THREE-DIMENSIONAL FLAME STRUCTURE IN WALL-FIRED SWIRL FLAME COMBUSTORS

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ABSTRACT

A CFD solver CFX is used to analyze the complex behavior of turbulent reacting flow inside the furnace. The flow characteristics for various combustor geometries, fuel/air ratios, and injection velocities, and swirl levels are investigated. Starting with a cylindrical furnace fired with gaseous fuel from a concentric tube burner (both with and without swirl), the mixture-fraction is predicted using the k-ε and RSM turbulence models. The discrepancies between the predictions and measurements are most significant in the flame core of upstream regions. It may stem from inappropriateness of the assumed inlet conditions and the combustion model. However, the calculated results are still qualitatively acceptable. After the validation work of the numerical model, a rectangular furnace with four wall-fired swirling combustors is employed to investigate the effect of neighboring burners and geometry on combustion characteristics. The central recirculation zone which appeared in the isothermal flowfield vanished in the combustion case. It may be attributed to the fact that the hot gas suddenly expands outward and destroys the recirculation mechanism. Thus, the central flame could not hold. In addition, the four corner flames are stretching against the wall and their shapes are similar to a "cam" profile. The results are intended to assist in the development and validation of a numerical model for predicting furnace flows in wall-fired power plants.

INTRODUCTION

The efficient design and development of combustion systems requires careful characterization and optimization of very complex flow phenomena. The combustor designer has traditionally been forced to rely heavily on experience, experimental data and empirical expressions. However, such an approach, if used to the exclusion of available computational or analytical techniques, would soon prove both expensive and incapable of assimilating the vast amount of design information. Thus, computer modelling is becoming increasingly attractive as a complementary and supplementary tool to aid design procedures. The appropriate degree of numerical and analytical computations, when coupled with reliable data base, would greatly improve the predictions of trends for plant scale-up or enhance the intuitive understanding of fundamental physics.

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Industrial boilers in general are rectangular and with several wall-fired burners, as shown in Fig. 1A. The Primary air is introduced through a large fan, distributed to the various wall-burner ducts, and then passed through a concentric guide vane device to create swirling flow. The secondary air exhausts from a central, axial pipe in the wall-burner and carries pulverized coal dust to the mixing chamber. The combustion process is initialized via an appropriate mixing ratio and an ignition source. Therefore, the combustion efficiency and the exhaust flow concentrations are strongly dependent on the swirling flow strength, the main/secondary air ratio and the reversed flow structure. The main objectives of the study are to determine the effects of swirl and combustor geometry on combustion flowfield and to establish an improved simulation in the form of a computer prediction code equipped with suitable turbulence and combustion models.

In numerical predictions, Lilley and Rhode (1982) developed a numerical code -STARFIC (Swirling Turbulent Axisymmetric Recirculation Practical Isothermal Combustor) to calculate the axisymmetric swirl turbulent isothermal flowfield. Lockwood et al. (1974) made some measurements in a gas-fired cylindrical furnace, in which the effects of various fuel/air ratios, burner geometries, Reynolds numbers, and swirl numbers on flame structure were investigated experimentally. In addition, the numerical results which based on very simple physical modelling, are made and in good agreement with data. Soong (1992) made predictions for confined diffusion flames by using density-unweighted and density-weighted averaging approaches. The results are validated with Lockwood data. The comparisons reveal that the performance of the density-weighted model is unsatisfactory, although it is preferred for the exothermically reacting flows. Sokolov (1995) used the k-W model to discuss the heat and mass transfer characteristics in an annular combustor with opposite swirling air jets and assessed the dependence of combustor characteristics as a function of geometric and operational parameters.

We start with an axisymmetric swirl flow combustion geometry for the model validation effort and compare with the experimental data of Lockwood et al. (1974). After the model validation tasks, the furnace
Turbulence modelling

Finite difference equations

The finite difference equations are derived by integrating the governing equations over a control volume and expressing the results in terms of neighboring grid point values. To represent the convective and diffusive terms over the cell surface, the hybrid scheme is used. Upon volume integration, the convection and diffusion terms become surface integrals of the convective and diffusive flux and the source term is linearized. The resultant difference form may be rearranged to yield the general finite difference equations:

\[ A_i \phi_i = \sum A_{i} \phi_{n} + S_{i} \]

where \( \phi \) is the dependent variable, \( p \) is the nodal point, \( S_{U} \) is the constant part of the linearized source term, and \( nb \) denotes each of the six neighbouring nodes in the three-dimensional geometry.

Turbulence modelling

One of the most tempting and traditionally popular methods of formulating the Reynolds stress model, has been to assume that the turbulent stresses act analogously to the viscous stresses. The k-\( \varepsilon \) model is the key representative. However, one of the primary limitations of the k-\( \varepsilon \) model is the assumption of isotropy for the eddy viscosity. Therefore, stress models, such as the algebraic stress model (ASM) (Rodi, 1982) and the Reynolds stress model (RSM) (Hanjalic et al., 1979) are employed to improve computational accuracy. However, the RSM model requires a large amount of memory and computing time and it may be difficult to obtain a convergent solution. Consequently, in order to save computational time, the RSM model is employed in the present study only for the validation case. On the other hand, ASM models produce better predictions for the flowfield variables in most cases, but their performance is no better than the k-\( \varepsilon \) models for the confined, swirling flowfield (Sloan et al., 1986). Therefore, the standard two-equation k-\( \varepsilon \) turbulence model is employed for the 3-D flow simulation, which has been used in a wide variety of turbulent flow situations and which has achieved good predictive capability (Gupta and Lilley, 1983). The constants used in the model are defined as follows (Sloan et al., 1986):

\[ C_{p} = 0.09 \quad C_{1} = 1.44 \quad C_{2} = 1.92 \quad \sigma_{i} = 1.0 \quad \sigma_{r} = 1.3 \]

Combustion modelling

Recent modelling of turbulent diffusive flames often involves the fast chemistry assumption. Here, it is assumed that if fuel and oxidant are simultaneously present at the same point, an instantaneous reaction occurs. This mixed-burned model assumes that fuel and oxidant cannot coexist instantaneously. The mean mass fractions of fuel, oxidant and products are obtained from the mean and variance of the mixture fraction using an assumed form for the probability density function (PDF) of the mixture fraction. The double delta function PDF is assumed and used in the present investigations.

Boundary conditions

1. Inlet conditions
   1.1 Axisymmetric combustion flowfield
      Two different swirl strengths were generated by varying the vane angles of the burner in order to compare with Lockwood's (1974) measured data. The inlet flow conditions are shown in Table 1. The forced vortex inlet radial velocity profile is selected for all cases. The fuel and air properties are listed in Table 2.

1.2. Three-dimensional multi-burner furnace
      The guide vane angle is set to 45° and 60°, the inlet temperature is 300°K, and the inlet fuel axial velocity is 8.26 m/sec. The operating range of the equivalence ratio, \( \phi \), is from 0.1 to 0.6.

1.3 Turbulence properties
      The inlet values of k and \( \varepsilon \) are prescribed in accordance with the following equations (Lilley and Rhode, 1982):

\[ k_{i} = 0.03 \cdot u_{*}^{3} \quad \varepsilon = \frac{k_{i}^{2}}{0.0025D} \]

where \( D \) is the inlet diameter.

2. Outlet conditions
      Neumann boundary conditions are imposed on all transported variables, such as \( u, v, w, k, \varepsilon, f, g \), etc. This is equivalent to an assumption of fully developed flow at the outlet.

3. Solid boundary
      No slip and adiabatic conditions are imposed on the wall.

RESULTS AND DISCUSSIONS

Axisymmetric and three-dimensional multi-burner furnace flowfields are calculated in the study: A three-dimensional generalized curvilinear coordinate system is used in the numerical model. In addition, a body-fitted grid generation technique is employed to minimize the numerical errors from boundary approximation of the solid walls. The results of the predicted isothermal flowfield (Su et al., 1995) are not discussed here. The geometric models are shown in Fig. 1B. All numerical solutions presented here were obtained by using a commercial flow solver named CFX (formerly CFDS-FLOW3D), which is developed by AEA Technology/Harwell Laboratory.

Confined axisymmetric combustion flowfield

Predictions of mixture fraction have been made for a small, cylindrical furnace, which is axially-fired with gaseous fuel and a 0° and 45° swirl air supply. Since the fuel inlet and its neighbouring fluid are transported to the wall.
inlet are separated by a tube wall, the wall thickness may significantly affect the accuracy of the prediction, especially in the near field of the burner. Thus, care should be exercised in constructing the grid for this region. Since generating a well-distributed grid for the annular inlet cannot be completed by the built-in function of FLOW3D using one step, the annular region is divided into four quadra-circle regions.

Figure 2 presents a comparison between the predicted and measured [Lockwood et al., 1974] sectional profiles of mean mixture-fraction at $5^\circ$ swirl angle. The discrepancies between the predictions and measurements are most significant in the flame core of near field regions. It is generally understood that the chemical reactions occurring in the flame core include complex fuel decomposition, soot formation, radical oxidation and so forth. The scattering effects for thermal radiation due to the formation of soot in this region are also significant. Because of the assumptions in the mixed-is-burnt combustion model and neglect of the radiation effects, the present model, which overlooked these facts, cannot predict accurately inside the flame core. Figure 3 shows the similar behavior for the mean mixture fraction along the central axis. The data comparison of mixture-fraction with Soong et al. (1992) and measured data at $6^\circ$ swirl are presented in Figure 4. Our model predictions (the solid lines) are slightly better than Soong's (the dotted lines). Figure 5 presents the mixture fraction for the $45^\circ$ swirl at F/A=0.0623. The measured and calculated profiles are in reasonable agreement near the upstream of flame core. However, the predictions are poor downstream of X/D=1.19. Since the swirling motion produces a very strong mixing between the fuel and air in the near field region, the mixed-is-burnt combustion model is appropriate. Obviously, the mechanism tends to the chemical-controlled process in the downstream region and thus the combustion process deviates from the assumptions of the combustion model. Figures 6-7 give the mixture-fraction contour plots under the same operating conditions. On the whole, the qualitative flame structure can be adequately predicted by present numerical model. The results also show that at this instance, the use of a complex turbulence model does not improve the predictions significantly. Therefore, only the k-epsilon model is used in the subsequent three-dimensional flow cases.

The mixture fraction contour and temperature distribution plots with and without swirl are shown in Figs. 8 and 9, respectively. The introduction of swirl and the subsequent creation of a region of recirculation along the centerline causes hot gases to recirculate, thus raising the temperature of the reactants entering this stabilized region. The increased rate of decay of mixture fraction with axial distance resulting from the increased swirl is apparent. The stronger shear layer in the recirculation zone resulting from swirl provides better mixing and moves the flame upstream.

Three-dimensional furnace reacting flow with four burners

The grid system generated for the computation of three-dimensional furnace flow is shown in Fig. 10. Since the grid distribution affects the solution convergence significantly, much effort was expended in generating a well-distributed grid. Using this grid system, the inlet boundary conditions were easily prescribed. The calculated results include 45 and 60 deg. vane angles and the equivalence ratios, $\phi$, are from 0.1 to 0.6. However, due to the concentric burner configuration, the flame will blow out if $\phi>0.3$ for the 45 deg. case, and also for $\phi>0.6$ in the 60 deg. case. It is believed that the reversed flow structure is totally destroyed due to high inlet velocity of fuel jet. For the purpose of illustrating the flame structure, only the results for 45 deg. and $\phi=0.1$ are presented. The velocity vector plots at various axial planes are shown in Fig. 11. The solid line(s) in the figures represent the reversed flow region(s). On the X/D=0.5 plane, the recirculation zones due to the sudden expansion are clearly shown. Because the incoming flow contains a swirl velocity component, the recirculation zone generated in the region close to the furnace wall appears asymmetric. From Figs. 11c and d, the recirculation zones due to vortex breakdown are formed; the recirculation zone shapes are distorted into the "cam" shape because of the swirling motion. These recirculation zones provide the mechanism for flame holding for reacting flows inside a furnace. According to the previous numerical results of the isothermal flowfield [Su et al., 1995], a recirculation zone can be formed in the central region in addition to the four corner recirculation zones. It is believed that the formation of the central recirculation zone results from the interaction of these four swirling jets which can produce a very strong counter-clockwise vortex in the central region. Interestingly, this central recirculation zone vanished in the reacting flow case (see Figs. 11c and d). Since the apparent swirl number decreases in reactive flow relative to isothermal flow, it may result from the quick expansion of outward flow due to combustion which destroys the vortex breakdown structure.

The sectional temperature contours which are shown in Fig. 12 also demonstrate that the central flame does not exist. As a result, the flow characteristics and the flame structure may be changed because of the presence of the chemical reaction. Thus, the furnace cannot be designed from the information obtained from the non-reacting flow data alone.

Figure 13 gives a clear view of three-dimensional reversed flow structure. The axial length of the reversed region is extended to X/D=2.2, which is much longer than that of the isothermal case. This is probably caused by the hot expanded gas accelerated in the downstream direction. However, the recirculation velocities of the reversed flow are weakened due to the accelerated motion. Consequently, the flame-holding mechanism will be weakened. Figure 14 shows the predicted mixture fraction contours of the cross-sectional plane. Figure 15 gives the corresponding temperature contours under the same flow conditions. The comparison of the two contour plots reveals that the location of fs does not necessarily match the highest temperature. It is attributed to the interactions between the flow and heat transfer environments.

CONCLUSIONS

The present study uses CFX software to simulate the combustion flowfield in gas-fired cylindrical and rectangular swirling flow furnaces. The agreement between the calculations and the experimental data is qualitatively acceptable and serves to demonstrate CFX's suitability and applicability. The purpose is to develop a tool that is useful for design purposes with practical equipment, such as wall-fired co-generation furnaces. Analysis of the calculated flame structures suggests that:

1. The turbulence and combustion models employed are sufficient to predict the flow structure in non-swirling as well as swirling reacting flow for single and multi-burner furnaces with reasonable accuracy.
2. Swirl plays a dominant role on flame size and shape. It can shorten the flame length and increase the combustion intensity in the near field region.

3. In the case of multi-burner combustion, the mechanism which was observed in the isothermal flow and responsible for the appearance of the central recirculation zone is destroyed due to combustion effect. Accordingly, the central flame does not exist. It may be attributed from the fact that hot gas is suddenly expanded outward and thus its outward momentum can overcome the adverse pressure gradient resulting from the central vortex.

4. The presence of chemical reaction can affect the reversed flow structure.

REFERENCES


### Table 1: Inlet Flow Conditions

<table>
<thead>
<tr>
<th>Vane Angle (°)</th>
<th>Axial Velocity (m/s)</th>
<th>Fuel/Air Ratio</th>
<th>Inlet Reynolds Number</th>
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<tr>
<td>0°</td>
<td>Fuel=26.17</td>
<td>0.0635</td>
<td>16030</td>
</tr>
<tr>
<td>45°</td>
<td>Fuel=32.8</td>
<td>0.0625</td>
<td>20460</td>
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</table>

### Table 2: Fuel and Air Properties

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Town Gas</th>
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<tr>
<td>Fuel Molecular Weight</td>
<td>1.78g/mole</td>
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<tr>
<td>(AFN) 1</td>
<td>10.59</td>
</tr>
<tr>
<td>Heating Value</td>
<td>2.85 ± 10^-4 J/kg</td>
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<tr>
<td>Inlet T/A</td>
<td>0.0435 &amp; 0.0433</td>
</tr>
<tr>
<td>Air Temperature</td>
<td>350 K</td>
</tr>
<tr>
<td>Fuel Temperature</td>
<td>350 K</td>
</tr>
<tr>
<td>Air Velocity</td>
<td>2.83X10^-4 m/s</td>
</tr>
<tr>
<td>Inlet Pressure</td>
<td>1 atm</td>
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</tbody>
</table>

Figure 1A The geometry of wall-fired swirl flame furnace.
Figure 1B Scale of geometric models

Figure 2 Comparison of predicted and measured mixture fraction for Lockwood case, zero swirl.

Figure 3 Comparison of the predicted and measured mixture fraction, zero swirl.

Figure 4 Comparison of predicted and measured mixture fraction contour, zero swirl, k-ε model.

Figure 5 Comparison of predicted and measured mixture fraction for Lockwood case, 45 deg. swirl.
Figure 6 Comparison of predicted and measured mixture fraction contour, 45 deg. swirl, RSM model.

Figure 7 Comparison of predicted and measured mixture fraction contour, 45 deg. swirl, k-e model.

Figure 8 Mixture fraction contour plot and temperature distribution for cylindrical furnace, F/A=0.0635, zero swirl, k-e model.

Figure 9 Mixture fraction contour plot and temperature distribution for cylindrical furnace, F/A=0.0623, 45 deg. swirl, k-e model.
Figure 10  Computational mesh of four burner system.

Figure 11  Velocity and reversed flow contour plot at various X/D locations, $\phi=0.1$ and 45 deg. swirl.

Figure 12  Temperature distribution at various X/D locations, $\phi=0.1$ and 45 deg. swirl.
Figure 13  Sectional reversed flow contour, $\phi=0.1$, 45 deg. swirl.

Figure 14  Sectional mixture fraction contour, $\phi=0.1$ and 45 deg. swirl.

Figure 15  Sectional temperature plot, $\phi=0.1$ and 45 deg. swirl.