EXPERIMENTS IN TURBULENT FLAMES: FROM INDUSTRIAL TO LABORATORY SCALE

by

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This paper reviews measurements obtained in turbulent flames and the consequent understanding on the fundamental processes of the flow, mixing and combustion in turbulent reacting flows. The flames considered range from large industrial flames in production glass furnaces and boilers to model-scale experiments, which are the result of a combined research effort in order to accomplish with the need of cleaner combustion strategies for the sustainable development.

Introduction

The current challenge of developing economic wealth, while protecting the natural resources, has stimulated the need for improved and novel technologies for generating and using energy and is demanding ever greater attention from government and corporations world-wide. In fact, environmental concerns shape current regulatory policies and critically influence every aspect of industrial production processes, along which the combustion of fuels and the consequent pollutant emissions are critical factors (*e.g.* Culick, Heitor and Whitelaw, 1996). To help minimizing their effects, there is the need for combined research efforts, ranging from large-scale to model-scale experiments and numerical simulations of the complex aerodynamics, chemical and thermodynamic processes occurring inside typical combustion furnaces. This paper contributes to achieve this objective, in that it reviews detailed measurements of the fluid dynamics and heat transfer characteristics obtained in large utility boilers and glass production furnaces, together with sample results obtained in large utility boilers and glass production furnaces, together with sample results obtained in laboratory arrangements.

Current requirements of reduced emissions and improved efficiency of industrial burning equipment, together with the increased use of low-quality residual fuels, has emphasized during the last decade the need for accurate methods of analyzing furnace performance and heat transfer as discussed, for example by Correia *et al.* (1996). Detailed knowledge of the combustion characteristics of the burning systems is, therefore, required to attempt to optimize current operating conditions and, also, to allow to validate and refine calculation methods (*e.g.* Jones and Kakhi, 1996). In this context, this paper considers experiments conduced in residual fuel-oil fired industrial plants, which are complemented by the implementation of advanced sensors in order to improve current control and monitoring of furnace operation (*e.g.* Heitor, 1991).

The analysis and modifications of existing burners have traditionally been based on empirical indices and correlations derived from full-scale performance, but in order to predict their effects there is a requirement for improved understanding of all the aerothermochemical phenomena building up the global combustion and heat transfer processes. This greatly depends on the availability of detailed and accurate measurements at full scale, which in turn are sparsely reported in the literature, see for example, the works of Robinson (1985), Boyd and Kent (1986), Bonin and Queiroz (1991), Buttler and Webb (1991), Heitor *et al.* (1994) or Cassiano *et al.* (1994-a, b).

On the other hand, comprehensive computer models are now emerging which can provide the combustion industry with a powerful new design tool and a method to quantify the effects of design modifications (*e.g.* Facchiano, 1990). However, the sub-models incorporated in those codes to simulate gas-side processes – such as mixing, radiative heat transfer or chemical kinetics (*e.g.* Robinson, 1985; Boyd and Kent, 1986; Lookwood *et al.*, 1988; Carvalho and Coelho, 1990) – are critical to the accuracy of the overall calculation scheme, but their capabilities depend on the validation against appropriate experimental measurements.

It should be noted that measurements of heat transfer characteristics of industrial furnaces and boilers are very expensive and limited by the geometry, time, number of instruments and skills required. The few works reported in the literature are concerned essentially with plants typically below 80 MWe (*e.g.* Wall and Stewart, 1975; Abraham and Raian, 1983; Robinson, 1985. On the other hand, the results are subject to inaccuracy (*e.g.* Boyd and Kent, 1986; Bonin and Queiroz, 1991), but provide useful information to improve and predict the performance of the combustion chambers.

Recent use of advanced instrumentation (*e.g.* Durao *et al.*, 1992; Taylor, 1993) in comparatively simple model burners (*e.g.* Mao *et al.*, 1986; McDonnel and Samuelsen, 1988; Durao *et al.*, 1994; Hardalupas *et al.*, 1990. 1994), have shown that an important effect of the related typical swirl flow is to centrifuge the fuel droplets to large radii and, therefore, to cause local regions of high fuel concentration and turbulence suppression. This has important consequences for practical burners since the fuel tends to be removed from the flame zone and turbulent fluctuations assist mixing controlled combustion. A compromise is required between the need for swirl to stabilize the flame and to increase the residence time through the formation of a recilculation zone, and the need to avoid centrifuging of fuel away from the flame, (*e.g.*, Liu *et al.*, 1989). In addition, local droplet concentration, gas equivalence ratio, degree of premixdness and strain rate of the reaction zone determine flame emissions, extinction, length, heat release pattern and combustion efficiency (*e.g.* Bradley *et al.*, 1992, Hardalupas *et al.*, 1994). It is important (*e.g.* for modeling purposes) that the physical understanding is available individually for each process involved, and here we review a detailed analysis of the combustion characteristics and mixing of disc- and swirl-stabilized flames.

FULL-SCALE COMBUSTION EXPERIMENTS

This section reviews experiments performed by the authors and co-workers in four different oil-fired industrial combustion chambers, namely two utility boilers of 250 MWe and 35 MWe, respectively, and two glass production furnaces of 90 t/day of soda-lime glass containers and of 40 t/day of tableware glass, respectively. The various combustion chambers are described and sample results are presented.

Combustion measurements in a 250 MWe utility boiler

Boiler geometry and operating conditions

The utility boiler used throughout this work is a subcritical, natural circulation, front wall oil-fired boiler, with a pressurized combustion chamber and a divided convection chamber where steam reheating and primary steam superheating takes place (see Cassiano *et al.*, 1994-b, for details). The boiler makes part of the Setubal's power plant of the Portuguese Electricity Generating Board. The furnace dimensions are approximately 19.9 m height (up to nose level), 11.4 m width and 8.5 m depth. It is fired with fuel oil by 12 burners arranged in an array of 4 burners in 3 different levels. The burners consist of a central tube through which the fuel-oil is pulverized into the furnace at the burner throat. Secondary air (*i.e.*, the main combustion air) flows from the windbox through air registers and enters the furnace through an annulus formed between the oil nozzle and the burner throat (0.9 m diameter). The residual fuel oil is supplied at a rate of 56 t/h with 5 % excess air.

The fuel oil includes 11% of asphaltenes and 220 ppm of vanadium (mass basis), and is out of the design specifications for the boiler.

Experimental method and results

Measurements of total incident wall heat flux, together with those of mean gas temperature and major gas species, were obtained through the ports and sample results are briefly analyzed here. A detailed description of the experimental techniques and of their accuracy are included in a review by Heitor and Moreira (1993).

Gas temperatures were measured using a 3.5 m long, double-shielded, water-cooled suction pyrometer, which consists of a Pt/Pt–13% Rh thermocouple made from 350 μm diameter wires, which are protected by a system of radiation shields heat losses to the wall chamber. The flow rate through the pyrometer could be varied so that the measurements presented here could be obtained independently of the suction velocity (*e.g.* Heitor and Moreira, 1993).

Major gas species concentrations were obtained by a gas sampling probe consisting of a sampling tube of 2 mm diameter mounted in a water cooled jacket of 50 mm diameter and 3.5 m length. The probe is connected to dedicated gas analyzers by a sampling system, which comprises a water cooled condenser, a diaphragm pump, a calcium-chloride drier and a cotton wool filter. Dry analysis of O_2 were made with a paramagnetic analyzer, while those of CO_2, CO and NO_x were made by infrared analyzers. The analysis are precise within 0.5% of full scale, but larger errors are likely to result from the sampling process.

A total heat flux meter was designed manufactured and used to measure the convective and radiative heat flux incident on the walls from a solid angle of 2π steradians up to a maximum of 500 kW m^{-2}, in the way suggested by Chedaille and Braud (1972). It consists of a water-cooled probe which houses a cylindrical, the front surface of which is serrated and blackned to provide high radiation absorptivity. The probe was calibrated making use of a purpose built black body furnace, which was periodically used between consecutive measurements. The response time of the meter is of the order of three minutes and, based on the repeatability of the results, an accuracy of ±5% can be claimed for the measured values of heat flux.

Figure 1 shows the distribution of total incident wall heat flux along the side and rear walls of the boiler, for nominal net electrical generation (*i.e.*, 250 MWe). The values varied between 200 kW m^{-2} at port 1.6 and 316 kW m^{-2} at port 3.4, with values measured on the rear wall greater than those on the side walls by 10%. The distribution is asymptotic with the values measured on the right side greater than those on the left side of the boiler by up to 22%.

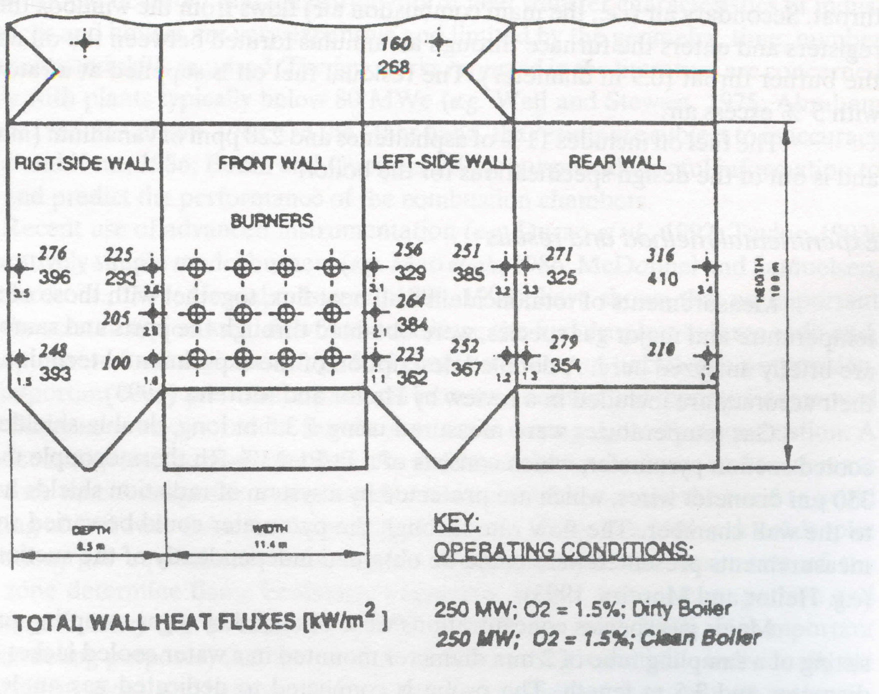

Figure 1. Distribution of total incident wall heat fluxes for two different wall conditions

The results of Fig. 1 include the analysis of the effect of the wall conditions on the heat transfer characteristics of the boiler. Typical incident heat fluxes for dirty walls are in general 20% higher than those obtained just after the boiler has been cleaned.

Figure 2 presents mean temperature profiles obtained through different ports in the boiler and shows that, with the exception of the near burner zone, the temperature of the combustion gases is uniform at each level with values in the range 1500–1600 °C.

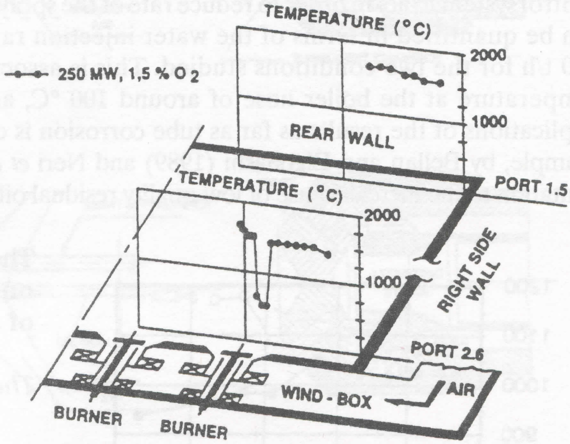

Figure 2. Mean gas temperature profiles obtained in the utility boiler through the inspection ports

The results in Fig. 3 show near uniform gaseous species concentration across the boiler with the expected concentrations of CO_2 and O_2 at the boiler nose. However, excess CO is observed around the rear wall and close to bottom, which is likely to be related with the inadequacy of the present furnace geometry for the fuel and operating conditions being used.

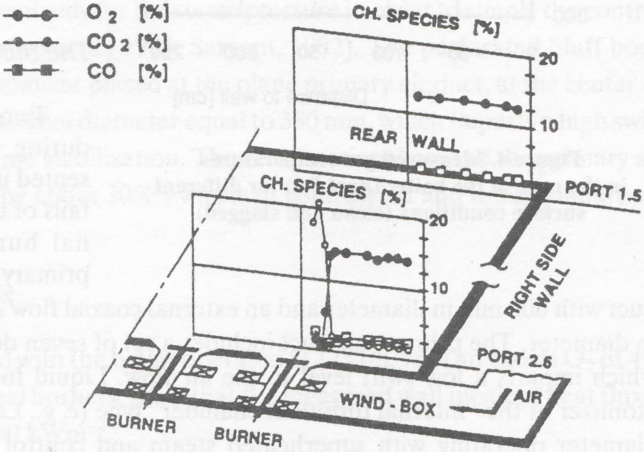

Figure 3. Profiles of mean concentrations of major species measured in the boiler through the inspection ports

The influence of the wall conditions on the boiler performance

The analysis of the previous section shows that the use of residual fuel-oils strongly affects the wall conditions, changing the thermal characteristics of the boilers (see also, Lawn, 1987). A constant superheat temperature is required and is achieved in the present installation by spraying water to attemperate the superheated steam. Slagging of the walls decreases heat transfer rates and thus, of the superheated steam, making the control system to act in order to reduce rate of the spray water flow. The extent of dirtiness can be quantified in terms of the water injection rate, which varied between 76 and 110 t/h for the two conditions studied. This is associated with an increase in the gas temperature at the boiler nose of around 100 °C, as quantified in Fig. 4. The direct implications of the results as far as tube corrosion is concerned have been discussed, for example, by Bellan and Elgobashi (1989) and Neri *et al.* (1991), and represent a serious limitation to the increased use of low-quality residual oils.

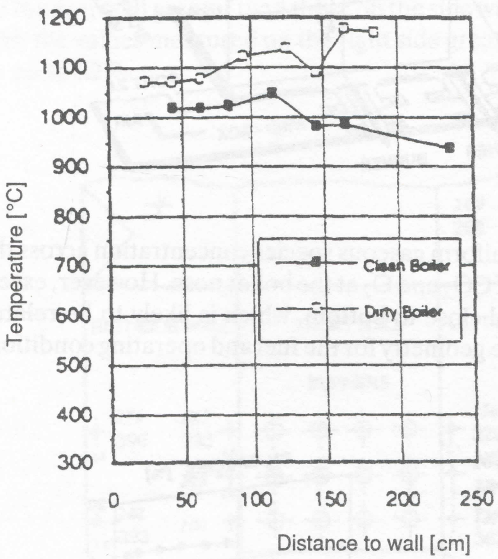

Figure 4. Measured gas temperatures in the nose of the boiler (port 5.1) for different surface conditions (clean and slagged)

The influence of burner design on the performance of a 35 MWe utility boiler

The utility boiler used

Measurements similar to those reported above were also taken in one of the boilers of the Barreiro's power plant of the Portuguese Electricity Generating Board, but with the main objective of studying the influence of burner design on the performance of the boilers (see details in Heitor *et al.*, 1994).

The burners

Two different burners were used during the experiments, as represented in Fig. 5 together with the details of the fuel atomizers. The original burner, Fig. 5a), includes a primary air flow through an annular duct with 600 mm in diameter and an external coaxial flow of secondary air with 700 mm in diameter. The primary air duct includes a set of seven deflectors to guide the air flux which imparts a low swirl level to the air flow. Liquid fuel is sprayed by a twin fluid atomizer of the "internal turbulent chamber" type (*e. g.*, Lefevbre, 1989) with 38 mm in diameter operating with superheated steam and control of the differential pressure

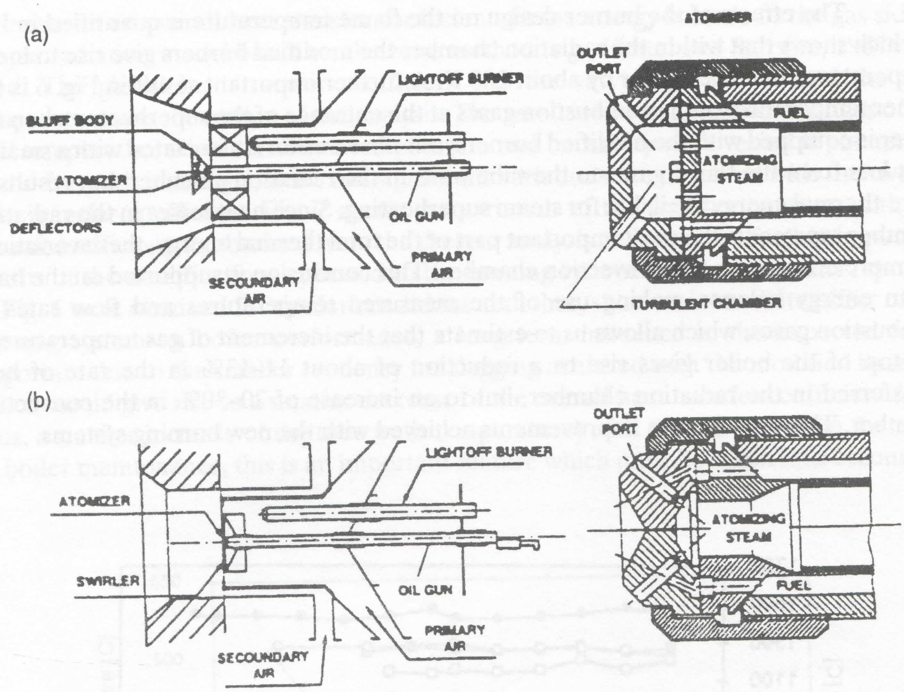

Figure 5. Burner designs considered throughout the experiments in a 350 MWe utility boiler
(a) original burner; (b) modified burner

achieved by a pneumatic control system. The atomizer is located downstream of the burner mouth at the center of a perforated conical flame holder.

For the modified burners, Fig. 5b), the atomizer was replaced by an Y-type atomizer with electronic control only for the steam pressure in order to simply the control of the fuel flow rate (*e.g.*, Bruce *et al.*, 1978; Sargent, 1982). The perforated bluff body has been removed and the atomizer placed at the plane primary air duct, at the center of a set of fixed vanes with an external diameter equal to 380 mm, which imparts a high swirl level in order to promote flame stabilization. The cross sectional area of the primary air duct has also been reduced by about 30% by a small contraction and the secondary air has remained unchanged.

Sample results and analysis

The results obtained with the boiler operated at 125 t/h, with an excess O_2 of 4% and equipped with the original burners, show that the measured wall incident heat fluxes range from 240 kWm^{-2} to 360 kWm^{-2}.

The effects of the burner design on the flame temperature is quantified in Fig. 6, which shows that within the radiation chamber the modified burners give rise to mean temperatures that are higher by about 200 K. A further important result in Fig. 6 is the higher temperature of the combustion gases at the entrance of the superheater when the boiler is equipped with the modified burners; this observation is associated with a smaller heat loss from the flame gases to the tube walls in the radiation chamber, but results in more thermal energy available for steam superheating. Since heat losses in the radiation chamber represents the most important part of the total thermal energy, these variations are more effective in the convection chamber. This conclusion is supported on the basis of an energy balance making use of the measured temperatures and flow rates of combustion gases, which allows us to estimate that the increment of gas temperature at the top of the boiler gives rise to a reduction of about 11–13% in the rate of heat transferred in the radiation chamber, but to an increase of 20–30% in the convection chamber. This explains the improvements achieved with the new burning systems.

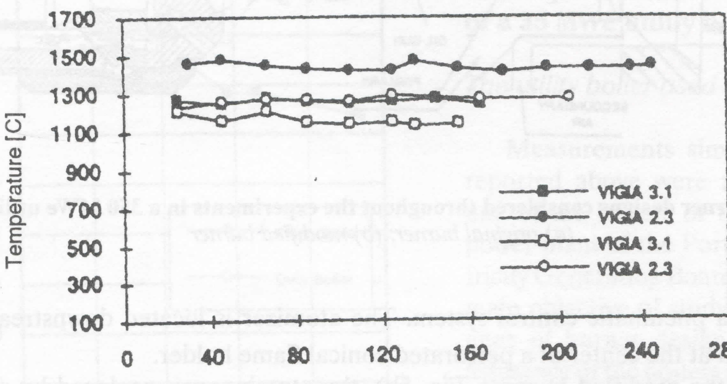

Figure 6. Effects of burner modifications on mean temperature within the radiation chamber (125 th^{-1}; 4% excess O_2)

Also, since radiation is the dominant mode of heat-transfer in the combustion chamber and this, in turn, is due mainly to particulate emission, the present results further suggest that a decrease in flame emissivity also occurs with the modified burners, which can be associated with a decrease in the concentration of combustion-generated particles.

The magnitude of that decrease can be estimated from overall energy balances, based on the experimental results and from knowing the relation between the particles emissivity and their concentration in the combustion gases (*e.g.*, Richter *et al.*, 1979).

The sensitivity of the boiler performance to the thermodynamic variables was considered by closing the heat balance to the gas side with that absorbed by the steam, by means of an overall heat transfer coefficient. In the energy balance to the gas side, a simple radiation model based on the zone method (*e.g.*, Hottel and Sarofim, 1967) was used. The global heat transfer coefficient was the varying parameter and the wall temperature was the output of the analysis. The results showed negligible changes of the wall temperature when the flame temperature and emissivity were varied in the ranges found for each burning system, but significant variations are observed with the overall heat transfer coefficient. Fig. 7 shows the variation of the wall temperature with the overall coefficient, as obtained from the present analysis and suggest a further important results of the modifications operated to the burning systems: while the results reported in Fig. 6 were obtained for similar conditions and, thus, show an immediate effect of the burning system on boiler performance, a long term influence is also expected to be important due to the smaller tendency for slagging achieved with the modified burners, which would give rise to a smaller decrease of the overall heat transfer coefficient (and, thus, a smaller increase of the tube walls temperature) with time of operation. In terms of boiler maintenance, this is an important feature which must be taken into account.

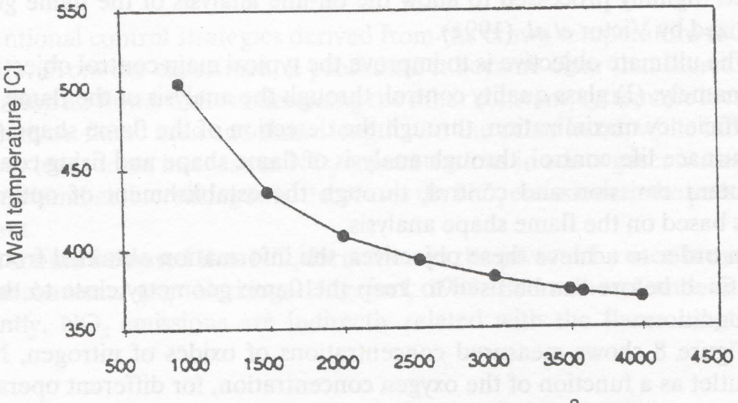

Figure 7. Effects of the overall heat-transfer coefficient on wall temperature

Combustion tests in glass-melting furnaces

Local gas temperatures, local concentrations of major gas species and incident wall heat fluxes were measured in a large end-port, fuel-fired regenerative furnace for manufacture of glass containers, with a nominal output of 90 t/day of soda-lime glass and a specific energy consumption of 5.9 GJ/t, (see details in Cassiano *et al.*, 1994-a).

The local wall heat fluxes range from 250 to 300 kW m^{-2}, with maximum variations of 20% along each wall. The maximum temperatures in the flame and in the exhaust gases were 1600 and 1540 °C, respectively, the latter associated with 2% excess O_2. The maximum measured volume concentrations of carbon monoxide varied between 5% in the flame and 0.5% at the exit of the combustion chamber. The nitrogen oxide levels were ≈ 4000 ppmv. The results have confirmed the need to develop new sensing and control methodologies to accomplish with the protection of both the environment and natural resources, as well as the increasingly tighter legislation of NO_x emission below 5000 mg/Nm^3 (*e.g.* Barklage-Hilgefort, 1989; Flamme *et al.*, 1995).

Furnace monitoring

In order to contribute to achieve these objectives, an advanced furnace viewing sensor was developed and tested in a regenerative, U-flamed, glass furnace with a pull up to 40 t/day of tableware glass, (see details in Duarte *et al.*, 1996). The furnace environment was analyzed making use of a conventional black and white CCD camera located at the front wall, in order to allow an upper view of the flame. The output of the camera was digitally processed to allow the on-line analysis of the flame geometry, as firstly realized by Victor *et al.* (1991).

The ultimate objective is to improve the typical main control objectives in glass furnaces, namely: (I) glass quality control, through the analysis of the flame length: (II) thermal efficiency maximization, through the detection of the flame shape,(III) refractory and furnace life control, through analysis of flame shape and firing conditions and (IV) pollutant emission and control, through the establishment of optimized firing conditions based on the flame shape analysis.

In order to achieve these objectives, the information obtained from the vision system defined before can be used to keep the flame geometry close to the optimum burning conditions.

Figure 8 shows measured concentrations of oxides of nitrogen, NO_x, at the furnace outlet as a function of the oxygen concentration, for different operating conditions corresponding to four different flame shapes as obtained by adjusting the swirl – and axial – air flow rates in the burner. The results have been normalized to 8% excess oxygen level in the exhaust stream and confirm the expected trend of NO_x reduction as the excess air is decreased. In addition, the analysis suggests that reductions in NO_x up to 20% can be achieved by proper adjustment of the burner operating parameters.

The results allow to establish optimized criteria to control process variables making use of the information derived from the flame viewing system and Fig. 9 exemplifies a possible control scheme.

In general, two categories of variables must be considered, namely: short-term control variables, including firing rate, air flow rate and batch feed rate; and long-term controlling parameters, such as batch preparation, fuel type and quality, and furnace condition.

In this context, the system of Fig. 9 will allow to integrate the control of the atomization air, making use of the information derived from the flame viewing system,

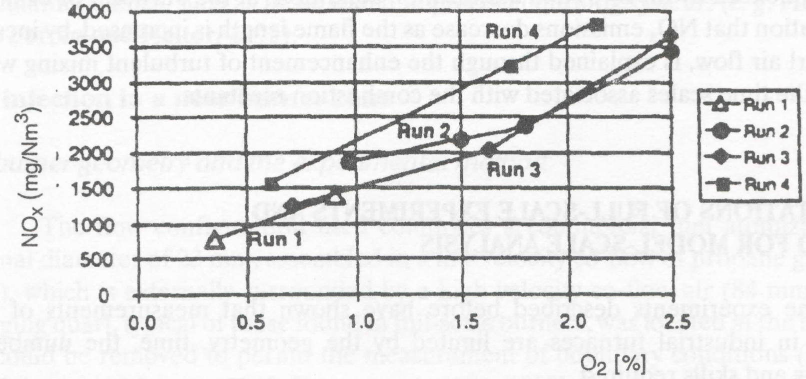

Figure 8. NO_x emmission as a function of burner operating parameters and excess oxygen levels measured at the upper chamber of the exhaust generator in the glass furnace of Duarte *et all.* (1996)

with conventional control strategies derived from the crown temperature to control the air-fuel ratio. From the experimental procedure it became clear that increasing one of the atomization air flows, while maintaining the other constant, systematic increases were obtained both on flame area and flame length. On the other hand, while increasing the axial air flow leads to an increase in NO_x concentrations in waste gases at the exit of the combustion chamber, increasing swirl decreases the NO_x emissions in the present furnace arrangement.

It should be noted that NO_x formation in the present furnace is mainly due to thermal mechanisms (*e.g.* Barklage-Hilgefort, 1989; Schmalhorst and Ernas, 1995). Consequently, NO_x emissions are indirectly related with the flame length, which is

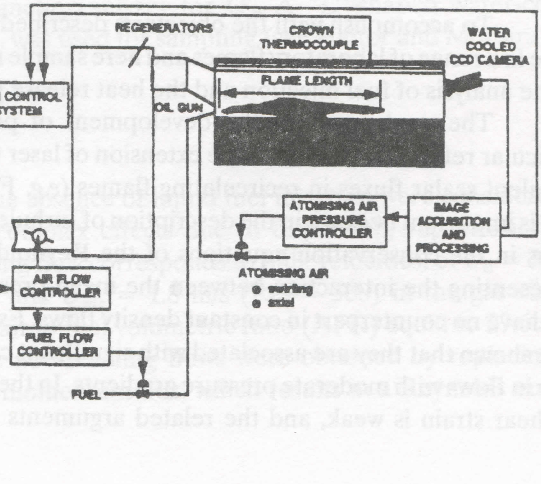

Figure 9. Schematic diagram of an integrated control system making use of a furnace viewing system

strongly influenced by the swirl and atomization conditions at the burner. For example, the observation that NO_x emissions decrease as the flame length is increased, by increasing the swirl air flow, is explained through the enhancement of turbulent mixing which decreases the time scales associated with the combustion reactants.

THE LIMITATIONS OF FULL-SCALE EXPERIMENTS AND THE NEED FOR MODEL-SCALE ANALYSIS

The experiments described before have shown that measurements of flow properties in industrial furnaces are limited by the geometry, time, the number of instruments and skills required.

Besides the difficulty in obtaining results, there is the high cost of tests on production furnaces, and high costs also make extensive plant modifications prohibitive. Computer codes incorporating adequate physical models are, therefore, required to perform, evaluate and test prospective furnace designs. However these codes have to be validated against experimental results.

The analysis above shows that the performance of industrial burning systems, namely quantified in terms of flame emissions, extinction, length, heat release distribution and combustion efficiency, is influenced by the interaction between basic phenomena, such as local droplet/particulate concentration, gas equivalence ratio, degree of premixedness and strain rate of the reaction zone.

There is, therefore, a need for laboratory scale experiments to determine the extent to which can be achieved and lead to reduced pollutant emissions with improved flammability limits, as described in the section below.

MODEL-SCALE EXPERIMENTS: SAMPLE RESULTS

To accomplish with the objectives described before, experimental studies were made in a range of laboratory flames and here sample results are presented with emphasis on the analysis of fuel injection and the heat release pattern.

The work involved the development of probe and optical techniques, and particular reference is made to the extension of laser velocimetry to the measurement of turbulent scalar fluxes in recirculating flames (*e.g.* Ferrao and Heitor, 1996-a and -b). This is important because in the description of turbulent reacting flows there arise source terms in the conservation equations of the Reynolds stresses and of the scalar fluxes representing the interaction between the mean pressure field and density fluctuations that have no counterpart in constant density flows. Estimates of these terms in jet flames have shown that they are associated with significant changes in the turbulence structure, even in flows with moderate pressure gradients. In these flames, the turbulent generation by shear strain is weak, and the related arguments may not hold in strongly sheared, recirculating flames, such as those found in practical burning systems (*e. g.* Heitor *et al.*, 1987; Ferrao and Heitor, 1995).

Fuel injection in a near burner zone

The burner geometry and the experimental method

The flow configuration used comprises a commercial fuel atomizer with an external diameter of 23 mm, assembled in a low velocity co-flow of propane gas (54 mm O.D.), which is externally surrounded by a high velocity co-flow air (84 mm O.D.). A diverging quarl, typical of those found in full-scale burners, was located at the burner exit and could be removed to permit the measurement of boundary conditions (for details see Heitor and Moreira, 1992; Durao *et al.*, 1991, 1992). Swirl can be imparted to both streams by means of fixed blades at 45° with resulting swirl numbers, estimated from the geometry of the blades, equal to $S_o = 0.77$ and $S_i = 0.85$ for the outer and inner streams, respectively.

The burner was located vertically directed upwards and the symmetry of the flow was verified by measuring several complete radial profiles in the horizontal plane. The origin of the axial axis, x, is taken at the center of the exit plane of the model burner.

Velocity measurements were obtained with a dual-beam laser-Doppler anemometer, based on an argon-ion laser light source at 514.5 nm (1 W nominal) with sensitivity to the flow direction provided by light-frequency shifting from acoustic-optic modulation (double Bragg cells) with a resulting shift of the Doppler signal in the range 0–10 MHz. The half-angle between the beams was 4.29° and the calculated dimensions of the measuring volume at the e^{-2} intensity were 1.528 and 0.132 mm.

Detailed mean temperature measurements were obtained with bare-wire thermocouples fabricated from 0.080 mm diameter Pt/Pt: 13% Rh wires, which were supported on 0.500 mm diameter wires of the same material an a long "L"-shaped probe. Mean concentrations of major species were measured on a wet basis with a flame ionization detector for unburned hydrocarbons and on a dry basis with infrared analyzers for CO and CO_2 and a paramagnetic analyzer for O_2. A "L"-shaped water-cooled stainless-steel probe of 7 mm O.D. was used for sampling (see Heitor and Moreira 1993 for details).

Sample results and discussions

A single phase flows in the absence of liquid fuel injection were considered, in order to remove the additional complex effects due to dispersion, vaporization and droplet/gas interaction. The reacting flow corresponds to bulk velocities. of $U_o = 30$ m/s ($Re_o = 49500$) in the air stream and $U_{gas} = 1.8$ m/s ($Re_i = 300$) in the gas stream, corresponding to a flame with an air to fuel volumetric ratio (AFR) equal to 27.6 and a heat load of about 350 kW. The non-reacting flows were obtained by replacing the propane gas by air with the same momentum flux, which results in a Reynolds number equal to 4000 in the inner flow.

Figure 10 shows the measured streamline distributions and indicate the most salient features of the mean flow in the vicinity of the burner head for reacting and non-reacting conditions. The two have similar patterns with qualitatively similar distributions of mean velocity, which are typical of those observed in highly rotating flows.

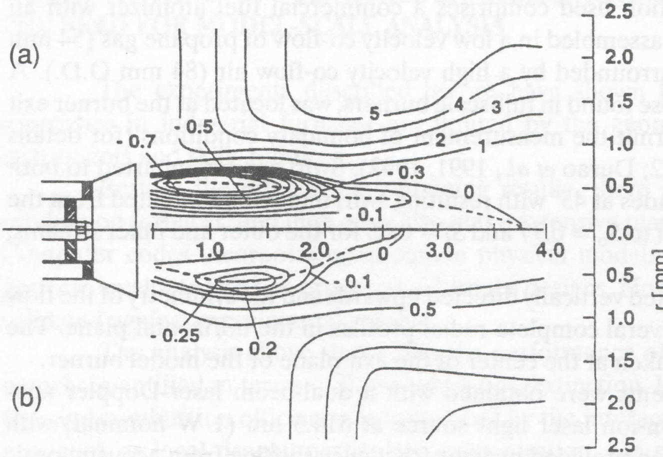

Figure 10. Measured streamline distributions
(a) non-reacting
(b) reacting flows,
AFR = 27, Re = 49500

Despite the qualitative similarities between the two flows, combustion induces significant quantitative differences: the mean axial and tangential velocities increase because the density is lowered and the axial and angular momentum must be conserved; the recirculated mass flow rate decreases from 67% in the non-reacting flow to 14% in the reacting flow; the length of the recirculation zone decreased to 32.5% and its maximum width increases by about 12%.

For reacting conditions the reverse flow zone is characterized by uniform concentrations of the major species and high mean temperatures, Fig. 11, as in other recirculating flames (*e.g.* Fernandes *et al.*, 1994). The stoichiometric line (*i.e.* f = 0.060) occurs for radii larger than those characterized by zero velocity and, consequently, the recirculation zone is fuel rich.

The reverse flow itself consists primarily of burnt exhaust gases in such a way that fuel chemical equilibrium is unlikely to provide a realistic description of its thermochemistry. The maximum values of CO and unburned hydrocarbons occur along the reacting shear layer for zones of near local stoichiometry.

Inspection of Fig. 12 shows that the general levels of velocity fluctuations for both reacting and non-reacting conditions are small inside the recirculation zone and large in the highly strained annular shear layer. In this region turbulence is strongly anisotropic with, Favre – averaged values $\widetilde{u'^2}_{max} = 1.33\widetilde{v'^2}_{max} = 2.0\widetilde{w'^2}_{max}$ in the reacting

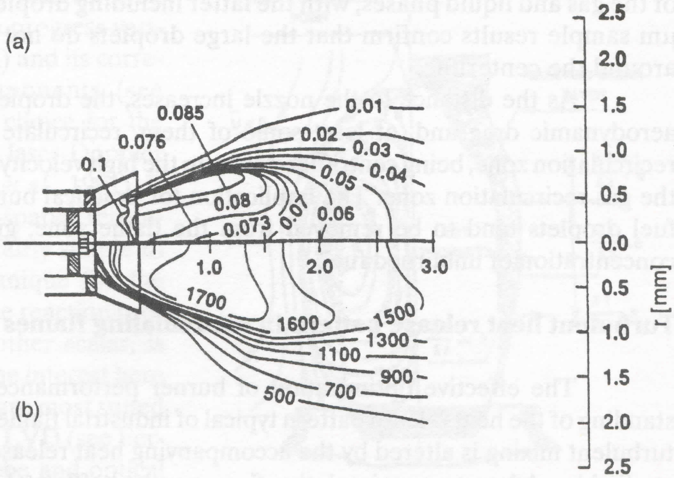

Figure 11. Distribution of measured scalar characteristics
(a) mixture fraction;
(b) gas temperature

flow, with $\widetilde{v''^2}$ and $\widetilde{w''^2}$ increasing considerably towards the stagnation point, in a way similar to that observed in highly recirculating flows downstream of baffles. The iso-contours of turbulent kinetic energy for the reacting flow show that maximum values occur in the vicinity of the rear stagnation point and analysis of the probability-density distributions suggests the presence of some flow periodicity.

We now turn to the analysis of the aerodynamics of the liquid droplets within the reverse flow zone analyzed before, which was carried out in the absence of combustion for ease of analysis. The measurements were obtained with the amplitude technique of Durao *et al.* (1992), which provides information about the velocity field

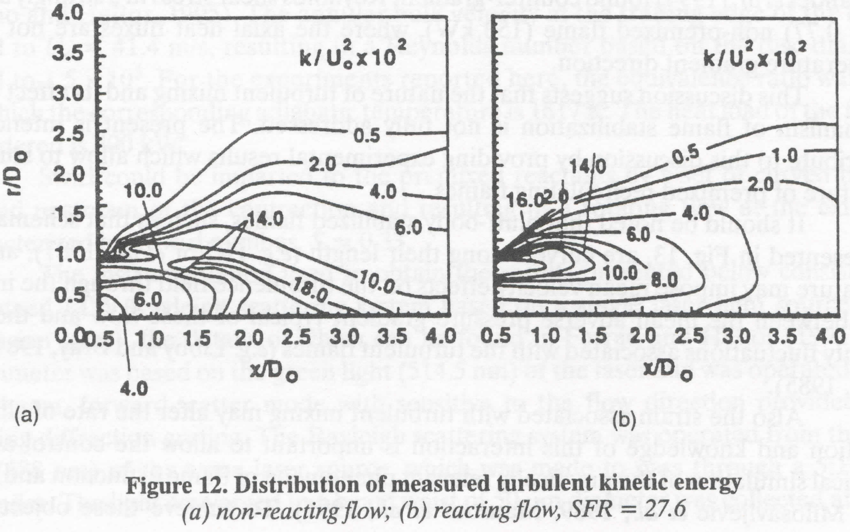

Figure 12. Distribution of measured turbulent kinetic energy
(a) non-reacting flow; (b) reacting flow, SFR = 27.6

of the gas and liquid phases, with the latter including droplet diameters larger than 40 μm sample results confirm that the large droplets do not follow the mean gas flow around the centerline.

As the distance to the nozzle increases, the droplets decelerate due to mean aerodynamic drag and, at least some of them, recirculate through the edges of the recirculation zone, being centrifuged away to the high velocity regions established around the gas recirculation zone. The implication for practical burning equipment is that the fuel droplets tend to be removed from the flame zone, giving rise to zones of high concentration of unburned fuel.

Turbulent heat release pattern in recirculating flames

The effective optimization of burner performance, requires a better understanding of the heat release pattern typical of industrial flames and of the extent to which turbulent mixing is altered by the accompanying heat release. This can be conveniently studied in a laboratory recirculating flames and a baffle-stabilized flows.

It should be noted that improved understanding of the turbulent mixing processes in combustion systems has been achieved in recent years with the use of advanced diagnostics for making spatially – and temporally – resolved measurements in flames of practical interest, Heitor *et al.* (1987), Ferrao and Heitor (1995), Duarte *et al.* (1995), Almeida *et al.* (1995).

The evidence of counter-gradient transport of heat and/or momentum has been demonstrated in swirling non-premixed flames. For example, Takagi *et al.* (1984) and Takagi & Okamoto (1987) have provided evidence of the existence of counter-gradient diffusion of heat in confined non-premixed flames at low swirl numbers, S.

These authors also found that the radial Reynolds stresses and the variance of temperature fluctuations were reduced in comparison to the unswirled flame. Also, Fernandes *et al.* (1994) found counter-gradient Reynolds shear stress in a strongly swirled ($S = 0.77$) non-premixed flame (150 kW), where the axial heat fluxes are not in the temperature gradient direction.

This discussion suggests that the nature of turbulent mixing and its effect on the mechanism of flame stabilization is not fully addressed. The present is intended to contribute to this discussion by providing experimental results which allow to study the structure of premixed recirculating flames.

It should be noted that bluff-body stabilized flames, such as that schematically represented in Fig. 13, are curved along their length (*e.g.* Heitor *et al.*, 1987), and this curvature may impose mean velocity effects on the turbulence field through the interaction between the mean adverse pressure gradient typical of these flow and the large density fluctuations associated with the turbulent flames (*e.g.* Libby and Bray, 1981: Bray *et al.*, 1985).

Also the strain associated with turbulent mixing may alter the rate of chemical reaction and knowledge of this interaction is important to allow the control and the physical simulation of the practically relevant phenomena of flame extinction and lift-off (*e.g.* Milosavljevic *et al.*, 1989; Mansour *et al.*, 1991). To achieve these objectives in premixed flames, the major interest is on the analysis of the reaction progress variable, $C = (T - T_o)/(T_{ad} - T_o)$ and its correlation with velocity components (see Bilger 1991). The obvious choice for the velocity measurements is a laser-Doppler velocimeter, LVD (Heitor *et al.*, 1991). It is non-intrusive, has good spatial resolution, is fast, accurate and fairly simple to use. The choice of a technique for the time-resolved analysis of the reaction progress variable, or of any other scalar, is however more varied and the interest here is on those methods which are most suited for simultaneous use with a LVD (see Ferrao and Heitor, 1992). Probe and optical techniques can be used and we selected thin digitally-compensated thermocouples and laser-Rayleigh scattering to achieve these objectives (Ferrao and Heitor, 1996). The simultaneous measurements are however difficult to implement.

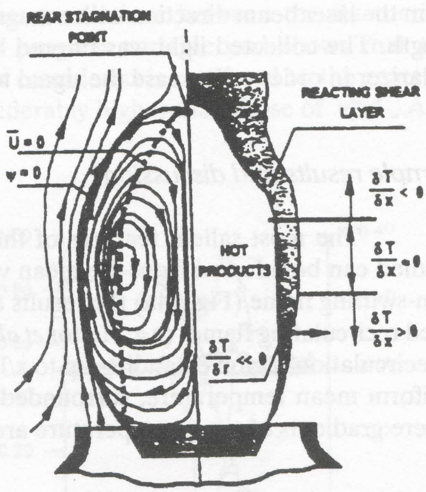

Figure 13. Schematic diagram of baffle-stabilized premixed flame

The recirculating flames studies

The experiments reviewed here conducted in an unconfined partially-premixed lean flame of air and propane stabilized on a disc with D = 56 mm in diameter, which is located at the exit section of a contraction with a diameter of 80 mm (for details, see Ferrao and Heitor, 1995). The annular bulk velocity at the trailing edge of the disc is equal to U_o = 41.4 m/s, resulting in a Reynolds number based on the disc diameter equal to 1.5×10^5. For the experiments reported here, the equivalence ratio was 0.55, for which the corresponding adiabatic temperature is 1617 K. The heat load of the flames considered is 200 kW.

Swirl could be imparted to the premixed reactants by a set of curved blades, located upstream of the contraction and resulting in a rotating flow at the exit duct characterized by a swirl number, S = 0.33.

The instrumentation used to obtain the results presented below consists in a combined LDV/Rayleigh scattering system based on a single laser light source (5 W argon-ion laser), the details of which can be found in Ferrao and Heitor (1995). The velocimeter was based on the green light (514.5 nm) of the laser and was operated in the dual-beam, forward-scatter mode with sensitive to the flow direction provided by a rotating diffraction grating. The Rayleigh scattering system was operated from the blue line (488 nm) of the same laser source, which was made to pass through a 5:1 beam expander. The light converged in a beam waist of 50 μm diameter was collected at 90 °C from the laser beam direction with a magnification of 1 and passed through a slit of 1 mm length. The collected light was filtered by a 1 nm bandwidth interference filter and a polarizer in order to increase the signal to noise ratio.

Sample results and discussion

The most salient features of the mean flow characteristics of the two flames studied can be inferred from the mean velocity vectors represented in Fig. 14. For the non-swirling flame (Fig. 14a) the results are similar to those found in other baffle-stabilized recirculating flames (*e.g.* Heitor *et al.*, Ferrao and Heitor, 1995), in that they exhibit a recirculation region extending up to x/D = 2.21, where the fluid has a large and fairly uniform mean temperature, surrounded by an annular region of highly sheared fluid where gradients of mean temperature are large.

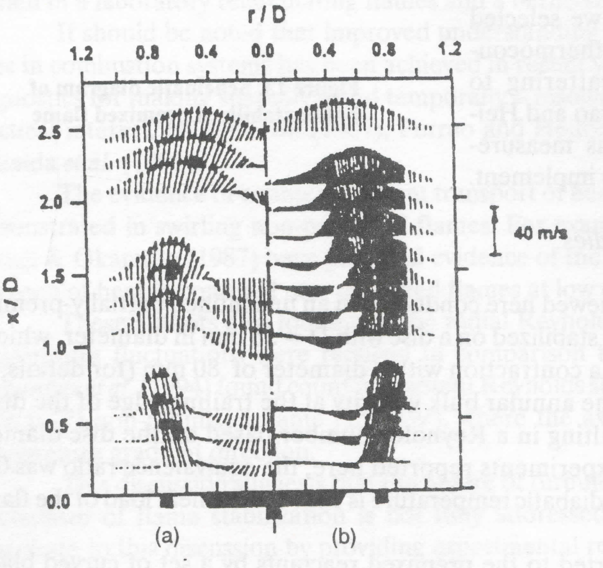

Figure 14. Distribution of mean velocity vectors along the vertical plane of symmetry
(a) non-swirling flame;
(b) swirling flame

The single recirculation zone of the unswirled flame is to be contrasted to that of the swirling flame, (Fig. 14b), which is shorter, wider and annular in shape because it includes an inner annular vortex with positive mean axial velocities along the centerline.

The inner recirculation zone is associated with positive mean axial velocities along the centerline up to the first stagnation point and rotates in the opposite sense to the outer recirculation zone.

Figure 15 shows radial profiles of the correlations between the fluctuation velocity characteristics and those of reaction progress variable and shows that the turbulent heat transfer rate for the two flames considered is restricted to the reacting shear layers, with absolute values of $\widetilde{u''c''}$ considerably higher than those of $\widetilde{v''c''}$. As a

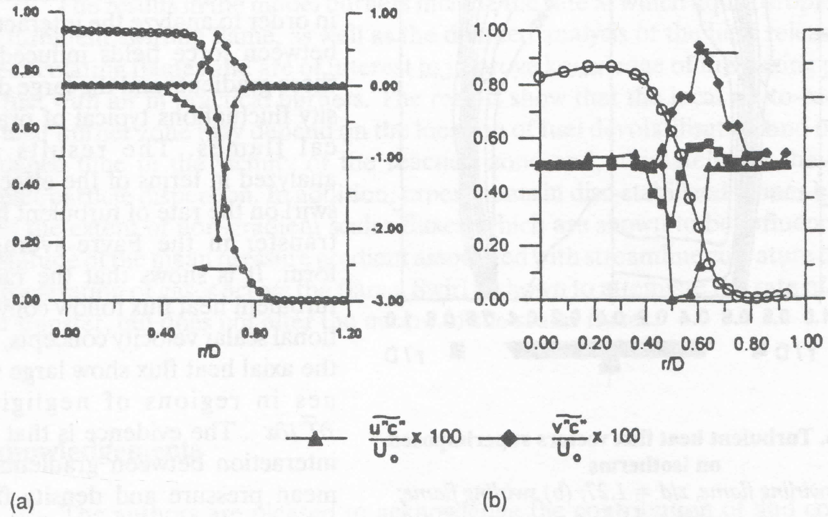

Figure 15. Axial and radial turbulent heat fluxes
(a) non-swirling flame (x/d = 1.27); (b) swirling flame (x/d = 1.21)

consequence, Fig. 16 shows that a large component of the vectors of turbulent heat transfer is directed along the isotherms, rather than normal to these, as would be expected from gradient-transport models of the kind used in non-reacting flows (*e. g.* Libby and Williams, 1995; Jones and Kakhi, 1995).

These quantities represent the exchange rate of reactants responsible for the phenomenon of flame stabilization and the results show that swirl decrease their magnitude due to the attenuation of the mean temperature gradients across the reacting shear layer.

The results for the non-swirling flame have been explained before in terms of the interaction between gradients of mean pressure and density fluctuations (*e.g.* Heitor *et al.*, 1987), which are important in the process of the turbulent transport typical of reacting flows. The present results provide new evidence that this interaction is affected by the degree of swirl imposed on the flows. In general, the results confirm that predictions of these kind of flames must be based on second moment, rather than on viscosity, turbulence model closures so as to capture the effects of the mean pressure gradient on heat fluxes.

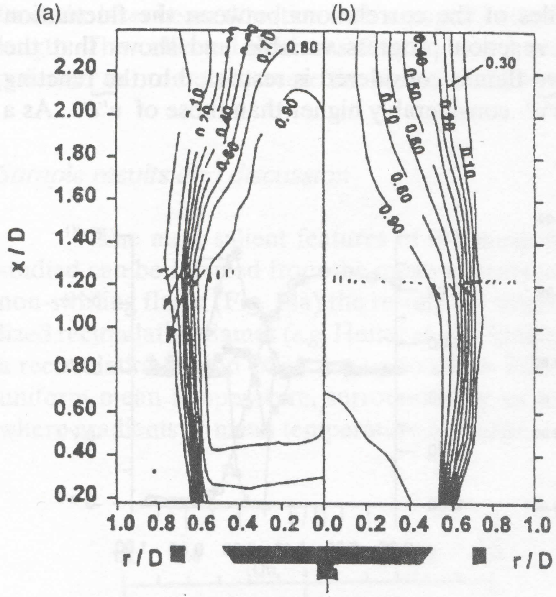

Figure 16. Turbulent heat flux vectors superimposed on isotherms
(a) non swirling flame, z/d = 1.27; (b) swirling flame; x/d = 1.21

Discussion

The measurements reported above have been used to obtain magnitude estimates of the mean pressure gradients from the conservation equations of momentum in order to analyze the interaction between force fields induced by these gradients and the large density fluctuations typical of practical flames. The results are analyzed in terms of the effect of swirl on the rate of turbulent heat transfer in the Favre-averaged form. It is shows that the radial turbulent heat flux follow conventional scalar velocity concepts, but the axial heat flux show large values in regions of negligible $\partial \overline{T} / \partial x$. The evidence is that the interaction between gradients of mean pressure and density fluctuations are important in the conservation of the fluxes, with comparatively large contributions in swirling flames. Swirl is also shown to induce large gradients of the radial mean velocity component, which may influence the conservation of the turbulent shear stress $\widetilde{u'v'}$.

One of the implications of these results to the modeling of reacting flows with practical relevance is that it is necessary to represent directly those processes arising from variable density. The need to avoid the use of gradient hypothesis, which requires the development of models at the level of the transport equations for stresses fluxes, depends on the particular flow geometry under analysis, and the present results show that swirl brings additional difficulties to the prediction of practical flames.

CONCLUSION

Measurements of the incident wall heat fluxes and local flame temperatures and major species concentrations obtained in large industrial utility boilers and glass production furnaces are reviewed and discussed together with results obtained in model burners. The work is aimed at improving knowledge of the thermal processes involved in large-scale oil-fired combustion systems and quantify the extent to which combustion is limited by chemical kinetics, as well as pollutant and particulate emissions are dependent on the fuel burning process. Attention is given to guide the development of low-NO_x combustion technologies, namely through the use of improved burner design and furnace control strategies. In this concern, the implementation of a flame viewing system in a glass production furnace is particularly described and shown to allow to reduce NO_x emissions by up to 20%.

The results in the model burners include the rate at which liquid droplets spread in a turbulent swirling flame, as well as the detailed analysis of the heat release pattern in recirculating flames and are of interest to improve knowledge of the mixing process of the fuel with air in practical burners. The results show that the local air-to-fuel ratio in the near burner zone may depend on the location of fuel devolatilisation and on the fuel residence time in the vicinity of the reaction zone and, ultimately, on the details of droplet/particle dispersion. In addition, experiments in disc-stabilized flames have quantified the extent of non-gradient scalar fluxes, which are shown to be influenced by the magnitude of the mean pressure gradient associated with streamline curvature or because of acceleration of gases across the flame. Swirl is shown to attenuate the rate of turbulent heat transfer, but does not alter the nature of the scalar fluxes.

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