Pulverised-Coal Combustion with Staged Air Introduction: CFD Analysis with Different Thermal Radiation Methods

Risto V. Filkoski*

University "Sts Cyril and Methodius", Faculty of Mechanical Engineering Karpoš II, P.O.Box 464, Skopje, Republic of Macedonia

Abstract: Combustion chamber designers endeavour to achieve optimum operating conditions that give maximum combustion efficiency, together with minimum pollutant formation rate. Modelling of fossil fuel utility boilers has reached a remarkable development in recent years. Particularly, the application of computational fluid dynamics (CFD) modelling technology and other advanced mathematical methods offer opportunities for analysis, optimisation and options examination in order to increase the overall efficiency of the energy facilities. The main purpose of the present study was to investigate how the results obtained with two radiative heat transfer methods, the P1 approximation method and the discrete ordinates (DO) method, fit temperature field in a boiler furnace on pulverised coal, with implemented over-fire air (OFA) ports. The overall framework of the CFD modelling approach is described. The numerical modelling results for boiler baseline operating conditions are compared with a test matrix of local temperature measurements. An accuracy analysis of the P1 and DO methods is done on a basis of a comparison between the numerically obtained and measured temperature profiles.

Keywords: Coal combustion, heat transfer, thermal radiation, CFD techique, modelling.

INTRODUCTION

Well-grounded technical solutions of boiler furnaces and combustion chambers are aimed to achieve operating conditions that give optimal heat transfer, maximum combustion efficiency and minimum pollutant formation rate. This leads to increasing the overall efficiency of the energy facilities and is particularly important in the cases when coal is used as fuel. Despite the calls for larger utilisation of gas and renewable energy sources, the role of coal is crucial and, probably, its importance as energy resource is not going to be endangered significantly in the coming decade or two. There is a certain re-emergence of coal as a primary fuel for power generation, as a result of technological changes that have reduced the amount of pollutants and particulate emissions to the atmosphere, and recent international worries about security of gas supply. At the same time, the interest on performance optimisa-tion of utility boilers has become very relevant, aiming at extending their lifetime, increasing the overall efficiency and reducing the emissions of pollutants, which is quite clear with the rising costs and increasing demand for energy.

In many combustion devices, including boiler furnaces, the radiant heat transfer is the dominant mode of energy transfer. In order to predict accurately the temperature field, which is in direct correlation with the combustion efficiency and NO_x formation, it is important to utilise proper radiative heat transfer model. As a part of the efforts to establish

appropriate modelling and simulation approach for certain conditions, the main objective of the present study was to investigate how some established thermal radiation methods describe the temperature field in a furnace of utility boiler on pulverised coal with staged air/fuel introduction.

MODELLING AND SIMULATION TECHNIQUES FOR FURNACE APPLICATIONS

The mathematical modelling and numerical simulation have already been widely accepted as low-cost, efficient engineering tools for analysis and optimisation of combustion systems [1]. These tools considerably facilitate the investigation of the influence of various process and design parameters, such as fuel proximate and ultimate analysis, particle size distribution, air and fuel inlet distribution, overall furnace geometry (furnace shape, air and fuel inlets, layout of air staging nozzles, recirculation holes), burners' design, flame-wall interaction phenomena, heat transfer degradation, slagging propensity etc. to the heat transfer conditions, overall combustion efficiency, plant efficiency and the emission of pollutants. Advanced combustion models and computational fluid dynamics (CFD) techniques are extensively used to accurately solve the aerodynamics equations involved in the problem, and to predict coal burnout and heat transfer in large utility boilers. Compared to a purely experimental study, CFD modelling approach offers significant benefits: cost and time reduction, reproducibility of boundary conditions, as well as detailed insight into the complex interacting physical phenomena determining the combustion process.

A comprehensive historical review of the modelling approaches and techniques of pulverised coal boilers is given in [2], pointing out very different standpoints and objectives.

*Address correspondence to this author at the University "Sts Cyril and Methodius", Faculty of Mechanical Engineering Karpoš II, P.O.Box 464, Skopje, Republic of Macedonia; Tel: +389 2 3099 224; Fax: +389 2 3099 298; E-mail: rvfilko@mf.edu.mk; rvfilko@yahoo.com

The traditional approach of boiler manufacturers, that is essentially identical to the usual design practices of boilers and furnaces of very different nature, is based on the use of relatively simple semi-empirical models, developed in pilot plants or inferred from the practical experience.

Since the early 70s, more elaborated models for predicting heat transfer in furnaces started to appear. A typical engineering calculation method is elaborated in [3]. In the same work, a plug-flow method is described, aimed for simple furnace calculations, and known as zone method. The method is based on division of the domain into a few amount of volume cells and it has been widely used by researchers and engineers (for instance [4, 5]). This was some kind of a predecessor of the application of the popular Hottel zone method for radiative heat transfer, already available in literature [6], but still not applied in the design of utility boilers, since it requires previously defined reliable velocity field and heat release (combustion) patterns. The former Soviet approach to the furnace modelling is also expressed through the works of several other authors [7, 8].

The adoption of the various forms of zone methods coincided and, in some extent, was one of the preconditions for development of CFD techniques, with all their advantages and shortcomings. Nowadays researchers' efforts are concentrated in two general directions: (1) to improve the fundamental models of the complex phenomena that occur in the combustion systems and (2) to adopt some simplifications in order to make possible the extrapolation of research results to industrial applications.

The permanent progress in computers capability over the last two decades has enabled development and application of massive mathematical models of turbulent flows, heat transfer and combustion in the furnaces and extensive use of CFD techniques, with quite satisfactory results [9-14]. Three-dimensional comprehensive models of industrial-scale furnaces have been developed and successfully applied for years now [9-20, 22]. There are also combinations of original models of separate processes and existing computational codes [23, 24]. Of course, there is still an area for further improvements, having as a subject a detailed mathematical description of physical and chemical processes in certain specific conditions. Numerical codes describing the processes in coal-fired boilers, based on solution of differential conservation equations, have been a subject of many investigations [10, 15-20, 22]. While some consider numerical aspects [17] or aerodynamics [22] in small-scale furnace isothermal models, others present comprehensive combustion models of large-scale furnaces. The majority of the models use variations of the SIMPLE algorithm for coupling the continuity and momentum equations. Despite some weaknesses, the k-ε turbulen-ce model, or some derivatives, like RNG k-ε model [10], or k-ε-k_p two-phase turbulence model [19], are often used in combustion systems. Gas phase conser-vation equations are mostly time-averaged, but some prefer the Favre-averaged equations instead [16].

Two-phase flow is usually described by Eulerian-Lagrangian approach and PSI-Cell method for taking into account the influences between phases, with some exceptions using Eulerian-Eulerian approach, or two-fluid trajectory model [19].

Most of the combustion sub-models given in [10, 15, 16, 18-20, 22] separately treat particle devolatilisation, char oxida tion and additional gas phase reactions. One of the most important segments of the furnace modelling is the way of radiative heat transfer incorporation. Thermal radiation in the furnaces can be modelled by means of various approaches, like discrete transfer method [15], discrete ordinates method [16, 19-21], six-fluxes method [18], Monte Carlo method [10], or so called P-N approximation method [25], including its P-1 variant, as it is used in [26, 27]. In general, it should be pointed out that a comprehensive model of the furnace processes must balance sub-models sophistication with computational practicality.

DESCRIPTION OF THE CASE STUDY: BOILER AND FUEL MAIN PROPERTIES AND FURNACE GEOMETRY

The modelling technique described in this paper has been validated against data acquired from the corner-fired pulverised coal boiler at a power plant with gross output of 160 MWe [28]. A top view (cross-section), showing the boiler dimensions and inlet air trajectories, is presented in Fig. (**1**) [28]. The boiler furnace is rectangular in shape, 11.58 m long and 8.53 m wide. The inlet air and coal are fed from the furnace corners, at 42° and 31°, as shown, to create a fireball swirling (counterclockwise, as viewed from the top). The boiler is retrofitted with low-NO_x burners, concentric firing system (CFS) ports, close-coupled overfire air (CCOFA) and separate overfire air (SOFA) ports. The boiler was modified with fifteen 152 mm diameter circular ports at various levels, which can accommodate appropriate water-cooled probes for temperature measurement. The port locations are selected taking into account the importance of data from various locations. A schematic diagram of the boiler furnace represented in Fig. (**1**), also indicates the location of various fuel, air and measurement ports.

The coal properties for the test case and the boiler main operating conditions are listed in Tables **1** and **2** [28]. The secondary air flow includes air through CFS, CCOFA and SOFA ports. A schematic representation of the ports: top SOFA, mid SOFA, bottom SOFA, top CCOFA, bottom CCOFA, coal burners No. 1-4, CFS ports and auxiliary oil ports, is given in Fig. (**1**) [28].

Some details of the burners' configuration, including the height of each port with respect to the datum, which is at the bottom of the ash hopper, and the port area, are shown in Table **3** [28]. For the modelling purposes, it was assumed that the air was evenly distributed between all inlet ports in proportion to the areas reported in Table **3** and that those areas were adjusted for the test according to the percentage opening. The inlet air was heated to 260^oC and the ratio of primary air to as-received coal mass flow rate was 1.7.

The simulations have been performed on a basis of the Fluent CFD code, version 6.1, adapted for pulverised coal industrial and utility scale boiler furnaces. Geometry outline of the calculation domain and the numerical mesh are presented in Fig. (**2**). Special attention has been paid to the grid design, in order to reduce numerical error as much as possible. The fact that the smallest lengths in combusting turbulent fluid flows are in the much smaller magnitude order, compared to the huge dimensions of utility boiler furnaces,
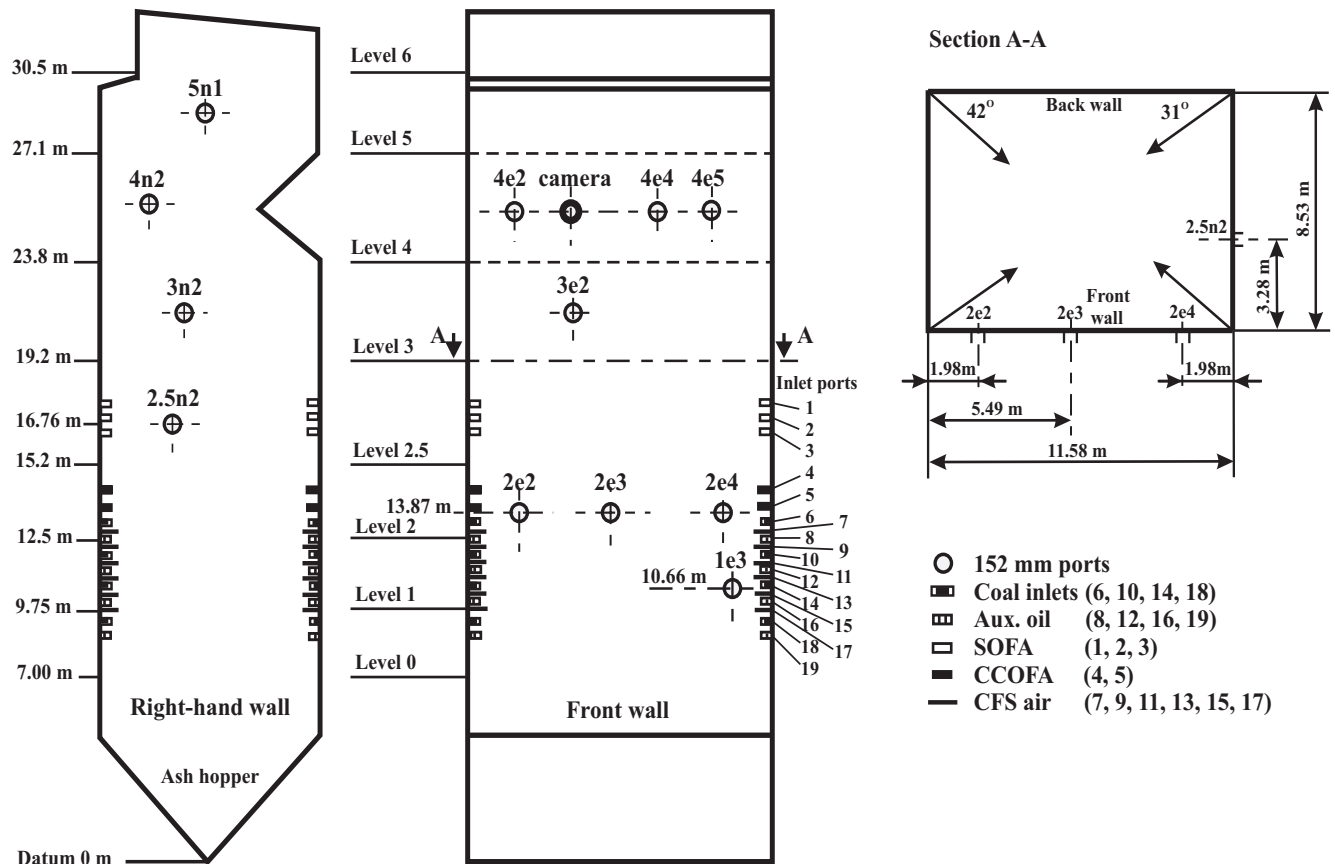
Fig. (1). Schematic representation of the boiler furnace with disposition of air and fuel inlets, measurement ports and direction of burners [28].

make it necessary either to discretise the solution domain using control volumes with dimensions of the order of some milimeters in all directions, either to find some other solution. Considering the time available and the computer resources, the first option is not feasible.

other boiler furnaces with comparable domain dimensions and complexity, the grid is fine enough to give grid-independent solutions, showing satisfactory convergence and accuracy and, at the same time, to meet the restrictions in computational time [26, 27, 29].

Table 1.   Coal Properties (as Received) [28]

Proximate Analysis	Value
Total moisture, wt %	4.1
Volatile matter, wt %	28.3
Fixed carbon, wt %	55.7
Ash, wt %	12.0
Mean particle diameter, μm	27
Coal heating value, kJ/kg	31040

Table 2.   Boiler Operating Conditions [28]

Parameter, Unit	Value
Gross load, MWe	156
Coal feed, kg/h	50,200
Primary air flow rate, kg/h	85,340
Air temperature, $^{\circ}$C	260
Secondary air, kg/h	546,000
Burner tilt, deg.	-5

As a compromise, much denser mesh is utilised in regions where large gradients of variables can be expected, such as the near-burner-regions. In this case, there are totally 688,886 tetrahedral hybrid cell elements in the domain. According to the previous experiences with simulations of

Thermocouple pyrometry, as a practical and reliable method of measuring gas temperature in full-scale boilers, is used in this case. The procedure for temperature measurements is described in details in [28]. Gas temperature measurements are obtained with a type S, Pt/Pt-10%Rh

Table 3.   Geometry Properties of the Furnace Inlet Ports

| No.	Description	Height - Centre, m	Area, cm^2	Damper Position, % Open
1	Top SOFA	17.4	677	0
2	Mid SOFA	16.9	677	0
3	Bottom SOFA	16.4	677	100
4	Top CCOFA	14.0	709	100
5	Bottom CCOFA	13.5	948	100
6	Top coal burner 1	13.0	1077	100
7	CFS port	12.7	215	17
8	Auxiliary oil ports	12.4	460	55
9	CFS port	12.1	215	17
10	Coal burner 2	11.8	1077	0
11	CFS port	11.5	215	17
12	Auxiliary oil port	11.2	302	55
13	CFS port	10.9	215	17
14	Coal burner 3	10.6	1077	100
15	CFS port	10.3	145	17
16	Auxiliary oil port	10.0	399	55
17	CFS port	9.7	191	17
18	Coal burner 4	9.4	1077	100
19	Auxiliary oil port	8.9	127	55
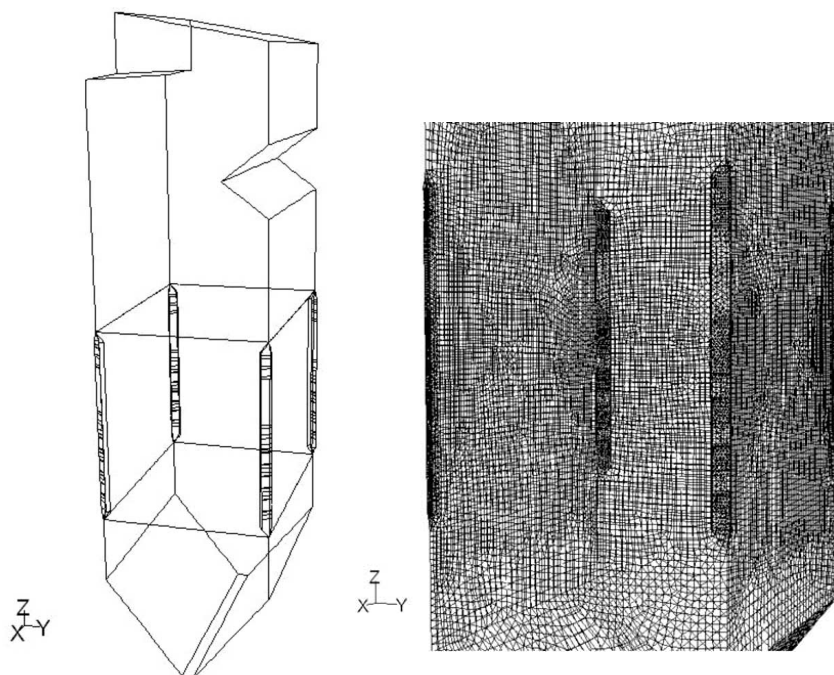
Fig. (2). Furnace geometry and the numerical mesh in the burners' region.

thermocouple in a triply shielded suction pyrometer. A venturi pump is used to draw combustion products through an opening in the outer ceramic tube and between the smallest and medium-sized tubes.

MATHEMATICAL MODELLING APPROACH

Aerodynamic processes in the boiler furnaces are complex and very important for overall proper operation of the energy facilities. A furnace of a steam boiler with pulverised coal combustion, as an object for numerical modelling, is characterised with a compressible particle-laden flow, reactions and heat transfer phenomena occurring in the turbulent flow. The most important physical and chemical processes, which have to be covered by the model, are the turbulent two-phase flow of gas and suspended particles, the release of the coal volatile matter, the chemical reactions of the released species, the burnout of the char particles and the heat transfer governed by thermal radiation. Accordingly, the comprehensive boiler furnace model consists of three main interacting parts: the flow model for the turbulent two-phase flow, the reaction model for the homogenous and heterogeneous combustion reactions, and the heat transfer model with emphasize on the thermal radiation as the most important mechanism.

One of the possibilities for improvement of the operation efficiency of the pulverised coal-fired boilers is well-grounded technical solution aimed at reduction of NO_x emission, together with the subsequent suppressing of heat loss due to inefficient combustion. The present research was concentrated on an analysis of the flow field, temperature field and combustion efficiency, as a result of a staged air introduction, but in this work only the temperature field is a subject of analysis.

Although the smallest lengths in combusting turbulent fluid flows are in the magnitude order that is much smaller than the boiler furnaces dimensions, in most cases, the time-averaged mean value of the flow situation is of primary interest. Proper turbulence models are used as a remedy for this problem. For the sake of computational performance, the simple k-ε turbulence model is applied in majority of applications, although some predictive quality is sacrificed. Due to the assumptions of prevailing isotropic turbulence, this relatively crude model exhibits some weaknesses when applied to flows with strong streamline curvature, as in the near-burner area of swirl burners. Also, very often, in boiler furnaces significant recirculation regions appear, especially near the furnace corners. The k-ε model is known to exhibit certain predicaments in accurately describing such phenomena. Therefore, the standard k-ω turbulence model is employed in this case as a reasonable compromise [30]. As the k-ω model has been modified over the years, production terms have been added to both the k and ω equations, which have improved the accuracy of the model predictions. Common values of the constants are used in the transport equations: α_∞^*=1, α_∞=0.52, α_0=1/9, β_∞^*=0.09, β_i=0.072, R_β=8, R_k=6, R_ω=2.95, ζ^*=1.5, M_{t0}=0.25, σ_k=2.0 and σ_ω=2.0. Coupling of velocity and pressure is achieved by the SIMPLEC algorithm.

A simulation of pulverised coal combustion involves modelling of a continuous gas phase flow and its interaction with a discrete phase. The particles, traveling through the gas, devolatilise and undergo char combustion, creating sources of fuel for reaction in the gas phase. In this work, the concentration of species and the chemical reactions are modelled using the mixture fraction/probability density function (PDF) approach and the full equilibrium chemistry, where the turbulence-chemistry interaction is modelled using a double-delta PDF [31]. It is assumed that the two-streams PDF mixture consists of 20 volumetric species. The equilibrium chemistry model assumes that the chemistry is rapid, so that chemical equilibrium always exists at the molecular level. An algorithm based on the minimization of Gibbs free energy [31] is used to compute species mole fractions. The equilibrium model is useful, since it can predict the formation of intermediate species and it does not require knowledge of detailed chemical kinetic rate data. Instead of defining a specific multi-step reaction mechanism, it needs simply to define the important chemical species that will be present in the system. The mole fraction of each species is then predicted, based on chemical equilibrium assumption.

Regarding the influence of the membrane walls to the aerodynamics, it is assumed that the particle-to-wall collisions occur with certain elasticity. This is defined by setting reflecting boundary condition type, with constant discrete phase reflection coefficients - normal and tangent.

The polydisperse coal particle size distribution is relatively narrow and it is assumed to fit the Rosin-Rammler equation. Particle trajectory data, size distribution parameters, coal devolatilisation and coal combustion parameters used in the model are given in Tables **4** and **5**.

Table 4.   Coal Particle Trajectory Data

Parameter	Value
Number of particle stream start locations	16
Maximum number of steps in each trajectory	1200
Length scale	0.1 m
Particles distribution – Rosin-Rammler	
Number of particle diameters	6
Mean diameter	0.027 mm
Minimum diameter	0.01 mm
Maximum diameter	0.15 mm
Spread parameter	2.0

HEAT TRANSFER AND THERMAL RADIATION MODELLING

In this case, the energy equation is used in the following total enthalpy form

$$\frac{\partial}{\partial t}(\rho H) + \nabla \cdot (\rho \vec{v} H) = \nabla \cdot \left(\frac{k_t}{c_p} \nabla H \right) + S_h \tag{1}$$

where ρ denotes gas-phase density, k_t is the turbulent thermal conductivity, defined according to the turbulence

model being used. The term S_h includes the heat of chemical reactions and any other volumetric heat sources.

Under the assumption that the Lewis number (Le) is equal to 1, the conduction and species diffusion terms combine to give the first term on the right-hand side of the above equation, while the contribution from viscous dissipation appears in the non-conservative form as the second term.

Table 5.   Coal Combustion Parameters

a) Coal devolatilisation data	
Devolatilisation model – two competing rates	
1) First rate	
- pre-exponential factor	$2.0 \cdot 10^5$ s^{-1}
- activation energy	$7.50 \cdot 10^7$ J/kmol
- weighting factor	0.3
2) Second rate	
- pre-exponential factor	$1.3 \cdot 10^7$ s^{-1}
- activation energy	$1.45 \cdot 10^8$ J/kmol
- weighting factor	1.0
b) Combusting particles properties	
Density	1250 kg/m^3
Specific heat capacity – polynomial profile	
Thermal conductivity	0.045 W/mK
Mass diffusion limited rate constant $4.5 \cdot 10^{-12}$	
Kinetic rate pre-exponential factor 0.002	
Activation energy	$7.5 \cdot 10^7$ J/kmol

The total enthalpy H is defined as:

$$H = \sum_j Y_j H_j \tag{2}$$

where Y_j is the mass fraction of species j and H_j is sensible enthalpy of species j:

$$H_j = \int_{T_{ref,j}}^{T} c_{p,j} dT + h_j^0 \left(T_{ref,j} \right) \tag{3}$$

In this equation, $c_{p,j}$ is specific heat capacity and $h_j^0 \left(T_{ref,j} \right)$ is a formation enthalpy of species j at the reference temperature $T_{ref,j}$.

Energy sources due to radiation, as well as the heat transfer between the continuous and the discrete phase, are included through the term S_h in Eq. (1).

The furnace of a pulverised coal-fired boiler is an example of a space with an emitting-absorbing and scattering medium. The balance of energy passing in a specified direction Ω through a small differential volume in an emitting-absorbing and scattering medium, at position **r** in direction **s**, including the effect of the presence of discrete phase, can be represented through this form of the radiative transfer equation (RTE)

$$\frac{dI(\mathbf{r},\mathbf{s})}{ds} + (a + a_p + s_p)I(\mathbf{r},\mathbf{s}) =$$

$$an^2 \frac{\sigma T^4}{\pi} + E_p + \frac{\sigma_p}{4\pi} \int_0^{4\pi} I(\mathbf{r},\mathbf{s}')\Phi(\mathbf{s} \cdot \mathbf{s}')d\Omega' \tag{4}$$

where I (**r**,**s**) [W/m^{2}srad] is the radiation intensity, which is a function of the wave length, position and direction; **r** is a position vector; **s** is a direction vector; a denotes absorption coefficient, a_p is the equivalent absorption coefficient due to the presence of particles; σ_p is the equivalent particle scattering factor and E_p is the equivalent emission of the particles, explained bellow. Also, in Eq. (4) Φ is phase function, which represents the probability that a wave of frequency ν' from direction **s** in elementary space angle $d\omega'$ will switch (deviate) towards direction **s'** inside the angle $d\omega$ with frequency ν; **s'** is a vector of scattering direction.

The particles equivalent emission coefficient (E_p) and the equivalent absorption coefficient (a_p) are given with the following expressions

$$E_p = \lim_{V \to 0} \sum_{n=1}^{N} \varepsilon_{pn} A_{pn} \frac{\sigma T_{pn}^4}{\pi V} \tag{5}$$

and

$$a_p = \lim_{V \to 0} \sum_{n=1}^{N} \varepsilon_{pn} \frac{A_{pn}}{V} \tag{6}$$

In Equations (5) and (6), ε_{pn}, A_{pn}, and T_{pn} are emissivity, projected area, and temperature of particle n. The summation is over N particles in volume V.

SELECTION OF A RADIATION MODEL

The selection of the most appropriate thermal radiation model in certain conditions depends on various factors, such as the optical thickness, the possibility for inclusion the scattering and emissivity effects, the way the model is treating the effects of the presence of discrete phase and the model behaviour in the case of medium with localised heat sources. In this case, the Discrete Ordinates (DO) and the P-1 models are utilised [21, 25, 32-35], since they effectively comprise the influence of the discrete phase presence in the boiler furnace, unlike some other frequently used models in thermal engineering applications, like the Discrete Transfer Radiation Model (DTRM) [15, 32, 36].

The optical thickness of the radiating medium aL, where L is characteristic path length, is one of the indicators of which model to use in the analysed problem. If the optical thickness is large, $aL \gg 1$ or $aL>1$, the P-1 model should typically be used. The DTRM and the DO model work across the range of optical thicknesses, but are substantially more expensive to use.

The P-1 and DO models account for scattering and emissivity, while the DTRM neglects it. Also, only the P-1 and DO models account for exchange of thermal radiation between the gas and particulates.

In problems with localised heat sources, such as solid fuel particles, the DO model is probably the best suited for computing radiation.

The DO model considers the radiative transfer equation in the direction **s** as a field function and, including the contribution of a discrete phase of particulates on radiation, the RTE can be written as Eq. (4). In this case, the so-called S6 approximation was applied in the framework of the DO model, corresponding to 48 flux approximations [21]. This approach yields sufficiently reasonable results for the amount of the numerical work. The higher-order approximations, such as the S8, with 80 flux approximations, require considerably more numerical effort.

The the P-1 model, as simplified P-N differential approximation, has certain specific advantages over other models in treating the radiative energy transfer in a grey absorbing and emitting medium with presence of particulates. It is relatively simple, treats the RTE as an easy-to-solve diffusion equation and it can be easily applied to complicated geometries. For a gray, absorbing, emitting, and scattering medium containing absorbing, emitting, and scattering particles, the transport equation for the incident radiation can be written as

$$\nabla \cdot (\Gamma \nabla G) + 4\pi \left(a \frac{\sigma T^4}{\pi} + E_p \right) - (a + a_p)G = 0 \qquad (7)$$

where E_p is the equivalent emission and a_p is the equivalent absorption coefficient of the particles.

The utilised values of the particles thermal radiation characteristics are derived as approximations of data from various sources [37,38]. The scattering coefficient, along with the absorption coefficient, describes the change in radiation intensity per unit length along the path through the furnace medium. In order to include the effect of an anisotropic scattering, in this case, a Delta-Eddington phase function is implemented in the following form:

$$\Phi(\mathbf{s} \cdot \mathbf{s'}) = 2 f \delta (\mathbf{s} \cdot \mathbf{s'}) + (1 - f)(1 + C \, \mathbf{s} \cdot \mathbf{s'}) \qquad (8)$$

The variable absorption coefficient, in this case, is defined as composition-dependent, with a local value of a as a function of the local mass fractions of water vapour (H_2O) and CO_2. For that purpose, the weighted-sum-of-gray-gases model (WSGGM) is used [39], which is a compromise between the over-simplified gray gas model and a complete model that takes into account particular absorption bands. According to the WSGGM, the total emissivity over the distance s is presented as

$$\varepsilon = \sum_{i=0}^{I} a_{\varepsilon,i}(T)\left(1 - e^{-\kappa_i p s}\right) \qquad (9)$$

where $a_{\varepsilon,i}$ are the emissivity weighting factors for the i-th "fictitious" gray gas, the bracketed quantity is the i-th gray gas emissivity, κ_i is the absorption coefficient of the i-th gray gas, p is the sum of partial pressures of all absorbing gases, and s is the path length. In the above equation, the path length s is calculated according to the mean-beam-length approach, based on an average dimension of the domain. The effect of the soot concentration on the radiation absorption coefficient is included in the simulations. The absorption coefficient of a mixture of an absorbing (radiating) gas and soot is calculated as a sum of the absorption coefficients of pure gas and pure soot [40]:

$$a = a_g + a_s \qquad (10)$$

where a_g is the absorption coefficient of gas without soot, obtained from the WSGG model and a_s is approximated as

$$a_s = b_1 \rho_s [1 + b_T (T - 2000)] \qquad (11)$$

with $b_1 = 1232.4$ m^2/kg, ρ_s denotes the soot density, and $b_T = 4.8 \cdot 10^{-4}$ K^{-1}.

RESULTS AND DISCUSSION

The present case study is a typical example where the CFD modeling can give much clearer insight into the processes in a furnace with design modifications. Practically all the parameters of the combustion process change when using the multi-staged (three-staged) combustion scheme. This fact, in turn, essentially influences the efficiency of the combustion process and the temperature field in the furnace.

Some results of the numerical simulations, comparing horizontal temperature profiles at different levels of the boiler furnce obtained by the use of DO and P-1 radiation models and measurements, as well, are presented in Figs. (3 and 4). The calculations were verified against the experimental data [28] and indicated good agreements in most cases, with some acceptable discrepancies. The temperature profiles showed that the use of over-fire air altered the near-burner temperatures, reducing the swirl relative to the incoming fuel. Similar effects are also properly predicted near the close-coupled over-fire air inlets. The general features of the flow indicate the formation of vortex that have reduced temperature levels at the middle of the furnace cross section in some cases, which is probably a result of the adverse pressure gradient.

Regarding the thermal radiation modelling approach, both models, the DO and the P-1, give relatively fair representationof the experimental results. In the most cases, the P-1 model slightly overestimates the heat fluxes from localised heat sources. However, it must be pointed out that the calculation using the DO model is more time consuming. The ratio of the computing times using the models DO (with the S6 approximation, which means 48 flux directions) and the P1 is about 2.0. In the case of the DO with S4 approximation (24 flux directions), the ratio of the computing times is 1.75.

Additional information on the results of the numerical simulations, showing the influence of the two radiation models to other properties, such as the average furnace exit temperature, the total heat transfer rate to the furnace surfaces, char conversion at the furnace exit, etc., is given in Table **6**.

An illustration of the change of the area-weighted average volume fractions of O_2 and CO along the furnace height is given in Fig. (**5**).

Profiles of the change of the total surface heat flux along the furnace height, obtained with the two radiation models, together with the average surface heat fluxes, denoted as P1-m and DO-m, are presented in Fig. (**6**).
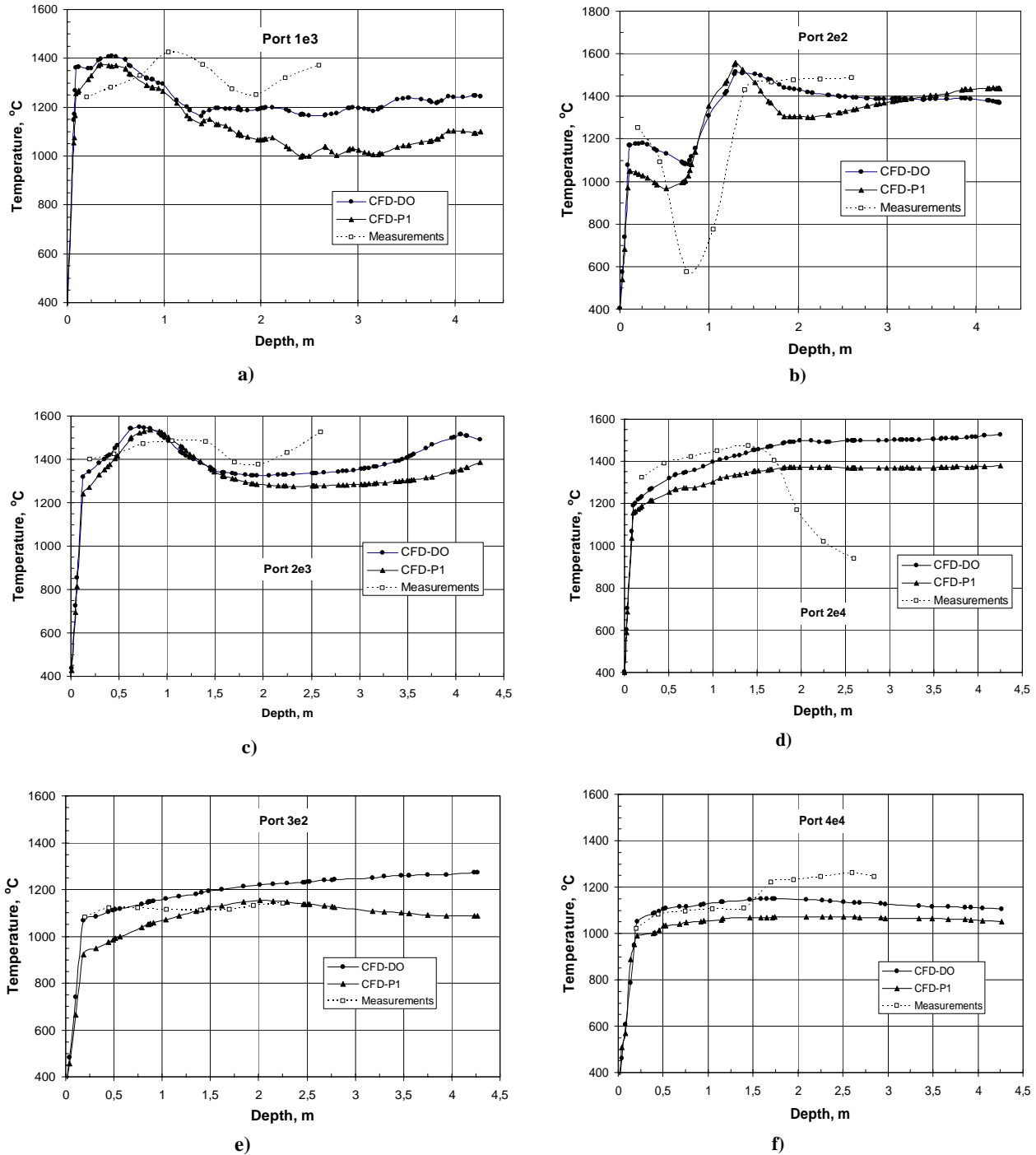
Fig. (3). Temperature profiles on the east (right-hand) wall for three levels of SOFA: **a**) Port 1e3, **b**) Port 2e2, **c**) Port 2e3, **d**) Port 2e4, **e**) Port 3e2, **f**) Port 4e4.

From the comparison of predicted and measured temperatures, the relatively high fluctuations of calculated temperature profiles, as well as certain discrepancies between the numerical predictions and measurements are obvious. The largest differences appear in the burnout zone, near the fuel and air inlets, and they can be put down to several possible reasons. Such differences could indicate that the flow field in these areas may not be very well predicted. However, there were no possibilities to check the quality of the on a basis of the available experimental results. Another reason may be the tendency of the P-1 model to over-predict radiative fluxes from localised heat sources, as it is suggested in [40]. Also, an important, but still relatively under-researched modelling area - the particles-turbulence interaction, could be a reason for appearance of differences between the calculated and measured temperature profiles. Finally, it must be
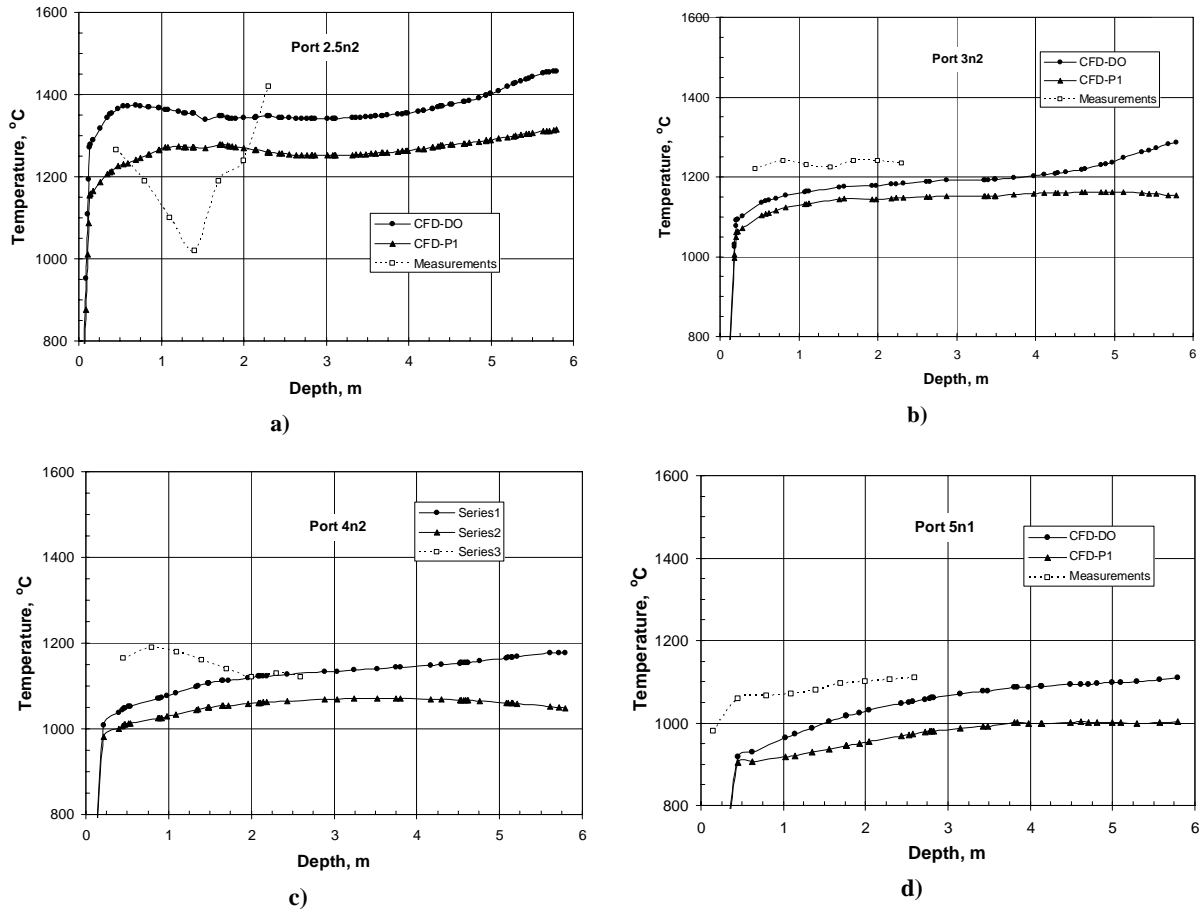
Fig. (4). Temperature profiles on the north wall for three levels of SOFA: **a**) Port 2.5n2, **b**) Port 3n2, **c**) Port 4n2, **d**) Port 5n1.
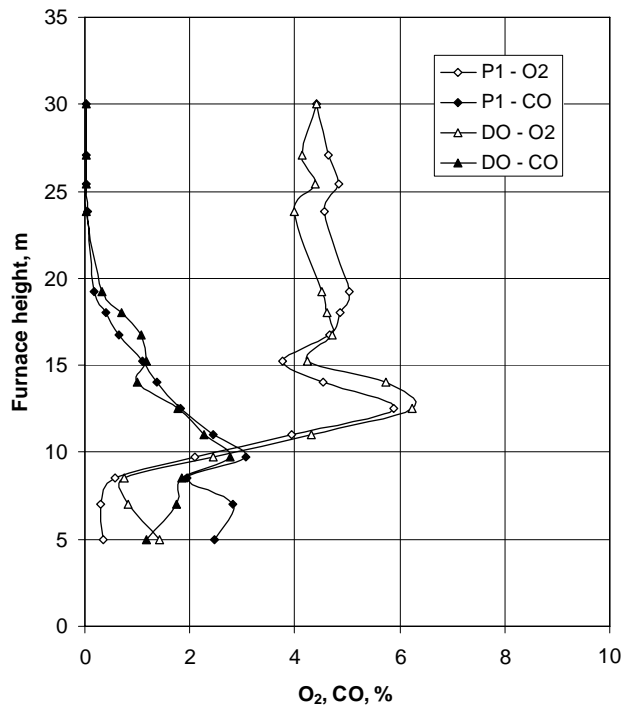

Fig. (5). Area-weighted average volume fractions of O_2 and CO along the furnace height.

Table 6. Some Results of the Numerical Simulation

Parameter, unit	Value	
	P-1 model	**DO model**
Furnace exit temperature (area-weighted average), ^oC	922	945
Total heat transfer rate to the furnace surfaces, MW	205.41	199.44
Area-weighted average total surface heat flux, kW/m^2	151.9	147.5
Char conversion, %	98.60	98.75

pointed out that the variability of coal properties requires detailed characterisation data for validation purposes of the boiler furnace processes.

CONCLUSIONS

The main objective of the present study was to investigate how the results obtained with two radiative heat transfer methods, the P1 approximation method and the DO method, fit the temperature field in a boiler furnace on pulverised coal, with implemented OFA ports. Despite the evident discrepancies, in general, the both thermal radiation modelling approaches, with the DO and the P-1 models, give fair repre-
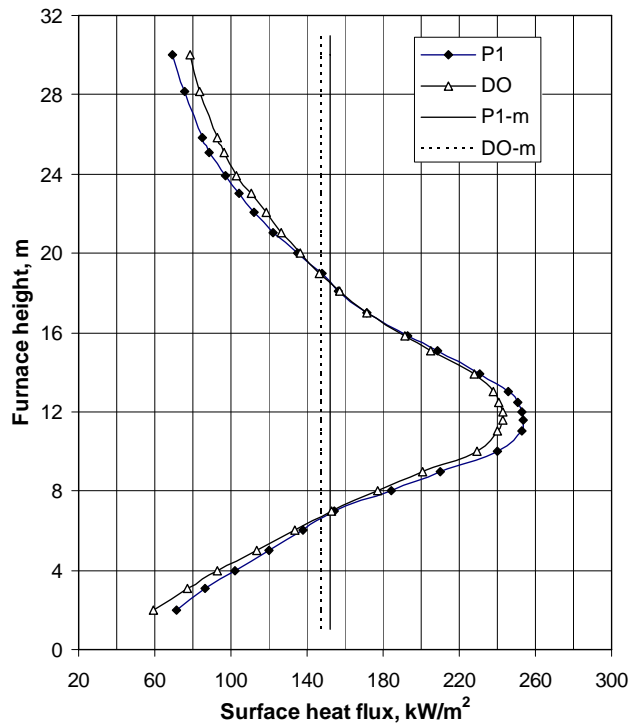
Fig. (6). Area-weighted average total surface heat flux along the furnace height.

sentation of the experimental results. The obtained temperature profiles with P-1 model are somewhat lower than in the case when the DO model is used. In that sense, the results obtained with the DO model better fit the measurements. The main differences between the modelling and the measurement values appear in the burnout zone, near the air/fuel injections, and need further investigation for appropriate explanation. In general, they can be addressed to several reasons: peculiarities of the flow field in the near-burner region, including the particles-turbulence interaction; the utilised thermal radiation models; and the variability of coal properties that requires much characterisation data for validation purposes of the boiler furnace processes. Finally, it must be noted that the computation with the DO model is more time consuming.

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