increased absorption of wall radiation by the furnace volume for larger ϵ_f ; however, for high wall temperatures and large volume emissivities, η_f is quite insensitive to changes in ϵ_f itself. This means, for instance, that in large coal-fired boiler combustion chambers with ash covered walls, a precise assessment of emission/absorption characteristics of the combustion products in the furnace volume is not required.

Wall Emissivity

Within the usual range encountered in industrial furnaces and boiler combustion chambers, wall emissivities exhibit only a moderate influence on overall heat transfer (Figures 2-8 and 2-9). Based on reasoning similar to that mentioned above for T_w , the influence of ϵ_w increases with increasing opacity of the furnace volume. However, as documented in the figures and also in Figure 2-10, a change in ϵ_w is almost negligible for values of ϵ_f in the range of 0.3 and less. For $\epsilon_f = 0.3$, lowering the wall emissivity by 0.1 lowers the furnace efficiency by only 0.7 percentage points. This behavior is not changed by altering the wall temperature (see Figure 2-10).

Convection

A more accurate estimation of the convective heat transfer coefficient α is only necessary in optically thin cases ($\epsilon_f < 0.3$). This can be concluded from Figure 2-11 where the $\eta_f = f(\epsilon_f)$ relations with parameter α are shown. Notably, that convective transport coefficients for smooth walls generally do not significantly exceed the value of $\alpha = 0.0116 \text{ kW/m}^2 \text{ }^\circ\text{K}$. The larger values of α cited in Figure 2-11 are applicable in the case of

impinging flames or for heat sinks (tubes) fully exposed to the furnace flow. An increase in the overall α , by which convection to the heat sinks can be characterized, diminishes the relative importance of ϵ_f on thermal performance.

2.2.2 Studies for Prescribed Furnace Dimensions and Gray Absorption Coefficients

For a given thermal input, the firing density depends on the furnace surface area. Since the emissivity of the furnace volume is also a function of the furnace geometry, it is useful to link these geometrical effects and show their combined effect on η_f . For this purpose, the gray absorption coefficient, K_a , is prescribed and a rectangular furnace shape is assumed, with a square cross-section and a furnace height, L , three times the furnace width. According to eqs. (2.16) through (2.19), the only remaining geometrical variable in this case is L . Figures 2-12 through 2-19 show some of the relations between L and the overall thermal performance. The input conditions of the baseline case may again be taken from Table 2-1.

Thermal Input as Parameter

Figure 2-12 and Figure 2-13 show the $\eta_f = f(L)$ and the $T_{ex} = f(L)$ relations, respectively, with the thermal input $\dot{Q}_0$ as parameter. The computations have been carried out with a gray absorption coefficient of $K_a = 0.1 \text{ l/m}$ which may be considered as typical for smaller gas-fired or larger oil- or coal-fired furnaces. By comparing Figure 2-12 with previous figures, it can be seen that increasing the furnace dimension is the most effective (yet, also most expensive) way to augment furnace heat absorption. However, for higher loads, to gain the same percentage points in efficiency requires a relatively larger size expansion. This is due to the reduction of η_f with increasing firing density. However, the effect is somewhat compensated by larger ϵ_f 's corresponding to the expansion in size.

In close relation to the $\eta_f = f(L)$ curves, furnace exit temperatures, which by definition approach the adiabatic flame temperature for a diminishing furnace length, substantially drop with increase in size (Figure 2-13). However, for loads above 1000 MW and furnace exit temperatures of 1500°K , which are typical for modern boilers, relatively large sizes of combustion chamber are required ($L > 35\text{m}$). The size of these large boiler combustion chambers must be further increased; first, for adiabatic flame temperatures lower than those considered in Figures 2-12 and 2-13 (compare Figure 2-2);

and, secondly, if the assumption of clean walls, on which these figures are based, is not fulfilled. Significant reduction of the adiabatic flame temperature is encountered especially for coals with high moisture content (see section 4.5). Coal firing will also cause slagging and fouling accompanied by deposits on the heat sink walls. The deposits reduce the heat transfer by action of dramatically increased temperatures of the radiating surfaces and, usually, also by lowering the emissivity of these surfaces.

Surface Temperature as Parameter

The dependency of η_f on L with temperature T_w of the radiating surface as parameter is shown in Figure 2-14. The computation was carried out for a thermal input of 500 MW. Corresponding to Figure 2-7, the effect of a change in surface temperature on η_f is only significant for a temperature T_w larger than 700°K ; however, in order to maintain, for instance, the same furnace efficiency of 0.40 ($T_{ex} = 1440^\circ\text{K}$) obtained for $T_w = 700^\circ\text{K}$ at surface temperatures of 1300°K , which can easily be reached by deposit layers, (see chapter 5), it is necessary to increase L by 16.5m or by 53.2 percent of the original height corresponding to clean walls. The impact of wall temperature on efficiency will be even stronger for adiabatic flame temperatures lower than 2200°K considered here. In the limit, or in furnaces with refractory walls, adiabatic flame temperatures approach each other at $\eta_f = 0$. It is also interesting to note that at low efficiencies; i.e., in smaller furnaces operated with the same thermal input, the presence of wall deposits requires only a moderate increase in size.

Wall Emissivity

A reduction of wall emissivities from 1 (for clean walls) to 0.4 (which is, for instance, the lowest value reported for ash deposits (ref. 4)) causes a smaller drop of η_f or requires a smaller increase in size as observed above for a change from surfaces with 700°K to surfaces with 1300°K (Figure 2-15). Furthermore, and contrary to the nonlinear behavior due to a change of surface temperature, the percentage size increase necessary to maintain the same furnace efficiency varies nearly linearly with ϵ_w between $\epsilon_w = 1$ and $\epsilon_w = 0.5$ (Figure 2-16). Beyond $\epsilon_w = 0.2$, a considerable nonlinear increase of L for different η_f is found.

When a deposit layer is formed, temperature size and reduction of wall emissivity occur at the same time. However, the superposition of these effects on η_f is somewhat less than a pure summation; since, by reducing ϵ_f , the net heat flux to the layer is lessened, thus retarding the increase of surface temperature to a certain extent.

Gray Absorption Coefficient as Parameter

For values of the gray absorption coefficient K_a between 0.05-0.4 1/m which is the range encountered in practical situations (Figure 2-17), there is only a moderate dependency of η_f on K_a (Figure 2-18). A change in the absorption coefficient in the 500 MW furnace of 30m height from 0.05 to 0.1 1/m, which may be due to a change from gas to oil firing and which corresponds to a change in emissivity ϵ_f from 0.32 to 0.53 (see Figure 2-17), increases furnace efficiency by .6 percentage points provided the adiabatic flame temperature and the boundary conditions are not changed. The possible reduction of furnace height would be -16.7 percent. If a change of the absorption coefficients occurs at higher values (for instance, in coal-fired systems) the sensitivity of such a change is even smaller (Figure 2-19). This is, however, again only true for the same T_{ad} and the same wall conditions. On the other hand, in optically thin systems (i.e., smaller gas-fired furnaces), the required furnace size is more sensitive to the exact determination of K_a or ϵ_f with sharp increase of the characteristic curves $L = f(K_a)$ for $K < 0.05$ 1/m.

3.0 THE INFLUENCE OF RADIATIVE PROPERTIES OF COMBUSTION PRODUCTS ON FURNACE HEAT TRANSFER

In the preceding study of overall thermal performance of furnaces, furnace emissivities or absorption coefficients were prescribed instead of computed from fuel properties. In order to assess furnace performance when the fuel is changed, which is the major objective of the current project, ϵ_f or K_a must be at least approximately related to fuel properties and operating conditions. In the following, the simplest approaches to achieve this purpose are described. Some of the approximations obtained for ϵ_f or K_a are then used with the analysis of the perfectly stirred combustor to demonstrate the relative importance of the different radiating species. The results of this study are also valuable in assessing the accuracy necessary in the theoretical or experimental determination of radiative properties of the various species.

3.1 Radiating Species and Ad Hoc Modeling of Cloud Emissivities or Absorption Coefficients

The most important species contributing to radiative heat transfer in furnaces fired with fossil fuels are:

- Gaseous Species H_2O and CO_2 and Less Significantly CH_4 and CO
- Soot Particles
- Coal and Char Particles
- Fly Ash Particles

Radiation originating from these species is superposed by radiation of more or less hot furnace walls; whereas gaseous species and soot particle clouds influence the radiative transfer between the furnace volume and heat sinks only by emission and absorption, coal, char, and fly ash particles may also scatter the radiation. However, throughout the studies on which the present report is based scattering has been neglected.

3.1.1 Gaseous Species

Radiation of the gaseous species is essentially nongray and selective. Much effort has been spent in the past in order to determine emissivities or absorptivities of gas layers of different species depending on partial pressure, layer thickness, temperature, total pressure and overlapping

radiation. Detailed reviews of such work and corresponding emissivity charts have been presented by Hottel and Sarofim (refs. 2 and 3). It can be concluded from their data that it is sufficient to assume that H_2O and CO_2 are the only species contributing to the radiative transfer from furnace gases under practical circumstances. An exemption from this rule may apply for small furnaces fired with low excess air so that CO is found in larger concentrations in considerable parts of the furnace.

The emissivity of the gas volume of the furnace containing H_2O and CO_2 can be determined in several ways. The way most frequently used by furnace designers, is the following:

1. Evaluation of the partial pressures P_{H_2O} and P_{CO_2} (see section 3.2).
2. Determination of a mean radiation beam length L_m of the furnace volume, for instance by eq. (2.17).
3. Estimation of mean radiating temperature T of the furnace volume; $T = T_{ex}$ for perfect stirring!
4. Determination of the individual emissivities ϵ_{H_2O} and ϵ_{CO_2} corresponding to T and to the products $p_{H_2O} L$ and $p_{CO_2} L$, respectively, using the emissivity charts of ref. 2.
5. Correction of ϵ_{H_2O} and ϵ_{CO_2} depending on partial and total pressures, respectively (ref. 2).
6. Determination of the correction factor $\Delta\epsilon_g$ due to overlapping again with charts in ref. 2.
7. Computation of emissivity of the gas volume according to

$$\epsilon_g = \epsilon_{H_2O} + \epsilon_{CO_2} - \Delta\epsilon_g \quad (3.1)$$
8. Repetition of steps 3) through 8) when the mean radiating temperature calculated with ϵ_g in a total heat balance differs from the estimated one.

For cases of pure gas radiation by combustion products, the furnace emissivity ϵ_f used in the preceding parametric study may be replaced by ϵ_g . Similarly, a pseudo-gray absorption coefficient K_a may be derived from ϵ_g according to:

$$K_a = - \frac{\ln(1 - \epsilon_g)}{L_m} \quad (3.2)$$

Thus, all the results presented in section 2.2 can be used for a first assessment of thermal performance of gas-fired furnaces; however, it should be kept in mind, that gas radiation is actually nongray.

Nongrayness of gas radiation may have an impact on radiative transfer, especially when the furnace walls radiate at an elevated temperature ($T < 700^\circ\text{K}$), or when the furnace walls are partially reflecting. The simplest approximation of exchange of nongray radiation between a well-stirred furnace and its surrounding walls is given by eq. (3.3):

$$\dot{Q}_r = \frac{A}{\frac{1}{\epsilon_g} + \frac{1}{\epsilon_w} \frac{\alpha_g}{\epsilon_g} - \frac{\alpha_g}{\epsilon_g}} \sigma \left(T^4 - \frac{\alpha_g}{\epsilon_g} T_w^4 \right). \quad (3.3)$$

This expression must replace the corresponding expression of gray radiative exchange in the balance equation (2.2) when the effect of nongrayness of gas radiation on overall thermal performance shall be studied. α_g is the absorptivity of the furnace volume dependent on the temperature T_w of the furnace walls. An approximate method to determine α_g from the emissivity charts is presented in ref. 2. According to this method α_g is related to ϵ_g calculated at wall temperature and adjusted partial pressures. Thus, the evaluation of α_g includes similar steps listed above for the evaluation of ϵ_g .

The procedure used to obtain ϵ_g and α_g of $\text{H}_2\text{O}/\text{CO}_2$ mixtures from the emissivity charts of ref. 1 is rather tedious although some interpolation formulae exist for these charts (e.g. ref. 5); furthermore, the required accuracy of ϵ_g and α_g values should be considered in the light of other approximations of the heat transfer furnace model. Finally, there is considerable doubt (ref. 6) whether the H_2O emissivity chart of ref. 1 is valid for large values of the product $p_{\text{H}_2\text{O}} L_m$, necessary, for instance, to predict the performance of large boilers.

For the reasons mentioned above, simplified approaches to the determination of ϵ_g and α_g are justified, especially in combination with the simple well-stirred analysis presented in section 2.1 and 2.2. Such an approach, for instance, is the representation of ϵ_g as the sum of weighted contributions of one clear and several gray gas components:

$$\epsilon_g = \sum_{\ell=1}^{n_g} a_{\ell}(T) \left(1 - e^{-k_{g,\ell} \sum p_j L_m} \right). \quad (3.4)$$

The temperature dependence of the weighting factors of the ℓ th gray gas component are usually expressed in polynomial form according to:

$$a_{\ell}(T) = \sum_m b_{m,\ell} T^{m-1} \quad (3.5)$$

Whenever real gas radiation was considered in the present studies, a "one clear/two gray gas" approach (i.e. $\ell = 3$) was used with constants $k_{g,\ell}$ and $b_{m,\ell}$ listed in Table 3-1. These constants were suggested by Johnson and Beer (ref. 7). They are based on a combined emissivity chart $\epsilon_g T = f(p_{H_2O} + p_{CO_2}) L_m$ for a ratio $p_{H_2O}/p_{CO_2} = 1$ presented in ref. 8.

Figure 3-1 shows the relation $\epsilon_g = f(p_{H_2O} + p_{CO_2}) L_m$ with the gas temperature as parameter obtained from the above "one clear/two gray gas" approach with the corresponding constants of Table 3-1. With $\epsilon_f = \epsilon_g$ and neglecting the difference between ϵ_g and α_g this figure, in combination with the well-stirred results of chapter 2.2 permits at least a qualitative assessment of the effect of gas radiation on furnace performance, even when the ratio p_{H_2O}/p_{CO_2} differs from 1 (see chapter 3.3). For a more accurate assessment of gas radiation, a "one-clear/three-gray approach" with sets of constants determined in ref. 6 may be applied. In this reference, the constants are determined for $p_{H_2O}/p_{CO_2} = 1$ and $p_{H_2O}/p_{CO_2} = 2$ and for both ϵ_g and α_g . The computation of the constants is based on an exponential wideband model which, according to the authors of ref. 6 yields better values of the H_2O emissivities at larger $p_{H_2O} L_m$.

When using a weighted gray gas approach the difference between α_g and ϵ_g due to nonequilibrium may be approximated in the simplest manner by equating the absorptivity α_g to the emissivity ϵ_g at the radiation source; i.e., at the wall temperature T_w of the well-stirred furnace.

$$\alpha_g \approx \sum_{\ell} a_{\ell}(T_w) \left(1 - e^{-k_{g,\ell} \sum_j p_j L_m} \right) \quad (3.6)$$

Thus, the same constants $k_{g,\ell}$ and $b_{m,\ell}$ can be used for both α_g and ϵ_g . However, it is then recommended to compute the net radiative exchange between the well-stirred furnace volume and the walls from eq. (3.7).

Table 3-1. Constants of Three-Weighted-Gray-Gas
Approach for Combined Gas and Soot
Radiation (ref. 7)

ℓ -	$b_{1,\ell}$ -	$b_{2,\ell}$ 1/K	$k_{g,\ell}$ 1/m · bar	$k_{s,\ell}$ 1/m kg/m ³
1	0.130	+0.000265	0	3460
2	0.595	-0.000150	0.824	960
3	0.275	-0.000115	25.907	960

Figure 3-1. Emissivity of Gaseous Combustion Products Obtained With "One-Clear/Two-Gray"-Gas Approach (ref. 7).

$$\dot{Q}_r = A\sigma \sum_{\ell=2}^{n_g} \frac{1}{\frac{1}{\epsilon_{g,\ell}} + \frac{1}{\epsilon_w} - 1} \left[a_{\ell}(T) T^4 - a_{\ell}(T_w) T_w^4 \right]$$

$$\text{with } \epsilon_{g,\ell} = 1 - e^{-k_{g,\ell} \sum p_j L_m} \quad (3.7)$$

Compared to eq. (3.3) equation (3.7) has the advantage that the absorption of reflected gas radiation by the furnace gases themselves is adequately handled. Eqs. (3.6) and (3.7) are only valid for the single zone well-stirred model. However, it should be pointed out that the more rigorous multizone furnace model applied later (chapter 4.0) utilizes a similar approach to account for the difference between ϵ_g and α_g .

3.1.2 Soot Particles

Depending on fuel and local combustion conditions, soot particles (small solid particles of high C/H ratio) are often formed in the gas phase of burning hydrocarbons. Typical mean diameters of the spherical, nonagglomerated soot particles range between $30 \cdot 10^{-9}$ and $65 \cdot 10^{-9}$ m. Soot particles are responsible for the yellow luminosity of flames and can contribute significantly to the radiative heat transfer, even when the mass concentrations at which they are present in flames is small. This is due to the large specific surface area (m^2/kg) of a soot particle cloud and to the high number density of the particles. With the optical parameter

$$p = \frac{\pi X}{\lambda} < 1 \quad , \quad (3.8)$$

extinction in a cloud of soot particles does not obey the laws of geometrical optics; rather, the so-called Mie theory has to be applied. Fortunately, soot particles are so small, that the Mie equations for particle absorption efficiency can be approximated by a simple expression (see for instance ref. 2), which is valid for the so-called Rayleigh regime of scatter. Scattering at soot particles can completely be neglected. By using a mean wavelength of thermal radiation which varies with temperature according to Wien's law, the following simple expression for the emissivity of a layer of soot particles is obtained.

$$\epsilon_s = 1 - e^{-k_s c_s L_m} \quad (3.9)$$

$$\text{with } k_s = c_1 T_s \quad (3.10)$$

c_s represents the mass concentration of soot particles (kg/m^3). k_s is the temperature-dependent specific absorption coefficient which varies linearly with the temperature T_s of the particles according to eq. (3.10). It can be assumed with almost no error that the temperature T_s equals the temperature T of the surrounding combustion gases. Frequently, ϵ_s is also expressed in terms of a volume fraction f_v of the soot particles per unit volume furnace space:

$$\epsilon_s = 1 - e^{-k_s^* f_s f_v L_m} \quad (3.11)$$

However, k and k_s^* are related through the density ρ_s of the soot material:

$$k^* = k/\rho_s = c_1 (T_s/\rho_s) \quad (3.12)$$

Values of the constant c_1 , published in literature, vary between 0.47 and $2.09 \text{ m}^2/\text{kg K}$ (ref. 9). This large variation is due to the uncertainty of ρ_s and the complex refractive index of the soot material. Both parameters are needed to compute c_1 from first principles. Furthermore, ρ_s and n may depend on the fuel type and on locally varying combustion conditions. Based on the work of Johnson and Beer (ref. 7) and of Tietze (ref. 9), a value $c_1 = 1.49 \text{ m}^2/\text{kg}^0\text{K}$ seems to be adequate to describe the radiative behavior of heavy fuel oil flames. Since no conclusive data for soot from other hydrocarbons are known, especially from coal volatiles, the value of c_1 cited above may also tentatively be applied for soots of other fuel types-

Figures 3-2 displays the dependency of ϵ_s on the product $c_s L_m$ with the temperature T as parameter. The curves were obtained from eq. (3.9) with c_1 as recommended above. Contrary to the emissivity of gas layers (Figure 3-1) the emissivity of soot clouds increases with temperature; however, for large optical thicknesses, this temperature dependence becomes weaker. Due to the continuous radiation of a soot cloud, all curves approach the value $\epsilon_s = 1$ for large $c_s L_m$. As mentioned earlier, nongray behavior of radiation in multi-zone furnace models can more conveniently be handled by so-called weighted-gray-gas approaches. Although soot radiation, as represented by eq. (3.9), is

Figure 3-2. Emissivity of Soot Particle Layers.

essentially handled as being gray (however, with a temperature-dependent absorption coefficient), this equation may be approximated by

$$\epsilon_s = \sum_{\ell=1}^{n_g} a_{\ell}(T) \left(1 - e^{-k_{s,\ell} c_s L_m}\right) \quad (3.13)$$

Eq. (3.13), which was at first proposed by Johnson und Beer (ref. 7), has the important advantage that use is made of the same weighting factors a_{ℓ} as for real gas radiation (eqs. 3.4 and 3.5). Thus, the overlapping of gas and soot radiation can be handled more appropriately than by a simple combination of eqs. (3.4) and (3.9). The values of $k_{s,\ell}$ for $n_g = 3$ are taken from ref. 7 and listed in Table 3-1. It can be seen that the absorption coefficient $k_{s,1}$ which corresponds to the clear gas component has a higher value than for the other two gray components. This is reasonable, because the spectrally more important contribution of soot radiation to total radiative transfer in a gas-soot suspension is located in spectral windows of gas radiation. Therefore, the approximation (3.13) also yields a qualitatively correct temperature dependence. However, since the curve fitting of eq. (3.9) can only be carried out with the three coefficients $k_{s,\ell}$, the quantitative agreement between eq. (3.9) and its approximation (3.13) is not very good (Figure 3-2). Future work on soot radiation should start with an improvement of this approximation. It is perhaps feasible, to restore a certain temperature dependency of the absorption coefficients $k_{s,\ell}$, since the multizone furnace model currently developed can easily handle such a dependency.

3.1.3 Coal and Char Particles

The optical parameter p of coal and char particles as well as the p of ash particles dealt with later, is in the range of 1 and larger. Thus, contrary to the behavior of soot, the emissivity of clouds of these particles depend on their size distribution and scattering becomes more important. Larger particulates ($x > 20 \mu\text{m}$) may be treated, at least approximately, by the laws of geometrical optics. This is not the case for smaller particles. Thus, clouds of these finer particles cause, like soot particle clouds, a nongray behavior of radiation.

However, since particle size distribution of pulverized coal usually has a weight mean diameter of 50 μm , the assumption of gray radiation of coal or char particles is often sufficient. Assuming the same density ρ_p for each particle, the emissivity of the particle cloud is represented by:

$$\epsilon_p = 1 - \exp \left[-X_a \left(\frac{3}{2\rho_p} \int_0^1 \frac{dR}{x_p} \right) c_p L_m \right] \quad (3.14)$$

X_a is the efficiency factor for absorption, which is constant according to the assumption of gray radiation. c_p is the concentration of the coal or char particles, respectively; and dR , finally, is the mass fraction of particulates of diameter x_p . The expression in parentheses corresponds to the specific projected area of the particle cloud. The product of this expression and X_a can be identified as a gray absorption coefficient.

$$k_p = X_a \frac{3}{2\rho_p} \int_0^1 \frac{dR}{x_p} \quad (3.15)$$

Values of X_a of the (larger) coal or char particles taken from literature vary between 1 and 0.8 (refs. 1, 10-11). The apparent density ρ_p of the coal particles ranges from 1200 kg/m^3 to 1500 kg/m^3 , char particles have an apparent density between 700 kg/m^3 and 1000 kg/m^3 .

In order to evaluate the integral $\int_0^1 dR/x_p$, knowledge of the size distribution of the particulates is required. The value of the integral is $1/x_{p,0}$ for a monosized distribution with particles of diameter $x_{p,0}$. Polysized distributions are frequently approximated by a so-called Rosin-Rammler relation:

$$R = \exp - \left[\Gamma \left(\frac{1}{m} + 1 \right) \left(\frac{x}{x_p} \right)^m \right] \quad (3.16)$$

In this equation, R is the mass-fraction of particles with a diameter larger than x_p , $\bar{x}_p$ is the weight mean diameter, m is the so-called dispersion parameter of the size distribution, and $\Gamma \left(\frac{1}{m} + 1 \right)$ the gamma function of the expression $\frac{1}{m} + 1$. The combination of eqs. (3.14) and (3.15) with (3.16) yields:

$$\epsilon_p = 1 - e^{-k_p c_p L_m} \quad (3.17)$$

with a gray cloud absorption coefficient

$$k_p = X_a \frac{3}{2\rho_p \bar{x}_p} \frac{\pi}{m \sin(\pi/m)} \quad (3.18)$$

Figure 3.3 shows the dependency of cloud emissivity on the product of concentration and layer thickness for various values of the dispersion parameter. The plotted curves were computed from eqs. (3.17) and (3.18) using $X_a = 1$, $\rho_p = 1300 \text{ kg/m}^3$, and $\bar{x}_p = 50 \cdot 10^{-6} \text{ m}$. Figure 3-3 and eq. (3.18) in general, should only be used for a first assessment of the cloud absorption coefficient, since they are based on a pure Rosin Rammler distribution. By using such a distribution, the influence of the smallest size fractions on ϵ_p or k_p is overestimated, especially when m approaches 1. For a more accurate evaluation, it is therefore recommended to solve the integral in eq. (3.14) numerically on the basis of the actual size distribution, at least for the smaller size fractions.

3.1.4 Fly Ash Particles

The theoretical treatment of radiation from fly ash particles is difficult, mainly because efficiency factors for absorption and size distribution of fly ashes seem to depend on the fuel type and combustion conditions and vary over a wide range. The absorption index of the ash material may also be strongly dependent on radiation wavelength (refs. 2, 12). Furthermore, a considerable fraction of the ash particles is smaller than $10 \text{ }\mu\text{m}$, so that the efficiency factor for absorption X_a will depend on the particle size. A more systematic study of the X_a dependence on fuel and firing conditions is one research subject in the current project. Since conclusive experimental results are not yet available, the present screening studies treat ash radiation (excluding radiation from submicron particles which will be discussed in section 3.1.5) as gray. Thus, the emissivity of the fly ash particle cloud is expressed in analogy to eqs. (3.14) and (3.15):

$$\epsilon_a = 1 - e^{-k_a c_a L_m} \quad (3.19)$$

$$\text{with } k_a = X_a \left(\frac{3}{2\rho_a} \int_2^1 \frac{dR}{x_a} \right) \quad (3.20)$$

Figure 3-3. Emissivity of Char Particle Layers for Various Dispersion Parameters of Rosin-Rammler Size Distribution (Based on Same Spec. Surface Area as Parent Coal).

According to ref. 3, $\rho_a = 2200 \text{ kg/m}^3$ and $X_a = 0.2$ may be considered as representative values. However, it should be kept in mind that the latter value is very tentative.

In-situ size distributions of ash particles depend not only on coal type, composition of the minerals and combustion conditions, but also on the distribution of ash minerals within the coal particles, primarily whether this is included or excluded mineral matter (ref. 13). During combustion, the coal particles including mineral matter tend to form droplets of melted material at the char surface which agglomerate after further combustion of the char skeleton. One char particle may form several ash particles mainly due to detachment of the droplets from the surface or due to fragmentation of the parent char particle. The number n_a of ash particles formed per coal particle is found to vary between 1 and 34 (refs. 3,13). Provided mineral matter is present as inherent inclusion according to an ash content, a , of the coal, the diameter x_a of the n_a formed ash particles is related to the diameter x_p of the parent coal particle in the following way:

$$\frac{x_a}{x_p} = \left(\frac{a \rho_p}{n_a \rho_a} \right)^{1/3} \quad (3.21)$$

With ρ_p , ρ_a , n_a , and a , independent of particle size and assuming only inherent ash, the ash will obey the same size distribution as the parent coal and equation (3.21) is also valid for the mean diameters $\bar{x}_p$ and $\bar{x}_a$.

For example, a coal with $a = 0.064$, $\rho_p = 1300 \text{ kg/m}^3$, $\rho_a = 2200 \text{ kg/m}^3$ and $\bar{x}_p = 50 \cdot 10^{-6} \text{ m}$ would yield mean diameters $\bar{x}_p$ of the ash particles of about $6 \text{ } \mu\text{m}$ and $17 \text{ } \mu\text{m}$ with the assumptions $n_a = 20$, and $n_a = 1$, respectively.

Relations $\epsilon = f(c_a L_m)$ have been computed from eqs. (3.19) and (3.20) for layers of fly ash particles with these mean diameters assuming a Rosin-Rammler distribution with $m = 1.5$ (Figure 3-4, curves a and b). Curve c in Figure 3-4 was obtained assuming only external ash of the same mean diameter and size distribution as the original coal. Curves a and curves c are supposed to represent approximately the limiting cases; thus, unless better data are available, radiative heat transfer calculations in fly ash clouds will mainly be based on the approach with $n_a = 1$. Similar to coal

Figure 3-4. Emissivity of Ash Particle Layers for Various Mean Diameters of Particles.

and char particles, it is also recommended to evaluate the absorption coefficient of the ash cloud by a numerical integration over the actual size distribution rather than over a Rosin-Rammler distribution, provided the former is known.

3.1.5 Submicron Particles (Fly Ash)

During the assembling of the experimental equipment used in the present project to study the emission/absorption characteristics of fly ash layers, the question arose whether submicron particles, other than soot, would significantly contribute to radiative transfer. This would then decide the choice of the measuring technique of particle sizes. For this purpose, a screening study was carried out (ref. 14), with the result that, similar to the behavior of soot, the emissivity of a layer of submicron fly ash particles can be expressed by

$$\epsilon_{sa} = 1 - e^{-k_{sa} c_{sa} L_m} \quad (3.22)$$

$$\text{with } k_{sa} = c_2 T_{sa} \quad (3.23)$$

T_{sa} , the temperature of the submicron particles, should almost be identical to the temperature of the surrounding gas. $c_2 = 0.021$ was concluded as 'typical' for submicron fly ashes from data found in literature (see ref. 14).

3.2 Assessment of Species Concentration

Depending on circumstances discussed in section 3.3, a reasonable assessment of species concentration may be expected to have the same impact on the accuracy of radiative and total heat transfer predictions as the assessment of the optical properties (specific absorption coefficients) of the species. In the following, the assessment of species concentration is first discussed for the well-stirred case, so that the furnace efficiency diagrams of section 2 and the emissivity charts of section 3.1 can be combined to allow a closed assessment of well-stirred furnace behavior for gaseous, liquid, or solid fuels; thereafter, it will be described how in each case inhomogenous species concentrations within the furnace might be approximated by simple means when multizone models are used.

Gaseous Species

Since most furnaces are operated under fuel-lean conditions, the most adequate assessment of concentration of gaseous species in the well-stirred case is obtained from a stoichiometric calculation assuming complete burnout regardless of fuel type. However, a slight error may then be made for pulverized coal, since (even with large excess air) complete burnout is not guaranteed in every case. Table 3-2 (taken from ref. 3 and reviewed) gives some idea how the concentration of the radiating species of the gaseous combustion products of solid, liquid, and gaseous fuels vary with fuel type and fuel composition. The partial pressures were calculated for 30 percent excess air and complete burnout. The sum $p_{H_2O} + p_{CO_2}$ which is required in the simplified emissivity chart of Figure 3-1 or in eq. (3.6) varies only between 0.192 bar and 0.322 bar in spite of the variety of fuels.

The assumption that H_2O and CO_2 are uniformly distributed in the furnace at values corresponding to complete burnout can also be used in multi-zone models as a sufficient approximation, since the actual flame zones are usually small compared to the rest of the furnace volume which is often occupied by recirculation zones. This has been proved in a recent verification of the multizone heat transfer model with experimental data from a full-scale 145 MW_{th} combustion chamber of a natural gas-fired boiler.

In all cases where more accurate data of the distribution of the radiating gaseous species are required; i.e., in furnaces with extended reaction zones, the species distribution can be computed from the progress of burnout. Burnout is usually prescribed, although in the case of slowly reacting char particles it may also be partially computed from carbon and O_2 balances.

Soot Particles

The screening studies and computations carried out to date have shown that the correct determination of soot concentration is a crucial point for heat transfer predictions of all furnaces fired with luminous flames. Unfortunately, there is no theory in general and no simple theory in particular that can predict soot concentrations in a furnace with an accuracy necessary for a reliable quantitative prediction of thermal performance. Measurements have shown, that soot is very inhomogenously distributed in

Table 3-2. Composition of Selected Fuels, Their Gaseous Combustion Products (ref. 3) and Emissivities, for $L_m = 3.86m$ at $1600^{\circ}K$ and 1 Bar Total Pressure

<u>Gaseous Fuels</u>										
Denomination	Derived From	CO ₂ Vol %	CO %, Dry	H ₂	CH ₄	N ₂	P _{CO₂} Bar	P _{H₂O} Bar	ϵ_g From Charts Ref. 2	ϵ_g From Eq. (3.6) and Table 3-1
Methane	-	0	0	0	100	0	0.074	0.157	0.287	0.276
Lurgi--O ₂	Lignite	33	13	37	16	1	0.143	0.171	0.323	0.315
Lurgi--Air	Lignite	14	16	25	5	40	0.129	0.141	0.301	0.296
Koppers--Totzek	Bit. Coal	7	56	35	0	2	0.185	0.115	0.287	0.309
Winkler--O ₂	Lignite	20	35	40	3	2	0.173	0.149	0.315	0.319
Winkler--Air	Lignite	10	22	12	1	55	0.162	0.083	0.261	0.283
Wellman--Galusha--O ₂	Bit. Coal	12	52	33	1	2	0.193	0.116	0.289	0.313
Wellman--Galusha Air	Char	3	29	15	3	50	0.138	0.095	0.266	0.277
Carbon Monoxide		0	100	0	0	0	0.275	0.012	0.179	(0.304) ⁽³⁾
Hydrogen		0	0	100	0	0	0.000	0.287	0.285	0.304
<u>Liquid and Solid Fuels</u>										
Denomination	C	H	O	S	N	Ash	Moi Wt %			
Weight %, Dry										
Nr. 6 Heavy Fuel Oil	85.7	10.5	0.52	2.8	0.4	0.08	-	0.113	0.090	0.243
Solvent Refined Coal	86.8	7.0	3.8	0.7	1.3	0.4	-	0.126	0.069	0.237
Texas Lignite	64.1	4.6	18.3	0.8	1.2	11.0	26.5	0.132	0.113	0.285
Wyoming--Subbit. C Coal	61.7	4.1	14.6	3.4	1.3	14.9	22.3	0.131	0.100	0.273
Illinois--Subbit. B Coal	74.6	4.8	8.9	1.8	1.5	8.4	8.6	0.133	0.071	0.240
Penn.--High Vol. Bit. Coal	79.5	5.2	6.1	1.3	1.4	6.5	1.4	0.133	0.062	0.238

- (1) Computed for 30% excess air, half saturated with water at $289^{\circ}K$, gaseous fuels saturated, solid fuel H₂O as received.
- (2) Small concentrations of SO₂ were added to those of CO₂.
- (3) Coefficients in Table 3-1 only valid for $0.5 \leq P_{H_2O}/P_{CO_2} \leq 2$.

the flame zones. Furthermore, a certain small amount of unburnt soot may also be found outside the flame zones. The screening studies (see also section 3.3) have shown that this "background" soot plays a significant role for heat transfer in larger combustion chambers ($L_m > 5m$) fired with heavy fuel oil or coal.

A simple rational approach for the assessment of soot concentration, based on fuel and furnace properties, was suggested by Sarofim and Hottel (ref. 3). In this approach, the percentage of c of the gaseous hydrocarbons of a fuel is prescribed which is initially converted into soot; however, the actual soot concentration is obtained by considering that a certain amount of the initially formed soot is burned depending on the ratio of residence-to-mixing time.

In the well-stirred case, the final soot concentration is expressed by equation (3.24):

$$c_s = m_c \cdot m_{c,vol} \cdot m_{c,vol,s} \cdot m_{c,vol,s,m} \cdot \rho_m \left(\dot{M}_f / \dot{M}_o \right) \quad (3.24)$$

The quantities in (3.24) have the following meaning:

c_s soot concentration (kg/m^3)

ρ_m density of combustion products (kg/m^3)

$\dot{M}_f$ fuel flow rate (kg/s)

$\dot{M}_o$ total mass flow rate (kg/s)

m_c amount of carbon in fuel (kg/kg)

$m_{c,vol}$ ratio of fixed carbon in volatiles to total carbon in fuel (kg/kg)

$m_{c,vol,s}$ amount of carbon of volatiles, which initially forms soot (kg/kg)

$m_{c,vol,s,m}$ amount of soot which is not burnt in furnace volume and therefore capable to emit luminous radiation.

m_c is taken from the ultimate analysis of the fuel. $m_{c,vol}$ is calculated according to eq. (3.25).

$m_{c,fix}$ is the amount of fixed carbon determined by the proximate analysis. $m_{c,vol,s}$ depends on fuel type and combustion aerodynamics with values in the range from 0 and 0.2. Finally, $m_{c,vol,s,m}$ is evaluated from a well-stirred analysis resulting in the simple expression:

$$m_{c,vol,s,m} = \frac{1}{1 + t_r/t_b} \quad , \quad (3.26)$$

where t_r/t_b is the ratio of mean residence time to burning time of the soot in the furnace volume. In turbulent flow, which is encountered in most industrial furnaces, burnout of soot is controlled by mixing. In this case, values of t_r/t_b lie approximately in the range between 10 and 20 (ref. 3).

In order to determine the soot concentration in the luminous flame zones of the multizone furnace model, a similar approach as above was chosen. However, since the residence time of the flame zone is smaller than that of the whole furnace volume, the ratio t_r/t_b is reduced by V_{fl}/V_f . When "background" soot is considered in the rest of the furnace, its concentration is computed from

$$c_{s,b} = m_{s,fl} \cdot \frac{1}{1 + t_r/t_b} \rho_m \dot{M}_f / \dot{M}_o \quad , \quad (3.27)$$

where $m_{s,fl}$ is the soot concentration in the flame zone. The ratio t_r/t_b in this case is of the same order of magnitude as mentioned above for the well-stirred analysis of the whole furnace (i.e., between 10 and 20).

A key value in the above analysis is $m_{c,vol,s}$, the amount of carbon from gaseous hydrocarbons converted to soot. Although values of this quantity are based on pure guesses, it was found that those guesses which yielded quantitatively correct predictions of thermal performance were based on soot concentrations at levels which did not contradict experimental observations (ref. 15)

Coal and Char Particles

The concentration c_p of coal and char particles depends, like the soot concentration, strongly on fuel and furnace type and on the firing conditions. A first, very approximate assessment can be obtained in the well-stirred case by assuming uniform mass concentration based on a burnout

either prescribed or computed from a well-stirred approach similar to eq. (3.24). The progress of burnout causes changes in the apparent density of the particles $\bar{\rho}_p$ and in the size distribution R . However, it is assumed in the present study that these changes compensate each other so that the initial specific surface area of the particle cloud is maintained.

An approach for a more detailed assessment of char concentration was recently built into the multizone furnace model. The approach is based on C and O_2 mass balances of each zone using prescribed flow patterns and a prescribed heat release pattern for the gaseous fuel components. Carbon burnout is calculated taking chemical or diffusional control of the burning rate into account. The solution of the C and O_2 balances is coupled with the solution of the zonal heat balances. This upgraded model is still in a test phase, thus results cannot yet be presented in the current report.

Ash Particles

The uniform ash concentration, in the case of the well-stirred furnace, can be directly computed from the ash content of the coal and from furnace input conditions.

Neglecting inhomogeneities in the flame zone due to incomplete mixing and burnout, and neglecting segregation, uniform ash concentrations will also be assumed in the case of the multizone models. A certain amount of ash may vaporize and form submicron particles after condensation. An accurate estimation of this quantity is not possible, and depends in a complicated way on the properties of the mineral matter and on the burning conditions. However, based on measurements in a coal-fired utility boiler (ref. 16), one can conclude that about 1 ÷ 2 percent of the coal ash forms submicron particles. For a first assessment of the impact on radiative transfer, one can also assume that these particles are distributed uniformly in the furnace.

3.3 Sensitivity Study on the Impact of Species Type and Radiative Properties on Overall Heat Transfer in Furnaces

The results of the well-stirred modeling approach for overall furnace performance presented in section 2.2, the emissivity relations of section 3.1, and the recommendations given in section 3.2 for the assessment

of species concentrations, are used in the following to demonstrate the impact of the different radiating species on overall furnace heat transfer. It will be shown, also, how sensitive the predictions are with respect to uncertainties in the determination of either optical properties or concentrations of the radiating species. The study is basically carried out for five different cases; four cases deal with a 5 x 5 x 15 m combustion chamber with mean radiation beam length of 3.8571m fired with methane, heavy fuel oil, and a low- or a high-ash coal. In order to demonstrate the effect of "background" soot, a large combustion chamber, namely, a chamber with 10 x 10 x 30 m dimension and a mean radiation beam length of 7.7143m fired with heavy fuel oil is additionally considered. The input conditions for the investigated cases are listed in Table 3-3, and the properties of the fuels studied are displayed in Table 3-4. The most important parameters used for modeling radiation from the different species are given in Table 3-5. Although more sophisticated approaches could be applied, a simple gray-gas approach is used to calculate the combined emissivity ϵ_f of the mixture of the combustion products:

$$\epsilon_f = 1 - (1 - \epsilon_g) (1 - \epsilon_{\text{soot}}) (1 - \epsilon_{\text{char}}) (1 - \epsilon_{\text{ash}}) \quad (3.28)$$

The results of the sensitivity study are presented in Tables 3-6 through 3-10.

Table 3-6 shows concentrations and emissivities computed for the various radiating species considered in the five test cases. Corresponding values of the product between mean beam length and concentration are also displayed in the emissivity charts, Figures 3-1 through 3-4. With overall emissivities obtained from eq. (3.28) furnace exit temperatures T_{ex}^* and furnace efficiencies η_f^* are evaluated. These quantities represent the base line in each case. The sensitivity of furnace performance with respect to concentration is demonstrated by increasing the concentration of each radiating species individually by 10 percent and repeating the well-stirred analysis. This yields finally, as a key quantity, the percentage change $100 \cdot \Delta\eta/\eta_f^{*1}$ of total heat absorption due to a 10 percent concentration

¹For convenience, the factor 100 is omitted in Tables 3-6 through 3-10.

*Denotes the baseline case.

Table 3-3. Input Conditions of Combustion Chambers
Investigated in the Sensitivity Study

		Case I	Case II	Case III	Case IV	Case V
		Methane	Oil Nr.6	Oil Nr.6	Low Ash Coal MVD	High Ash Coal HVBB
Furnace Geometry	m	5x5x15	5x5x15	10x10x30	5x5x15	5x5x15
Net Thermal Input	MW	117	117	700	117	117
Excess Air	%	30	30	30	30	30
Air Temperature	$^{\circ}\text{K}$	298	298	298	298	298
Total Mass Flow	kg/s	54.55	53.27	318.71	54.78	58.18
Wall Temperature	$^{\circ}\text{K}$	700	700	700	700	700
Wall Emissivity	-	0.8	0.8	0.8	0.8	0.8
Convective HT Coeff.	$\text{kW/m}^2\text{K}$	0.0116	0.0116	0.0116	0.0116	0.0116
Ambient Temperature	$^{\circ}\text{K}$	298	298	298	298	298
Total Pressure	Bar	1.013	1.013	1.013	1.013	1.013

3-4. Composition and Lower Heating Values of Fuels Investigated
in Sensitivity Study

	C _{fix}	Vol	Moi	Ash	C	H	N	O	S	Ash	Lower Heating Value kJ/kg _{dry}
	kg/kg				kg/kg _{dry}						
Methane	————— CH ₄ Saturated at 289 ⁰ K —————										49,962
Oil Nr. 6	0	0.9920	0	0.0080	0.8570	0.1050	0.004	0.0052	0.0280	0.0080	40,881
Low Ash Coal	0.6502	0.2828	0.0225	0.0445	0.8212	0.0521	0.0179	0.0556	0.0076	0.0456	32,853
High Ash Coal	0.3232	0.2327	0.0689	0.3752	0.4769	0.0340	0.0096	0.665	0.0100	0.4030	19,098

Table 3-5. Assumptions and Parameters for Modeling Properties and Concentrations of Radiative Species in Sensitivity Study With Well-Stirred Approach

Species	Cases	Radiative Properties	Concentration
H ₂ O + CO ₂	All	Eq. (3.4) and Table 3-1	From complete burnout
Soot	II-V II,IV,V III	Eq. (3.9) with $c_1 = 1.49 \text{ m}^2/\text{kg}$	$m_{c,vol} = 0.05 \text{ kg/kg}$ $t_r/t_b = 10$ $t_r/t_b = 80$ ($V_{III}/V_{II} = 8!$)
Char	IV,V	Spec. surface area of original coal with $\bar{x}_p = 50 \text{ }\mu\text{m}$, $m = 1.5$ and $\rho_p = 1300 \text{ kg/m}^3$ $X_a = 1.0$	Mass concentration = 2% of initial unburnt coal
Ash	IV,V	$\bar{x}_{ash} = 16.8 \text{ }\mu\text{m}$, $m = 1.5$ and $\rho_{ash} = 2200 \text{ kg/m}^3$ $X_a = 0.2$	From ash content

Table 3-6. Impact of Concentration of Radiative Species on Overall Thermal Performance of Furnaces Computed From Well-Stirred Analyses for Different Fuels

		CH ₄		Oil Nr. 6			Oil Nr. 6		
		Case I, L _m = 3.86m		Case II, L _m = 3.86m			Case III, L _m = 7.71m		
		Base *)	10% Increase P _{H₂O} + P _{CO₂}	Base *)	10% Increase P _{H₂O} + P _{CO₂}	10% Increase C _{soot}	Base *)	10% Increase P _{H₂O} + P _{CO₂}	10% Increase C _{soot}
P _{CO₂} + P _{H₂O}	Bar	0.2341	0.2575	0.2045	0.2249		0.2045	0.2249	
ε _{gas}	-	0.2883	0.3025	0.2937	0.3066	0.2948	0.3565	0.3690	0.3570
C _{soot}	10 ⁻⁶ kg/m ³	-	-	80		88	9.9		10.9
ε _{soot}	-	-	-	0.4818	0.4818	0.5116	0.1632	0.1632	0.1781
ε _f	-	0.2885	0.3025	0.6362	0.6407	0.6556	0.4615	0.4720	0.4715
T _{ex}	K	1542	1534	1423	1421	1417	1569	1565	1565
ΔT _{ex}	K	-	-8	-	-2	-6		-4	-4
η _f	%	27.61	28.13	38.35	38.46	38.72	29.33	29.58	29.57
Δη _f	%	-	0.52	-	0.11	0.37	-	0.25	0.24
Δη _f /η _f [*]	%	-	1.9	-	0.3	1.0	-	0.85	0.82

Table 3-6. Impact of Concentration of Radiative Species on Overall Thermal Performance of Furnaces Computed From Well-Stirred Analyses for Different Fuels (Continued)

		Low-Ash Coal					High-Ash Coal				
		Case IV, $L_m = 3.86m$					Case V, $L_m = 3.86m$				
		Base ^{*)}	10% Increase $P_{H_2O} + P_{CO_2}$	10% Increase C_{soot}	10% Increase C_{char}	10% Increase C_{ash}	Base ^{*)}	10% Increase $P_{H_2O} + P_{CO_2}$	10% Increase C_{soot}	10% Increase C_{char}	10% Increase C_{ash}
$P_{CO_2} + P_{H_2O}$	Bar	0.1971	0.2168				0.2116	0.2327			
ϵ_{gas}	-	0.2744	0.2876	0.2752	0.2751	0.2749	0.3090	0.3220	0.3091	0.3091	0.3095
C_{soot}	$10^{-6} kg/m^3$	11.2		12.3			17		18.7		
ϵ_{soot}	-	0.0921	0.0921	0.1004	0.0921	0.0921	0.1270	0.1270	0.1364	0.1270	0.1270
C_{char}	$10^{-3} kg/m^3$	0.323			0.355		0.631			0.694	
ϵ_{char}	-	0.0671	0.0671	0.0671	0.0736	0.0671	0.1270	0.1270	0.1270	0.1388	0.1270
C_{ash}	$10^{-3} kg/m^3$	0.720				0.792	11.838				13.022
ϵ_{ash}	-	0.0539	0.0539	0.0539	0.0539	0.0582	0.5919	0.5919	0.5919	0.5919	0.6269
ϵ_f	-	0.4186	0.4291	0.4245	0.4232	0.4218	0.7851	0.7891	0.7874	0.7880	0.8043
T_{ex}	K	1500	1495	1497	1498	1499	1366	1365	1365	1365	1362
ΔT_{ex}	K	-	-5	-3	-2	-1	-	-1	-1	-1	-4
η_f	%	33.77	34.06	33.93	33.89	33.86	38.39	38.44	38.42	38.43	38.63
$\Delta \eta_f$	%	-	0.29	0.16	0.12	0.09	-	0.05	0.03	0.04	0.24
$\Delta \eta_f / \eta_f^*$	%	-	0.86	0.47	0.36	0.26	-	0.13	0.08	0.10	0.63

Table 3-6. Impact of Concentration of Radiative Species on Overall Thermal Performance of Furnaces Computed From Well-Stirred Analyses for Different Fuels (Concluded)

		Low Ash Coal Case IV, $L_m=3.86m$		High-Ash Coal Case V, $L_m=3.86m$	
		Base*)	2% Sub- micron Ash	Base*)	2% Sub- micron Ash
$P_{CO_2 + PH_2O}$	Bar	0.1971		0.2116	
ϵ_{gas}		0.2744	0.2744	0.3090	0.3090
C_{soot}	$10^{-6} kg/m^3$	11.2		17	
ϵ_{soot}	-	0.0921	0.0921	0.1270	0.1270
C_{char}	$10^{-3} kg/m^3$	0.323		0.631	
ϵ_{char}	0	0.0671	0.0671	0.1270	0.1270
C_{ash}	$10^{-3} kg/m^3$	0.720		114838	
ϵ_{ash}	-	0.0539	0.0539	0.5919	0.5919
$C_{sub,ash}$	$10^{-6} kg/m^3$	-	14.4	-	236.8
$\epsilon_{sub,ash}$	-	-	0.0017	-	0.0258
ϵ_f	-	0.4186	0.4195	0.7851	0.7906
T_{ex}	K	1500	1500	1366	1365
ΔT_{ex}	K	-	-	-	-1
η_f	%	33.77	33.79	38.39	38.46
$\Delta \eta_f$	%	-	0.02	-	0.07
$\Delta \eta_f / \eta_f^*$	%	-	0.06	-	0.18

Table 3-7. Impact of Emissivity of Radiative Species on Overall Thermal Performance of Furnaces Computed From Well-Stirred Analyses for Different Fuels

Fuel			Base*)	10% Increase ϵ_{gas}	10% Increase ϵ_{soot}	10% Increase ϵ_{char}	10% Increase ϵ_{ash}	Submicron Fly Ash
CH ₄	η_f	%	27.61	28.67	-	-	-	-
Case I	$\Delta\eta_f/\eta_f^*$	%	-	3.8	-	-	-	-
Heavy Fuel Oil	η_f	%	38.35	38.60	38.94	-	-	-
Case II $L_m = 3.86\text{m}$	$\Delta\eta_f/\eta_f^*$	%	-	0.7	1.5	-	-	-
Heavy Fuel Oil	η_f	%	29.33	30.33	29.59	-	-	-
Case III $L_m = 7.71\text{m}$	$\Delta\eta_f/\eta_f^*$	%	-	3.41	0.89	-	-	-
Low Ash Coal	η_f	%	33.77	34.77	33.95	33.89	33.88	33.79
Case IV $L_m = 3.86\text{m}$	$\Delta\eta_f/\eta_f^*$	%	-	2.96	0.53	0.36	0.33	0.06
High Ash Coal	η_f	%	38.39	38.51	38.43	38.43	38.80	38.46
Case V $L_m = 3.86\text{m}$	$\Delta\eta_f/\eta_f^*$	%	-	0.31	0.10	0.11	1.06	0.18

Table 3-8. Estimated Uncertainty in Determination of Properties of Radiative Species

	Gaseous Species	Soot $L_m=3.86m$	'Background' Soot $L_m=7.71m$	Char	Ash
Uncertainty in Concentration	10%	100%	200%	100%	50%
Uncertainty in Emissivity	20%	100%	100%	50%	200%

Table 3-9. Projected Uncertainty of Furnace Efficiency and Total Heat Absorption Due to Uncertainty in Evaluation of Concentration of Radiative Species

Fuel			Gaseous Species	Soot	Char	Ash
CH ₄	$\Delta\eta_f$	%	0.5	-	-	-
Case I L _m = 3.86m	$\Delta\eta_f/\eta_f^*$	%	1.9	-	-	-
Heavy Fuel Oil	$\Delta\eta_f$	%	0.11	3.7	-	-
Case II L _m = 3.86m	$\Delta\eta_f/\eta_f^*$	%	0.3	10.0	-	-
Heavy Fuel Oil	$\Delta\eta_f$		0.25	4.8	-	-
Case III L _m = 7.71m	$\Delta\eta_f/\eta_f^*$	%	0.85	16.4	-	-
Low Ash Coal	$\Delta\eta_f$	%	0.3	1.6	1.2	0.5
Case IV L _m = 3.86m	$\Delta\eta_f/\eta_f^*$	%	0.9	4.7	3.6	1.3
High Ash Coal	$\Delta\eta_f$	%	0.05	0.3	0.4	1.2
Gas V L _m = 3.86m	$\Delta\eta_f/\eta_f^*$	%	0.13	0.8	1.0	3.0

Table 3-10. Projected Uncertainty of Furnace Efficiency and Total Heat Absorption Due to Uncertainty in Evaluation of Emissivity of Radiative Species

Fuel			Gaseous Species	Soot	Char	Ash
CH ₄	$\Delta\eta_f$	%	2.0	-	-	-
Case I L _m = 3.86m	$\Delta\eta_f/\eta_f^*$	%	7.6	-	-	-
Heavy Fuel Oil	$\Delta\eta_f$	%	0.5	5.9	-	-
Case II L _m = 3.86m	$\Delta\eta_f/\eta_f^*$	%	1.4	15.0	-	-
Heavy Fuel Oil	$\Delta\eta_f$	%	1.9	2.6	-	-
Case III L _m = 7.71m	$\Delta\eta_f/\eta_f^*$	%	6.8	8.9	-	-
Low Ash Coal	$\Delta\eta$	%	2.0	1.8	1.2	2.2
Case IV L _m = 3.86m	$\Delta\eta_f/\eta_f^*$	%	6.0	5.3	3.6	6.6
High Ash Coal	$\Delta\eta$	%	0.2	0.4	0.2	8.1
Case V L _m = 3.86m	$\Delta\eta_f/\eta_f^*$	%	0.6	1.0	0.6	21.2

change of each species. A similar study was carried out to show the effect of a 10 percent change in species emissivities on total heat absorption (Table 3-7). The results contained in Tables 3-6 and 3-7 can be summarized as follows:

- The gaseous radiating species are the main contributors to total heat transfer in gas-fired furnaces, in larger oil-fired furnaces, and, surprisingly, also in furnaces fired with coal of low ash content.
- The contribution of soot to total heat transfer is more important in smaller furnaces. However, the influence of background soot in larger units should not be neglected even at concentrations as small as 10 mg/m^3 .
- In coal-fired furnaces, the amount of radiation from soot and char is similar (see also ref. 17); however, their contribution is almost negligible for coals with high ash content.
- Radiation by ash particles seems to be the dominant mode of heat transfer in furnaces fired with the high ash coals.
- Radiation of submicron fly ash particles can possibly be neglected.

Due to the uncertainties in the determination of appropriate concentration levels and in the evaluation of the radiative properties or emissivities, the above conclusions are only preliminary; however, the present study allows an assessment of the effect of possible uncertainties on η_f .

Table 3-8 shows an estimation of errors which might occur in the determination of concentration and emissivity of radiating species for the various test cases. The percentage error in the estimation of concentration is probably largest for soot in large furnaces due to the small concentration values. The largest errors in the determination of emissivities are expected for ash particles. Tables 3-9 and 3-10 reveal the uncertainties of total heat transfer predictions due to the uncertainty in concentration and emissivity values. The predictions were obtained by a linear projection of results in Tables 3-6 and 3-7, which is sufficient for the current purpose.

With the exception of furnaces fired with gas or high-ash coal, the uncertainty in soot concentration and emissivity is the major source of

error in the prediction of total heat absorption. In the test furnaces fired with gas and low-ash coal, as well as in the large furnace fired with oil, the uncertainty in the approximation of gas emissivity plays also a significant role for the accuracy of η_f . Considerable inaccuracies in total heat transfer predictions of coal-fired furnaces may be due to the uncertainty in radiative properties of ash particles, even for a coal with low ash content (Table 3-7). In the furnace fired with high ash coal, reliable modeling of radiation from ash particles clouds is absolutely crucial.

4.0 IMPACT OF FLOW PATTERN, INHOMOGENEITIES IN HEAT RELEASE, AND EMISSION/ABSORPTION CHARACTERISTICS ON LOCAL AND OVERALL HEAT TRANSFER IN AN IDEALIZED COMBUSTION CHAMBER

4.1 Deviation From Well-Stirred Behavior of Furnaces

The parametric studies of the preceding sections were carried out with, among other simplifications, the assumption that the combustion chambers were perfectly stirred ($T = T_{ex}$). In almost every furnace encountered in practice, more or less significant departures from perfect stirring can be expected. The deviation from perfect stirring is mainly due to:

- Flow pattern with insufficient backmixing by recirculation or turbulent diffusion, even if the heat would be uniformly released over the furnace volume.
- Locally confined heat release or flame zones.

Departure from perfect stirring is associated with an inhomogenous temperature distribution within the furnace, and with an inhomogenous distribution of the radiating species and their optical properties. These inhomogeneities cause a more or less strong curvature of the heat flux distribution at the heat sink surfaces.¹ Since the radiative heat transfer is a strong nonlinear function of temperature, inhomogeneities within the furnace will also change the overall furnace heat absorption.

Besides the well-stirred case, two other limiting cases of overall thermal performance of furnaces are identified:

- Pure plug flow through the furnace and infinitely fast heat release at furnace inlet.
- Retarded heat release with flame extending to the furnace exit and with weak outer recirculation.

Figure 4-1 shows how the (reduced) furnace efficiency η_f^* depends, under these circumstances, on the reduced firing density q_o^* as defined in

¹Even in case of perfect stirring, local heat flux distributions at the furnace walls are not uniform due to the shape influence of the radiating furnace volume (see ref. 1).

$$\eta_f^* = \frac{\eta_f (T_{ad}^* - T_o)}{T_{ad}^*} \quad \dot{q}^* = \frac{\dot{Q}_o}{\epsilon_t^* \cdot A \sigma T_{ad}^{*3} (T_{ad}^* - T_o)}$$

Figure 4-1. Overall Thermal Performance of Furnaces With Departure From Well-Stirred Behavior Obtained for $T_w/\dot{T}_{ad} \approx 0.32$.

section 2.1. The relations shown in the figure were obtained by Hottel and Sarofim for nonluminous combustion (ref. 2).

Pure plug flow with immediate heat release (curve a) yields always the highest furnace efficiency. The efficiency of furnaces fired with long flames and weak outer recirculation (curve c) may result in absorption efficiencies below those corresponding to well-stirred behavior (curve b). However, it was pointed out in ref. 18, that curve c is probably based on a too-high convection heat transfer coefficient and on cold, highly absorbing gases surrounding the flame. Our investigations in a pressurized gasifier confirm the fact that highly absorbing recirculation zones can cause furnace efficiencies to drop below those of the well-stirred case.

The majority of furnaces encountered in practice exhibit, however, an overall thermal performance which lies between that of pure plug flow furnaces and that of well-stirred furnaces, tending towards the latter case. This is demonstrated in Figure 4-1 by correlating results from our multi-zone furnace calculations to the $\eta_f^* = f(\dot{q}_0^*)$ relation. In the case of the oil- and coal-fired furnaces, the correlation is only approximate due to a simplified determination of the contribution of the local, luminous flame zones to the well-stirred emissivity ϵ_t^* .

In order to apply the more convenient well-stirred analysis to combustion chambers which depart from well-stirred behavior ($T \neq T_{ex}$), Sarofim and Hottel (ref. 3) suggest an empirical correction of the radiating furnace temperature T according to:

$$T = a T_{ex} + (1 - a) T_{ad}^* , \quad (4.1)$$

where a is a weighting factor with values between 1 and 0. The above authors found a value $a = 0.75$ satisfactory for a wide range of furnace types and operating conditions. This is in agreement with our own observations (see Figures 4-1 and 4-5).

In spite of the surprising ability of the well-stirred analysis to allow interpretation of furnace performance, quantitatively accurate predictions of overall performance of furnaces, especially of those with local flame zones, can, if at all, only be expected from multizone models. These models are also the only ones which can provide local gas and wall

Figure 4-2. Flow and Heat-Release Patterns Investigated.

Figure 4-3a. Influence of Flow and Heat Release Pattern on Temperature Distribution in a Methane-Fired Boiler Combustion Chamber (One-Gray-Gas Approach).

Figure 4-3b. Influence of Recirculation in Furnace Bottom on Temperature Distribution in a Methane-Fired Combustion Chamber (One-Gray-Gas Approach).

Figure 4-4. Influence of Flow and Heat Release Pattern on Heat Flux Densities at the Side Walls of a Methane-Fired Combustion Chamber (One-Gray-Gas Approach).

Figure 4-5. Furnace Efficiencies of the Methane-Fired Boiler for Different Flow and Heat-Release Pattern.

temperatures and heat flux profiles which are often required for successful furnace design and operation. Parametric studies of local and overall furnace performance carried out with such a multizone model are described in the following sections.

4.2 Short Description of Furnace Heat Transfer Model

The mathematical furnace model used in the study is basically a multizone model. The radiative exchange between the furnace zones is calculated by a so-called semistochastic method. The idea of this method is to distribute the emission power of each zone under a discrete number of radiative beams, taking multiple reflection at furnace walls into account. The beams are traced throughout the furnace volume until final absorption. Contrary to pure Monte Carlo Methods, some events in a beam's history are calculated from geometrical and optical laws rather than determined from random numbers. Thus, the statistical error of heat flux predictions is reduced. The accuracy of the method has intensively been tested (refs. 1,19).

Nongrayness of radiation of the combustion products is modeled with a weighted gray gas approach. Radiating species considered are: H_2O and CO_2 and the particulates soot, char, and ash. Currently, char and ash particles are handled as gray radiators. Details of modeling the radiative properties have been described in section 3.1.

The model of radiative exchange is directly coupled with a total heat balance of zones with unknown temperatures. In balances for volume zones, net convective, diffusive and radiative heat fluxes are equated to the net release of chemical heat. The calculation of the convection heat fluxes is based on total mass flows prescribed at the boundaries of each zone. Similarly, the net zonal heat release is computed from a prescribed fuel flow across a zone. More details of the multizone furnace model can be found in refs. 1 and 19.

The above-described furnace heat transfer model was used to evaluate the influence of variations of heat-release patterns and the change from gas to coal firing on thermal performance of a typical boiler configuration. The calculations were carried out for a fictional medium-sized combustion chamber with 117 MW thermal input. The most important operating and

boundary conditions of the modeled boiler are listed in Table 4-1. The somewhat ideal input data were chosen wherever possible to allow comparison with well-stirred test cases discussed by Sarofim and Hottel (ref. 3).

With the exception of the well-stirred analyses, the predictions were obtained with an arrangement of $2 \times 2 \times 6$ cubic-shaped volume zones of the rectangular combustion chamber. The emission of a volume zone was simulated with 8 cells and 96 beams per cell. Taking symmetry with respect to two cross-sections of the furnace into account, 7,296 beams per gray gas were traced. After convergence of the mean values of temperatures of volume zones, temperature fluctuations were less than 0.5°K assuming one gray gas and less than 3°K when an approach with three weighted gray gases was used.

4.3 Impact of Flow and Heat-Release Pattern on Thermal Performance of a Gas-Fired Boiler

This parametric study was carried out for methane as a gaseous fuel. It was assumed that H_2O and CO_2 are the only radiating species and that their concentration ($p_{\text{H}_2\text{O}} = 0.157 \text{ bar}$, $p_{\text{CO}_2} = 0.074 \text{ bar}$) is uniform throughout the furnace corresponding to complete combustion and to operating conditions listed in Table 4-1. The radiative behavior of the mixture was approximated by one gray gas with a uniform absorption coefficient $K_a = 0.089 \text{ l/m}$. K_a was evaluated from an emissivity of the gas volume of $\epsilon_g = 0.29$ which, in turn, was determined for the mean beam length of the furnace and for the radiating temperature $T = 1631^{\circ}\text{K}$ obtained from the well-stirred analysis in ref. 3. Also according to this reference, the difference between emissivity and absorptivity of real radiation was approximated by diminishing the total radiative interchange between zones by 10 percent.

The various flow and heat-release patterns studied are outlined in Figure 4-2. Three basic systems were investigated: a well-stirred system with uniform heat release throughout the furnace volume, a plug-flow system with the flame zones in the bottom of the furnace and a recirculating plug-flow system with flame zones raised by one zone layer. The well-stirred analysis was carried out with two different assumptions. In case a, radiating and exit temperature of the furnace were assumed to be the same ("perfectly stirred"), in case b the radiating temperature was weighted

Table 4-1. Operating and Boundary Conditions of Modeled
Boiler Combustion Chambers

	Gas-Fired Boiler	Coal-Fired Boiler
Furnace Geometry	5 x 5 x 15 m	
Fuel	CH ₄ (sat'd at 289 K)	HVA Coal (1.4% moisture)
Lower Cal. Value	49961.6 kJ/kg _{dry}	31506 kJ/kg _{dry}
Thermal Input	117 MW	
Excess Air	30%(half sat'd at 289 K)	
Air Temperature	298 K	
Total Mass Flow $\dot{M}_0$	54.55 kg/s	55.50 kg/s
Fuel Mass Flow	2.34 kg _{dry} /s	3.71 kg _{dry} /s
Wall Temperature	700K	
Surface Emissivity	0.8	
Convective HT Coefficient	$\approx 0.0058 \text{ kW/m}^2 \text{ K}$ (see ref.3)	
Ambient Temperature	298 K	

between the exit temperature and a pseudo-adiabatic flame temperature, with weighting factors 0.75 and 0.25, respectively, as suggested by ref. 3. A similar difference was made for the plug-flow system. In case c, all zones were assumed to be perfectly stirred whereas in case d, the heat-release zones in the bottom of the furnace had a higher radiating temperature than exit temperature according to the same weighting procedure as in case b. The recirculating plug system, finally, was investigated for six different rates of recirculation between the heat-release zones and the bottom zones of the furnace (cases e ÷ j).

Volume zone temperatures calculated for the different cases are shown in Figures 4-3 a ÷ b, and some corresponding heat flux distributions at the furnace side walls are plotted in Figure 4-4. Calculations with the well-stirred system (case b) are in agreement with those obtained by Sarofim and Hottel. From Figure 4-3 it can also be seen that empirical correction of the radiating temperature as suggested by ref. 3 yields, in contrast to the computation for perfect stirring (case a), a furnace outlet temperature which is close to that obtained with the more realistic assumption of local heat release and recirculating plug flow patterns. This indicates that, for the type of combustion chamber studied, a simple well-stirred analysis provides a reasonable value for the overall heat extraction (see also Figure 4-5). However, since this is achieved with an empirical correction, the well-stirred approach cannot determine the impact of combustion modification on total heat absorption; moreover, local temperatures and heat fluxes predicted with the more realistic flow patterns differ considerably from the average values produced from the well-stirred analyses.

As expected, a pure plug-flow assumption with early heat release yields the highest temperature peaks and lowest furnace outlet temperatures. A change in the combustion pattern in the flame zone itself, as simulated by case d for the plug-flow system, has only a restricted local influence. This can be seen by comparing case d with case c for a perfectly stirred flame zone. This indicates that an approximate local definition of the flame zones is sufficient for temperature and heat flux predictions for the bulk of the boiler.

An increase of the recirculation strength in the furnace bottom (e.g., by modified burner design or change of operating conditions) loses its effect on heat transfer in the bulk of the furnace for entrainment ratios greater than 1. The case e for zero entrainment is somewhat artificial, and predicted temperatures in the bottom zones of the combustion chamber are very low. This is also due to the fact that turbulent diffusion between furnace zones has been neglected.

Figure 4-5 illustrates the dependency of η_f on the recirculation strength in the cases investigated. Corresponding to the temperature distributions, furnace efficiencies predicted for the more realistic cases of combined recirculating plug-flow pattern and flame zones distant from the furnace bottom, are found between the upper limit for a furnace with pure plug flow and immediate heat release and a lower limit computed for a perfectly stirred furnace with uniform heat release. For ratios $\dot{M}_r/\dot{M}_0 >$, η_f is almost independent of the strength of recirculation.

The foregoing study was carried out with a gray approximation of gas radiation. In order to assess the effect of nongrayness on the results, and in order to allow comparison with nongray predictions for the coal-fired combustion chamber shown in section 4.4, heat transfer in the gas-fired chamber was also computed for one representative case ($\dot{M}_r = 0.5 \dot{M}_0$) with an approach of weighted gray gases as formulated by eqs. (3.4) ÷ (3.5). Values of the constants $b_{m,l}$ and $k_{g,l}$ used in this approach were taken from Table 3-1.

In Figure 4-6, volume zone temperatures calculated with the weighted gray gas approach (curve c) are compared to those obtained earlier for the same flow case by a gray gas analysis and by overall empirical consideration of the difference between emissivity and absorptivity (curve a). Inaccurate interpolation of the emissivity charts with the weighted gray gas approach can be assessed by comparing curve a with curve b. All other conditions being the same, curve a and curve b were obtained with gray absorption coefficients based on emissivities obtained from charts (ref. 2). The main difference in the temperature distributions a and c is due to nongrayness of radiation. The decrease of real gas emissivity with higher temperatures causes a reduction in heat transfer and consequently a higher

Figure 4-6. Comparison of Temperatures Predicted with a "One-Gray" and a "Weighted-Gray" Gas Approach for Methane-Fired Boiler ($\dot{M}_r = 0.5 \dot{M}_0$).

temperature level in the first part of the furnace which is not fully compensated in the cooler part of the combustion chamber. Thus, the furnace efficiency predicted with the more realistic weighted gray gas approach is about 2.2 percent less than that of the gray analysis. Since all other examples of the foregoing study will show a similar trend, the above conclusions concerning the influence of the flow and heat-release pattern are not in question.

4.4 Thermal Performance of a Coal-Fired Boiler Combustion Chamber and Comparison to Gas Firing

The series of test calculations initiated to study the impact of a change from gas to coal firing on thermal performance, was carried out for the same fictional combustion chamber operated under similar conditions to that of the gas-fired boiler (Table 4-1). Two different flow- and heat-release patterns were investigated; namely, the well-stirred case b of Figure 4-2 to allow comparison with ref. 3, and the recirculating plug-flow system with $\dot{M}_r = 0.5 \dot{M}_0$ which were found to be representative for the thermal performance of the gas-fired chamber.

The fuel studied was a high-volatile bituminous coal which also served as a sample fuel in ref. 3. The properties of this coal are listed in Table 4-2.

For both the well-stirred and the recirculating plug-flow system, the concentration of radiative species was modeled in a similar way.

Radiating gaseous species and ash particles were assumed to be uniformly spread over the whole combustion chamber. The corresponding concentration values determined from fuel composition and operating conditions were:

$$p_{H_2O} = 0.062 \text{ bar}, p_{CO_2} = 0.133 \text{ bar}, \text{ and} \\ m_{ash} = 0.0043 \text{ kg/kg.}$$

The presence of char particles and soot--the latter accompanying incomplete combustion of volatiles--was restricted to the heat-release zones. Thus, in the case of the single-zone well-stirred analysis, char and soot were also uniformly spread throughout the furnace volume.

Table 4-2. Composition of HVA Coal Used in the Model

C _{fix}	0.627	kg/kg _{wet}	C	0.850	kg/kg _{daf}
Volatiles	0.295	"	H	0.056	"
Moisture	0.014	"	O	0.065	"
Ash	0.064	"	S	0.014	"
			N	0.015	"

The mass concentration of char particles was supposed to be 1/10 of the unburnt coal for the given mass flows (i.e., assuming 90 percent overall burnout in the heat release zones). This yields a value of $m_{\text{char}} = 0.0068$ kg/kg in both flow systems investigated.¹

Soot concentrations in the heat-release zones are evaluated from the empirical model of formation and burnout (eqs. 3.24-3.26). It is assumed that 10 percent of the volatile carbon corresponding to 2 percent of the coal as fired forms soot. For the selected furnace operating conditions, this corresponds to a concentration of $(m_{\text{soot}})_0 = 1.36 \cdot 10^{-3}$ kg/kg. It is further assumed that the burning time t_b , which is controlled by turbulent mixing, is one-tenth of the residence time t_{res} of the well-stirred furnace. Thus, in the well-stirred case, the final soot concentration is $m_{\text{soot}} = 0.123 \cdot 10^{-3}$ kg/kg. The residence time in the heat-release zones of the recirculating plug-flow system is approximately one-sixth of that of the well-stirred furnace. In this case, a soot concentration of $m_{\text{soot}} = 0.510 \cdot 10^{-3}$ kg/kg is computed from eq. (3.26).

The radiative properties of the suspension in the coal-fired chamber were alternatively approximated with one gray gas or with three weighted gray gases to allow for nongrayness. However, the contribution of ash and char particles was in each case considered to be gray.

Gas and soot absorption coefficients used for the gray approach were determined to match the corresponding emissivities of ref. 3 at mean beam length of the furnace and at well-stirred temperatures. Using the more realistic approach with weighted gray gases, the combined absorption coefficient of the l th gray gas component was calculated from

$$K_{a,l} = k_{g,l} (p_{\text{H}_2\text{O}} + p_{\text{CO}_2}) + k_{s,l} \frac{m_{\text{soot}}}{\rho_m} + k_{\text{char}} \frac{m_{\text{char}}}{\rho_m} + k_{\text{ash}} \frac{m_{\text{ash}}}{\rho_m}. \quad (4.2)$$

The constants used in this approach were taken from Table 3-1; k_{ash} and k_{char} are determined from eqs. (3.15) and (3.20), respectively; k_{ash} was calculated with an absorption efficiency $X_a = 0.2$, particle density $\rho_p = 2200 \text{ kg/m}^3$, weight mean diameter $\bar{x} = 16.8 \text{ } \mu\text{m}$ and with a dispersion factor $m = 1.5$ of a Rosin-Rammler distribution. X_a of the char particles

¹This mass concentration is only used for the computation of radiative transfer. Heat release is based on complete burnout.

is taken as 1 and it is further assumed that the specific surface area of these particles is the same as for the parent coal of $\rho_p = 1300 \text{ kg/m}^3$, $\bar{x} = 50 \text{ }\mu\text{m}$ and $m = 1.5$.

Figure 4-7 shows temperatures of the coal-fired furnace calculated for the well-stirred system and for the recirculating plug-flow pattern with $\dot{M}_r = 0.5 \dot{M}_o$, respectively. These results are based on an approach with one gray gas and the assumption of uniformly distributed species within the furnace. Curves c ÷ d show to what extent the heat transfer is increased when radiation of char, ash and soot particles is consecutively added to the basic gas radiation (curves a).

In Figure 4-8, efficiencies of the coal-fired combustion chamber are plotted for the above test cases and compared to those of the well-stirred analysis of ref. 3. Good agreement is found between both well-stirred analyses. As discussed earlier for the gas-fired boiler, furnace efficiencies of the empirically fitted well-stirred approaches are similar to those calculated with the combined recirculation plug-flow system with $\dot{M}_r = 0.5 \dot{M}_o$.

In the above representation, the influence of char particles is dominant; however, this is caused by the relatively high value of surface area per unit volume adapted from ref. 3 and by the assumption of uniform distribution.

For a pulverized-coal-fired combustion chamber, however, inhomogenous distribution of absorption coefficients due to inhomogenous concentration of particulates and to temperature gradients is more likely. Such effects of inhomogeneity are included in the following example for the recirculating plug-flow pattern $\dot{M}_r = 0.5 \dot{M}_o$. The presence of char and soot is confined to the flame zones and the weighted gray gas approach described above is applied to account for the temperature dependence of radiative properties. Figure 4-9 shows the temperatures of volume zones predicted for this case and Figure 4-10 displays the corresponding net heat flux densities at the side walls of the furnace. The heat transfer peak vis-à-vis the flame zone, which is mainly due to an increase of emissivity by char and soot, is responsible for a considerable decrease of the temperature level in the

Figure 4-7. Temperature Distribution in the Coal-Fired Boiler and Effect of Particulates (One-Gray-Gas Approach)

Figure 4-8. Effect of Particulates on the Furnace Efficiency of Coal-Fired Boiler.

Figure 4-9. Temperature Distribution in a Coal and Gas-Fired Combustion Chamber and Effect of Particulates (Weighted-Gray-Gas Approach; $\dot{M}_r = 0.5 \dot{M}_0$).

Figure 4-10. Heat Flux Distribution at the Side Walls of a Coal and Gas-Fired Boiler Combustion Chamber and Effect of Particulates (Weighted-Gray-Gas Approach; $\dot{M}_r = 0.5 \dot{M}_0$).

whole furnace. This, in turn, causes the heat flux densities from the complete suspension (curve d) to drop in the upper part of the furnace to values below those for pure gas radiation of coal combustion products (curve a). Furnace outlet temperatures are 30°K higher than in the case where uniform distribution of char and soot (curve d, Figure 4-7) is assumed. This difference increased to 105°K when, in the latter case, the soot radiation model of Johnson was applied instead of that of ref. 3. The discrepancy between both models can implicitly be attributed to different assumptions for the apparent density of soot material ($\approx 1000 \text{ kg/m}^3$ in the former, $\approx 2000 \text{ kg/m}^3$ in the latter model). This reflects the current uncertainty in the modeling of radiative properties of particulates. However, the Johnson model, has been verified experimentally and is believed to yield more realistic results. Also shown in Figures 4-9 and 4-10 is a comparison between results for the coal-fired boiler and previously presented results for the methane-fired chamber with the same flow and heat-release pattern and the same weighted gray gas approach. Temperature and heat flux distributions of the methane-fired chamber (curves e) are similar to those of pure gas radiation of the coal-fired chamber (curves a), the gaseous species of the coal combustion products yielding a slightly higher efficiency (28.4 percent compared to 28.3 percent). However, this behavior changes dramatically when particulate radiation in the coal-fired chamber is taken into account. Then, furnace exit temperatures are 125°K lower than in the gas-fired chamber, corresponding to an increased efficiency (37.4 percent compared to 28.3 percent). Peak temperatures in the heat-release zones of the coal- and gas-fired chamber differ by 140°K and predicted peak heat fluxes for coal firing are almost twice as high as those predicted for gas firing.

The lower furnace exit temperature in the case of coal firing could lead to the conclusion that the size of the combustion chamber could be decreased; however, this would be wrong. Generally, coal-fired furnaces have a larger volume than gas-fired furnaces of the same thermal input in order to reduce the danger of slagging and fouling or in order to increase the heat sink area in case of deposits (see also section 5). Large furnace volumes are sometimes also necessary to provide complete burnout.

4.5 Furnace Performance for Different Coal Types

In a further study, the heat transfer behavior of the coal-fired combustion chamber was investigated for five different coal types. The analyses of these coals, which are typical for the U.S., are given in Table 4-3. The planned experimental investigations of emission/absorption characteristics of fly ashes will also be conducted with these fuel types. The input and boundary conditions of the theoretical study correspond to those conditions listed in Table 4-1. For three coals, an excess air factor of $\lambda = 1.1$ was additionally investigated. The computations were based on the same model parameters mentioned in section 4.4. In particular, the mean diameter of the fly ash particles was considered to be the same ($\bar{x}_{\text{ash}} = 16.8 \mu\text{m}$), independent of fuel type and ash content. Thus, differences in thermal performance of the furnace are caused only by the different heating values and different concentrations of radiating species due to coal composition. Figure 4-11a shows that these quantities produce differences in furnace exit temperatures of up to 180°K . However, with the exception of the lignite, which also shows a particular gas temperature profile (curves c and c'), peak heat flux densities differ only by $10\text{--}15 \text{ kW/m}^2$ (Figure 4-11b).

The strong influence of excess air on overall thermal performance of furnaces has already been shown in chapter 2 (see Figures 2-3 and 2-4). Figures 4-12a and 4-12b reveal that excess air also plays a considerable role in determining maximum temperatures and heat fluxes. The decrease of the peak values seems to be independent of coal type. However, it should be kept in mind that in reality soot and char reactions depend on the excess air factor λ . This has been neglected in the current study. It can also be seen from Figure 4-12a that the furnace height or length determines whether furnace exit temperatures will decrease or increase with excess air. In longer furnaces, an increase of the exit temperature will be found with increase of λ .

The difference of local and total heat transfer for the different coals fired at constant λ is mainly due to differences in adiabatic flame temperature and to the varying ash content of the coals. The latter fact can clearly be seen from Figure 4-13 where the furnace efficiency is correlated

Table 4-3. Composition and Lower Calorific Values of Coals Selected for Screening Studies

Denomination Origin Type	C _{fix}	Vol.	Mois.	Ash	C	H	N	O	S	Ash	Lower Cal. Value
	kg/kg				kg/kg _{dry}						kJ/kg _{dry}
Black Creek Alabama #19 MVD	0.6502	0.2828	0.0225	0.0445	0.8212	0.0521	0.0179	0.0556	0.0076	0.0456	32853
Price Utah-II HVBB	0.4492	0.3884	0.0741	0.0883	0.7224	0.0575	0.0155	0.1016	0.0076	0.0954	28577
Beulah North Dakota LA	0.3113	0.2865	0.3310	0.0712	0.6529	0.0396	0.0099	0.1798	0.0114	0.1064	24327
Four Corners #2A HVCB	0.3476	0.3594	0.0524	0.2406	0.5599	0.0471	0.0123	0.1165	0.0103	0.2539	22104
Freeport, Lower Free- port-30 HVBB	0.3232	0.2327	0.0689	0.3752	0.4769	0.0340	0.0096	0.0665	0.0100	0.4030	19098

(a)

(b)

Figure 4-11. Prediction of Temperature (a) and Heat Flux Distribution (b) in a Furnace Fired With Different Coals.

Figure 4-12. Effect of Excess Air on Local Temperature (a) and Heat Flux Distribution (b) in a Coal-Fired Furnace.

Figure 4-13. Dependence of Furnace Efficiency on Ash Content of Coal.

to the ash content of the dry coals. Except for the lignite, furnace efficiency increases with ash content. The lower furnace efficiency of the lignite originates from the very low adiabatic flame temperature due to the high moisture content of this coal.

The high moisture content acts in a similar way as the excess air; namely, by increasing the ratio of total mass flow to stoichiometric dry mass flow. This is demonstrated in Figure 4-14, which shows the dependence of furnace efficiency on the mass flow ratio.

Figure 4-14. Dependence of Furnace Efficiency on the Ratio of Total Mass Flow to Stoichiometric Dry Mass Flow for Different Coals.

5.0 INFLUENCE OF ASH DEPOSITS ON FURNACE HEAT TRANSFER

The screening studies of chapter 4, concerning coal-fired furnaces, have been carried out assuming clean cold walls ($T_w = 700^\circ\text{K}$); however, in almost all coal-fired combustors, ash deposits are likely to be formed at the heat sink surfaces. Depending on the thermal conductivity and the thickness of these deposits, the temperatures of surfaces exposed to furnace radiation will increase. Usually, there will also be a decrease of emissivity of the heat sink surfaces compared to clean conditions. Both increase of surface temperatures and decrease of surface emissivity will reduce the overall furnace heat absorption and alter local temperature and heat flux profiles.

The results of the well-stirred analyses of chapter 2 have shown that the changes in wall temperatures and emissivities due to deposits will reduce furnace efficiency more, the higher the emissivity ϵ_f (see Figures 2-7 through 2-9) or the size (see Figures 2-14 and 2-15) of the furnace volume. In particular, this means that the thermal performance of large combustion chambers will be very sensitive to any build-up of deposits at the heat sink surfaces. Projected from the preceding analyses, wall deposits in these larger furnaces will probably cause a drop in η_f of the same or even larger magnitude than any other comparable change of furnace variables (e.g., of emissivity of combustion products, of flow and heat-release pattern, etc.).

In the following, the effect of ash deposits on furnace heat transfer is investigated in more detail through the use of an upgraded version of the multizone furnace model. Provided outside temperature, thickness ΔS , thermal conductivity λ and emissivity ϵ_t of a deposit layer are known, this model allows the evaluation of profiles of inner surface temperatures by means of heat balances of the surface zones coupled with solution of heat balance of furnace volume zones.

¹ Index ϵ_t in this case means emissivity of 'tube' walls and is not identical with ϵ_t defined by eq. (2.15).

The study of the effect of ash deposits was carried out for a tangentially fired coal combustion chamber of 875 MW fuel heat input. Figure 5-1 shows the distribution of net heat flux densities along the furnace height computed for the center column of furnace side wall zones for different conditions of the tube wall surfaces. The surface conditions were specified by the following input data:

- Case A Clean surfaces, emissivity of tube surface $\epsilon_t = 0.9$.*
- Case B Powdery ash deposit, $\epsilon_t = 0.9$, $\Delta S = 0.5 \text{ mm}$, $\lambda = 0.3 \cdot 10^{-3} \text{ kW/mK}$.
- Case C Same as B; however, $\epsilon_t = 0.6$.
- Case D Properties of ash deposit layer in upper part of the furnace (above heat-release zone) as specified for case C; glassy ash deposit layer in lower part of the furnace with $\epsilon_t = 0.8$, $\Delta S = 4 \text{ mm}$, $\lambda = 10^{-3} \text{ kW/mK}$.

It should be pointed out that the properties for the powdery (primary) and for the glassy (molten) deposit layers of cases C and D correspond to the average data found in references 4 and 13.

It can be clearly seen from Figure 5-1 that build-up of ash deposits at the furnace walls has serious effects on overall and local heat transfer.

Computed furnace efficiencies η_f are:

- Case A $\eta_f = 31.6 \text{ percent}$
- Case B $\eta_f = 29.6 \text{ percent}$
- Case C $\eta_f = 26.7 \text{ percent}$
- Case D $\eta_f = 25.6 \text{ percent}$

The difference ($\Delta\eta_f$) in computed furnace efficiencies for the extreme cases, A (clean walls) and D (highest thermal resistance), is 6 percentage points. The assumed severe slagging and fouling in case D reduced maximum heat flux densities by 39 percent. The comparison of case D with case C shows that

*The actual calculations were carried out with an effective emissivity of the tube walls, taking the shadow effect of the gap between tubes into account.

- a Clean, $\epsilon_t = 0.9$
- b Powdery Ash Deposit, $\epsilon_t = 0.9$
 $\Delta S = 0.5\text{mm}$
- c Powdery Ash Deposit, $\epsilon_t = 0.6$
 $\Delta S = 0.5\text{mm}$
- d Same as c; however, glassy ash deposit in lower furnace ($<17.7\text{m}$)
With $\epsilon_t = 0.8$, $\Delta S = 4\text{mm}$

Figure 5-1. Distribution of Heat Flux Density Computed in Tangentially Coal-Fired Boiler at Side-Wall Center Zones for Different Conditions of Tube-Wall Surface.

an increase of the thickness of ash deposit layers opposite to upper heat-release zone moves the peak heat fluxes further upwards into the region of the thinner powdery deposit. This is one of several reasons why the build-up of deposit layers, once it has started, spreads into adjacent wall zones.

Computed surface temperature distributions corresponding to the heat fluxes of Figure 5-1 are shown in Figure 5-2. The computations for clean walls were carried out for a surface temperature of the tubes of 650°K . For simplicity, this temperature was also assigned to the metal tube surface when covered with ash deposits. The location of peak surface temperatures for layers of uniform thicknesses (cases B and C) coincide with computed peak heat flux densities. It is noteworthy that the deposit with a surface of lower emissivity (case C) has lower surface temperatures, although incident heat fluxes are larger than in case B. The absorption efficiency of the furnace computed for case C is smaller by $\Delta\eta_f = 2.9$ percentage points than for case B. The effect of a change in emissivity would have been stronger for clean walls.

The surface temperatures computed for cases B and C are still far beyond initial deformation temperatures of coal ashes. Yet, deposits can be built up when hotter particles from the flame zones, especially those of eutectic composition, come into contact with tube surfaces. Once the deposit layers are growing, surface temperatures soon reach values in the range between softening and melting temperatures of coal ashes as shown by curve D, which was obtained assuming a 4 mm ash layer in the bottom of the furnace. In principle, for given melting temperatures of coal ashes, the current furnace model would also be able to predict approximately the thickness and extent of molten surface layers.

a Clean, $\epsilon_t = 0.9$

b Powdery Ash Deposit,
 $\epsilon_t = 0.9$
 $\Delta S = 0.5\text{mm}$

c Powdery ash Deposit, $\epsilon_t = 0.6$
 $\Delta S = 0.5\text{mm}$

d Same as c; however, glassy ash
deposit in lower furnace ($<17.7\text{m}$)
With $\epsilon_t = 0.8$, $\Delta S = 4\text{mm}$

Figure 5-2. Distribution of Surface Temperatures Computed in Tangentially Coal-Fired Boiler at Side-Wall Center Zones for Different Conditions of Tube-Wall Surface.

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