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Imperial College, Mech. Eng. Dept.
KEROSENE SPRAY FLAMES

by

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B.Sc. (Eng), M.Sc.

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(Deptartment of Mechanical Engineering)

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ABSTRACT

Measurements of velocity, temperature and concentration of CO, CO₂, O₂ and UHC are presented for a range of confined and unconfined, preheated and unpreheated kerosene spray flames. The unconfined flames have been generated by a twin fluid atomizer and measurements have been obtained for two sprays with different flow rates of atomization air, and a consequent 10% difference in the mean Sauter diameter of the droplets. The reduction in the Sauter mean diameter is shown to result in an increase in the combustion intensity, particular in the near nozzle region which is influenced by increased evaporation rate due to reduction in the droplet lifetime. These flames have also been used extensively to quantify the performance and accuracy of temperature and species concentration measurements. The temperatures have been obtained by a suction pyrometer and bare-wire thermocouples of 40, 80 and 300 μm in diameter, and have allowed a detailed assessment of the precision of the temperature measurements. The uncertainties associated with the concentration measurements, due to quenching and non-isokinetic sampling, have been quantified.

In the confined flame experiments, thirteen flames have been studied corresponding to spray mean-droplet diameter of 54, 77 and 115 μm, combustion air swirl numbers of 1.4, 0.81 and 0.47 and air preheat temperatures of 125 and 225 °C. The axisymmetric combustor has been equipped with a rotating cup atomizer giving near mono-disperse diameters of droplet size. The results indicate that, for different swirl numbers, the increase of the mean-droplet diameter reduces the evaporation rate in the spray region and influences the flame structure and combustion intensity. The reduction in air swirl, for same mean-droplet diameter is shown to be
associated with reduced chemical reaction rate within the initial part of the flame and close to the centreline. The reduced combustion intensity in the central part of the flame was more significant for the lowest swirl (S = 0.47) and is related to the simultaneous reduction in the droplet evaporation and quenching of chemical reaction by the high velocity of the air stream surrounding the atomizer. The results obtained at different combustion air inlet-temperature, but with the same mean-droplet diameter and same combustion air swirl, have revealed that the preheat improved the combustion intensity and the flame performance especially for the large droplet diameter flame (D_m = 115 \mu m); this is expected due to the increased evaporation rate of the droplets. The results have implied that for the small droplet sprays and high levels of swirl, the contribution of the droplet evaporation to the combustion process is small and consequently the flame will be very similar to the turbulent gas diffusion flame.

The local properties of confined flames have been predicted and the results compared with the measurements. The mathematical formulation comprises Eulerian conservation equations for the gas phase and Lagrangian equations for the droplet motion and the thermal balance to the droplet size representing the spray. The latter is coupled with a droplet-tracking technique which allows the determination of droplet location and properties within the flow field. The spray combustion model assumes that the evaporating droplets act as distributed point-injectors of fuel vapour within the flame and that the combustion rate of fuel vapour is controlled by the turbulent mixing between the air and fuel streams. The experimental and predicted results are shown to be in qualitative agreement although with discrepancies which are related to the modelling assumptions.
ACKNOWLEDGEMENTS

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NOMENCLATURE

Roman characters

\( A \) Surface area of the thermocouple bead \((m^2)\)

\( A_s \) Cross-sectional area of the suction pyrometer \((m^2)\)

\( A_{sp} \) Cross-sectional area of the sampling probe \((m^2)\)

\( A_v \) Cross-sectional area of the LDA control volume, equation (2.31)

\( B \) Dimensionless transfer number, equation (4.33)

\( C \) Evaporation rate constant under forced convection, equations (3.11 and 4.31)

\( C_b \) Evaporation rate constant under stagnant conditions, equation (4.31)

\( C_p \) Specific heat under constant pressure \((J/Kg°K)\)

\( C_w \) Thermal conductivity of the thermocouple wire \((J/Kgm°K)\)

\( C_D \) Drag coefficient

\( C_1, C_2, C_\mu \) Turbulence model constants, table (4.2)

\( d_j \) Thermocouple junction diameter \((m)\)

\( D_p \) Diameter of the droplet \((m)\)

\( D_{sp} \) Inner diameter of the sampling probe \((m)\)

\( f \) Mixture fraction

\( G \) Acceleration of gravity, equation (3.1)

\( h \) Specific enthalpy \((J/Kg)\)

\( h_C \) Heat transfer coefficient of the combustion gases \((J/Kgm°K)\)

\( H_{fu} \) Fuel heat of reaction \((J/Kg)\)

\( i \) Stoichiometric oxygen to fuel ratio

\( I \) Total radiant intensity \((W/m^2)\)
$K_{R}$ Radiation absorption coefficient, equation (4.44)
$K_{r}$ Radiation view factor, equation (2.8)
k Kinetic energy of turbulence (m$^2$/s$^2$)
$L$ Fuel latent heat of vaporization (J/Kg)
$L'$ Macroscopic length scale (m)
m Mass fraction
$m_w$ Mass of the thermocouple (Kg), equation (2.14)
$m_{fu}$ Total mass flow rate of the fuel (Kg/s)
$m_p$ Droplet mass (Kg)
$n_2$ Number of the droplet per second
$Nu$ Nusselt number
$P$ Pressure (N/m$^2$)
$Pr$ Prandtl number
$P(\cdot)$ Probability density of ( )
$Q_c$ Heat transferred by convection to the thermocouple, equation (2.10)
$Q_k$ Heat transferred by conduction from the thermocouple, equation (2.10)
$Q_r$ Heat transferred by radiation from the thermocouple, equation (2.10)
$Q_s$ Stored energy by the thermocouple, equation (2.10)
r Radial distance (m)
$r_i$ Inner radius of the swirler (m)
$r_o$ Outer radius of the swirler (m)
$R$ Rate of arrival of scattering particles in measuring volume, equation (2.31)
$Re$ Reynolds number
$R_{CO}$ Concentration of carbon monoxide, equation (3.2)
$R_{CO_2}$ Concentration of carbon dioxide, equation (3.2)
\( R_H \) Concentration of hydrogen, equation (3.2)

\( R_{H_2O} \) Concentration of water, equation (3.3)

\( R_{Fu} \) Rate of reaction, equation (4.17)

\( R_r \) Total radial radiation flux, equation (4.41)

\( R_x \) Total axial radiation flux, equation (4.41)

\( S \) Swirl number

\( S \) Generation and/or destruction source term, gas field

\( S_d \) Droplet source term

\( t \) Time

\( \bar{t} \) Time constant (ms)

\( t_d \) Ignition delay time (ms)

\( t_e \) Evaporation time (ms)

\( t_{sl} \) Shear layer mixing time (ms)

\( T \) Temperature (°K)

\( T_a \) Ambient temperature (°K)

\( T_b \) Thermocouple support wire temperature (°K)

\( T_s \) Saturation temperature of the fuel (°K)

\( T_{sh} \) Suction pyrometer shield temperature (°K)

\( T_{sp} \) Sampling probe tip temperature (°K)

\( T_w \) Wall-temperature (°K)

\( U \) Axial velocity (m/s)

\( u', v', w' \) Fluctuating velocity components (m/s)

\( U_g \) Gas velocity of the flame (m/s)

\( U_s \) Suction velocity of the gases through the suction pyrometer (m/s)

\( U_{sp} \) Sampling probe suction velocity (m/s)

\( V \) Radial velocity component (m/s)

\( V' \) Convective velocity (m/s)
\( \vec{V} \) Total velocity (m/s)
\( \dot{V}_m \) Volumetric rate of the gases through the suction pyrometer (m\(^3\)/s)
\( \dot{V}_{sp} \) Sample flow rate (m\(^3\)/s)
\( W \) Tangential velocity component (m/s)
x Axial distance (m)
\( Y \) Radial position (m)
\( Y_i \) Species concentration

Greek characters
\( \Gamma \) Exchange coefficient
\( \Delta t \) Time step
\( \Delta T_r \) Temperature radiation correction (°C)
\( \Delta T_c \) Temperature conduction correction (°C)
\( \Delta T_v \) Temperature velocity error (°C)
\( \delta_{ij} \) Kronecker-delta ( = 1 for i = j; = 0 for i ≠ j)
\( \varepsilon \) Dissipation rate of turbulent kinetic energy
\( \varepsilon_{sh} \) Emissivity of the suction pyrometer shield
\( \varepsilon_w \) Emissivity of the thermocouple wire
\( \varepsilon'_{w} \) Emissivity of the combustor wall
\( \lambda \) Thermal conductivity (J/Kgm°K)
\( \lambda' \) Wave length (m)
\( \mu \) Viscosity (Kg/ms)
\( \mu_l \) Laminar viscosity (Kg/ms)
\( \mu_t \) Turbulent viscosity (Kg/ms)
\( \nu \) Kinematic viscosity (m\(^2\)/s)
\( \nu_D \) Mean Doppler frequency (HZ)
\( \nu'_{D} \) rms of Doppler frequency
\( \rho \) Density
\(\sigma\)  
Stefan-Boltzmann constant \((W/m^2K^4)\)

\(\phi\)  
General dependent variable

\(\alpha\)  
Half-angle between incident laser beams (degrees)

\(\theta\)  
Vane angle of the swirler (degrees)

\(\beta\)  
Equivalence ratio

\textbf{Subscripts}

\(\text{eff}\)  
Effective (including the effects of turbulence)

\(\text{b}\)  
Beginning of time step

\(\text{e}\)  
End of time step

\(\text{f}\)  
Fluid

\(\text{fu}\)  
Fuel

\(\text{g}\)  
Gas

\(\text{h}\)  
Enthalpy

\(\text{i, j}\)  
Components in Cartesian tensor notation

\(\text{in}\)  
Inlet

\(\text{L}\)  
Size range

\(\text{ox}\)  
Oxidant

\(\text{p}\)  
Droplet

\(\text{pr}\)  
Products

\(\text{th}\)  
Thermocouple

\(\text{w}\)  
Wire

\(\text{\&}\)  
General dependent variable

\textbf{Superscripts}

\(\text{-}\)  
Time averaged

\(\text{\sim}\)  
Density-weighted averaged

\(\text{'}\)  
Fluctuating component
CHAPTER 1

INTRODUCTION

1.1 Problems considered

The combustion of the liquid sprays is of considerable practical importance since combustion devices ranging from gas turbines to industrial furnaces involve the burning of liquid fuel sprays in turbulent gas flows. The combustion process of the spray can be separated into heterogeneous, fluid mechanic and chemical effects. The fluid mechanic and chemical effects are known to be governed by the turbulent mixing between the fuel and oxidant and the chemical kinetics respectively. However, the heterogeneous effects are linked to the spray characteristics including the evaporation rate, droplet size and velocity and fuel properties. Quantification of the extent to which the spray characteristics contribute to the overall combustion process represents a major goal in the current research. The practical relevance is made clear by the example of the gas-turbine combustor where fuel-air loading, combustor pressure and the degree of preheat vary considerably from idle to full power conditions with consequent variations in the spray characteristics. In addition, the role of aerodynamics is important since it influences the spray characteristics, flame stability, mixing of fuel and combustor power. Current restrictions on pollutant emission levels and the need for more efficient combustors and furnaces make it important to quantify the influences of the previous parameters on the combustion process.

Although general trends can be deduced from existing knowledge, comprehensive and accurate data are lacking with the
result that detailed understanding of the spray-combustion performance is not available. Thus a major purpose of the present contribution is to improve available knowledge by new experiments. In addition, it is intended that this should provide sufficient information to allow evaluation of the calculation methods.

Because of the importance of the interaction between the fuel spray and the surrounding combustion air and the spray characteristics on the combustion performance, a large part of the thesis involves measurements of the flame properties and droplet characteristics for a range of confined and unconfined kerosene spray flames. The precision of the measurements has been assessed by detailed consideration of the techniques used. A computational method has been applied to predict the flame properties and is appraised by comparison of the results with the experimental data.

1.2 Previous work

This section reviews relevant literature to the present study. In the next subsection the previous experimental investigations are considered and is followed by brief consideration of the calculation of spray combustion.

1.2.1 Experimental investigation

Spray combustion involves complex chemically reacting multicomponent two-phase flows with phase change. Numerous studies of the fundamental processes, such as individual droplet evaporation and combustion of simplified arrangements of droplets were performed to gain inside of the mechanisms governing the combustion process and its application to establish design criteria for efficient and stable


Investigations of the influence of the droplet diameter on the flame behaviour have been reported by El-Banhawy and Whitelaw (1981), Founti, Hutchinson and Whitelaw (1980), Komiyama et al (1977), Khalil and Whitelaw (1977), Onuma et al (1975,1977) and Beer
(1962), however, Tuttle et al (1976,1973), Shisler et al (1975), Nizami et al (1979) and Nicholls et al (1980) have reported results concerning the emission characteristics. The available evidence, in general, suggests that droplets do contribute to the combustion process but the extent of the influence is not fully quantified. For example, El-Banhawy and Whitelaw (1981) measured droplet velocity, temperature and species concentrations (CO₂, CO and O₂) for a range of kerosene spray flames in a cylindrical combustion chamber equipped with a rotating cup atomizer, to quantify the influences of the droplet diameter and combustion air swirl on the flame characteristics. The effect of the droplet diameter, with two levels of air swirl, showed that an increase in mean-droplet diameter is associated with a reduction in the combustion intensity with more significant influence accompanied with the low swirl flame. The measurements of Beier (1962) in heavy fuel oil spray flames indicated that the variation in the droplet diameter affect flame propagation and burnout of carbon. The results of Khalil and Whitelaw (1977) and Founti et al (1980) showed a reduction in the axial energy release rate with increase in the droplet diameter. Oruma et al (1975,1977) made a direct comparison between the structures of spray combustion flames, with mean-droplet diameters of 10 - 40 μm, and a turbulent gas diffusion flame in a vertical cylinder furnace. The spray flame was generated by an air atomizing nozzle with kerosene fuel and a secondary air supply. Measurements were made for temperature, velocity and gas concentrations by using a thermocouple, pitot tube and sampling tube with gas phase chromatography respectively. Further measurements were made in a turbulent gas diffusion flame, using the same apparatus and replacing the liquid kerosene fuel by propane gas. They concluded that most of the droplets do not burn individually but
that the vapour cloud from the evaporated droplets burn like a gas diffusion flame in the turbulent state. Komiyama et al (1977) studied the importance of the droplet evaporation in the overall fuel-air mixing process using air-blast and pressure jet atomizers for a range of confined spray flames. They indicated that where the characteristic time of evaporation is much less than the jet mixing time, the details of the evaporation process are not important and the jet length scale and kinetic energy govern the mixing process as for gaseous flames. On the other hand, when the characteristic times are comparable (pressure jet atomizer), the initial fuel-air mixing rate is determined by the evaporation characteristics of the fuel droplets and the kinetic energy of the jet. These conclusions were deduced from cross-sectional averaged oxygen concentration and no details of other flame properties were reported. Tuttle et al (1976,1973), Shisler et al (1975) measured temperature and pollutant species (CO, UHC and NO\textsubscript{x}) in simulated and actual gas turbine combustors under a range of fuel-air loading conditions. Related changes in droplet size, evaporation rate and penetration were found to have large influence on the flame structure and pollution characteristics. In general, heterogeneous effects, due to the fuel spray, were found to increase in importance in going from low to high fuel and air flow rates. It is, however, likely that the experimental results of these studies are influenced by the simultaneous variation in both the fuel and air operating conditions which allowed indirect quantification of the influence of the fuel spray. Nizami et al (1979) found that while NO\textsubscript{x} emissions display an overall increase with drop size, a local minimum in NO\textsubscript{x} emission occurred for a 50 \(\mu\) m droplet size. In contrast, Nicholls et al (1980) suggested that varying the droplet size has little effect on NO\textsubscript{x} emission levels.
El-Banhawy and Whitelaw (1981), Owen et al (1979) and Beer (1965) reported results concerning the influence of the combustion air swirl and its interaction with the fuel spray on the flame characteristics. The results indicated that a general trend of changes in the structure of confined spray flames and combustion intensity with the variation of the inlet air swirl. El-Banhawy and Whitelaw (1981) emphasized the contribution of the flow reversal regions and the recirculation of small droplets and vapour to the stabilization mechanism of the flames, however, Owen et al (1979) and Beer (1965) suggested an increase in the flame length with swirl.

The influence of inlet-air temperature has been reported by Katsuki et al (1976), in a model gas turbine combustor, and Yule et al (1982), for non-combusting free jet spray. The results, generally, emphasized the importance of air temperature on the evaporation rate of the spray and the reaction rate, but the information is incomplete and covers very limited operating conditions and geometries.

1.2.2 Computational investigations

Modelling of evaporating and combusting sprays is a difficult problem due to the diverse phenomena that must be considered, including: the hydrodynamic characteristic of injection and spray formation; the transport characteristics of individual droplets; the turbulent two-phase flow of a spray, and chemical phenomena in a turbulent environment leading to the formation of product species and pollutants. Thus, in some cases, models have ignored the details of spray evaporation and treated the system as a gaseous diffusion flame; in many circumstances, this simplification
is not adequate and turbulent two-phase flow must be considered.

Only recently, attempts have been made to consider the details of the fuel spray based on the behaviour of individual or group droplets vaporization in order to predict the local flow properties in spray flames. Different approaches have been reported and include those of Williams (1965), Crowe (1974), Faeth (1977), Crowe et al (1977), Labowsky (1980), El-Banhawy and Whitelaw (1980a, 1980b), Law (1982) and Correa et al (1982). There are two main alternative approaches and these are described briefly in the following paragraph and the reader is referred to the review by Faeth (1977) for further details and approaches.

The first approach is named "distribution function approach", was developed by Williams (1965) and considers a generalized spray distribution function originally defined in eight-dimensional space of droplet diameter, location, velocity and time. Conservation principles yield a partial integro-differential equation, together with the gas conservation equations, provides the required model of the spray. The main difficulty associated with this approach is the solution of the final set of equations. It has been applied with drastic simplifying assumptions by Westbrook (1977), Gupta et al (1978) and Jones and McGuirk (1981). Evaluation of this approach against experimental data, apart from Jones and McGuirk (1981), has not been reported.

The second approach is called the "discrete droplet model" and has been described in detail by Crowe (1974), Crowe et al (1977) and El-Banhawy and Whitelaw (1980b). In this method the spray is represented by individual droplets rather than by a continuous distribution function. Because of the large number of actual droplets
contained in the spray, the representation is confined to a statistical sample. Each of these sample droplets characterises a parcel of like numbers all having the same initial size, velocity and temperature. The motion, heating and evaporation of each sample as it traverses the gas are computed by solving numerically the Lagrangian ordinary-differential equations which govern the mass, momentum and energy conservation. The effects of the droplets on the gas phase are introduced by solving the Eulerian transport equations for the gas with extra source terms to describe the mass, heat and momentum exchange deduced from the analysis of the droplet parcel trajectories. A droplet tracking procedure allows the calculation of these source terms in the gas-phase equations. This approach has been employed in most of the current models of evaporating and combusting sprays and has been reported by El-Banhawy and Whitelaw (1980a, 1980b), Jones and McGuirk (1981), Gosman et al (1981), Dukowicz (1980), Boysan et al (1982) and Abbas (1983). Validation of this approach against detailed experimental data is incomplete and suggests that further developments, modifications and evaluation by experimental data are required.

1.3 The present contribution

The review presented in the previous section has shown that the influence of the combustion air preheat on the spray characteristics and the flame performance is rarely reported and not well quantified. It is evident that, although different aspects of spray combustion have been considered, the understanding of the mechanisms governing the evaporation and combustion are still far from complete and there has been no detailed consideration of the uncertainties associated with the measurement techniques in spray
flames. The review also indicated that it is necessary to apply the prediction procedures to a range of combusting sprays and to evaluate their performance by comparison with detailed experimental results.

The main contribution of this thesis can be separated into three parts. The first consists of quantification of the performance and accuracy of the temperature and gas analysis measurement techniques in kerosene spray flames. The main purpose of this investigation is to choose the suitable technique for the measurements and to estimate the related errors and type of averaging measured.

The second part consists of detailed measurements of droplet number density and velocity with corresponding rms of velocity fluctuation, gas temperature, species concentrations of CO\textsubscript{2}, CO, O\textsubscript{2} and UHC and wall-temperature in unconfined and confined kerosene spray flames. The unconfined flame was generated by a twin-fluid atomizer burning kerosene in still air with vertical axis. The experiment quantifies the uncertainties associated with the measurements described before and provides new information of unconfined spray flames. The confined spray flame experiment was conducted in a water-cooled cylindrical combustor equipped with a rotating cup atomizer capable of producing a near-monosized spray. The influences of the droplet diameter and combustion air swirl and inlet-temperature on the spray characteristics and the flame performance and structure were investigated and provided new and reliable data to assist the evaluation and development of the spray combustion models.

The third part is an application of a calculation method using the "discrete droplet model" to predict the confined spray
flame properties. The method is appraised by comparison of the predicted results with the experimental data.

1.4 Thesis outline

The remainder of the thesis is presented in four chapters. Chapter 2 and 3 are concerned with the experimental work of this study. The techniques for the measurement of temperature, species concentration and velocity used are described in chapter 2. Each method is followed by detailed investigation and discussion of the error sources and accuracy. The chapter ends with the main concluding remarks. Chapter 3 is devoted to the presentation and discussion of the experimental results obtained in the confined and unconfined kerosene spray flames. The results of each geometry are reported in a separate section and followed by a brief summary.

Chapter 4 is concerned with the prediction of the confined flame experiment of chapter 3. The mathematical formulations, including the turbulence, combustion, spray and radiation models, are summarized together with the relevant boundary conditions and the numerical scheme of the solution. The results are then compared with the measurements and discussed and the chapter ends with brief concluding remarks.

Finally, chapter 5 summarises the main conclusions and provides suggestions for further work.
CHAPTER 2

MEASUREMENT TECHNIQUES

2.1 Introduction

In the study of spray combustion, knowledge of temperature, gas analysis and aerodynamic of the flow are important. Therefore, it is necessary to consider measurement methods and this chapter describes the techniques used for the measurements of temperature, species concentration and velocity in this work and consider, their likely precision.

Temperatures were measured by a suction pyrometer and bare wire thermocouples with wire diameters of 40, 80 and 300 μm. Results were obtained in the unconfined kerosene spray flames described in chapter 3 and comparison of the results obtained with the different instruments assists the assessment of precision. The 80 μm wire diameter bare-wire thermocouple was preferred for its combination of longer life and smaller radiation and conduction losses.

Gas samples to allow the measurement of the concentrations of carbon dioxide, carbon monoxide, oxygen and unburned hydrocarbon were withdrawn with a stainless steel water-cooled sampling probe. The effects of quenching, sampling velocity and probe orientation were investigated in the spray flames reported in chapter 3.

A laser Doppler anemometer was used to measure the mean velocity, rms of velocity fluctuations and droplet number density. The receiving optics were arranged in the forward scattered mode for the unconfined flames and in backward scatter mode for the axisymmetric combustor experiments of chapter 3. The accuracy of measurements is discussed.
The chapter begins with a description of the different bare-wire thermocouples and the suction pyrometer, a discussion of the errors associated with each and a comparison between the temperature results obtained by the different methods. The sampling system and the equipment used for the species concentration measurements are described in section 2.3 with an assessment of the accuracy of the species analysed. The arrangements of the laser Doppler anemometers and the errors associated with the measurements are described in section 2.4. The final section presents a summary of the main conclusions of the chapter.

2.2 Temperature measurements

Mean gas temperature is most commonly measured with thermocouple probes. Many factors affect the measured value including, for example, temperature level, gas velocity, temperature of the surroundings, thermocouple construction and the type of flame. As a result, the temperature indicated by the thermocouple is usually lower than the true flame temperature due mainly to the heat loss to the surroundings by radiation and conduction along the thermocouple leads.

Bradley and Matthews (1968), Glawe et al (1956), Sato et al (1975) and Scadron and Warshawsky (1952) have obtained numerical solutions for the equations describing the heat transfer modes between the thermocouple bead and the surroundings. These calculation methods are subject to uncertainties relating to the calculation of the convective heat transfer coefficient which imply errors in the radiated and conducted heat transfer losses as indicated, for example, by Moffat (1963), Chedaille et al (1972) and Hayhurst and


The mean gas temperature can also be measured by a suction pyrometer, see Founti, Hutchinson and Whitelaw (1980), El-Banhaway and Whitelaw (1981), Land et al (1956), Chedaille and Braud (1972), Glawe et al (1956) and Terbush (1962). In suction pyrometers, the thermocouple hot junction is shielded and the gases are sucked over it. Suction of the gases through the pyrometer increases the convective heat transfer to a level which renders any radiation and conduction loss from the thermocouple negligible, provided that the hot junction is properly isolated and located inside the shield. Care must be taken to ensure that the suction rate has achieved a value which allows asymptote temperatures to be achieved. The extent to which the true temperature is measured by a suction pyrometer depends on the geometric detail and operating conditions, see for example, Chedaille and Braud (1972) and Glawe et al (1956). It is known that the number of shields, their material and geometry, the suction velocity, the bead position inside the shield and the flow conditions can all influence the results.

The above thermocouple probes have been used here and allow a comparison of the results and better assessment of precision.

The following two subsections, 2.2.1 and 2.2.2, present the details of bare-wire thermocouples and the suction pyrometer used and
provide an evaluation for the error associated with each type. Subsection 2.2.3 compares the temperature results obtained by the bare thermocouples and the suction pyrometer with a discussion of the type of average measured in terms of the mean and fluctuating values of the temperature and heat transfer coefficient.

2.2.1 Bare-wire thermocouples

The 300 µm bare-wire thermocouple was manufactured from platinum and platinum-13% rhodium wires of 300 µm diameter and had a hot junction of approximately 450 µm diameter which was formed by electric welding. As shown in figure 2.1, the wires were insulated and supported by a twin bore alumina sheath with an external diameter of 1.5 mm. The output signal from the thermocouple was measured with the equipment of figure 2.7. The large wire and hot junction diameters imply the need to correct for radiation and conduction heat losses. The following equations are commonly used to represent the two correction:

$$\Delta T_r = \sigma_w \left(\frac{T_{th}^4 - T_a^4}{h_c}\right) \quad (^\circ C) \quad (2.1)$$

$$\Delta T_c = \left(\frac{\delta_{th} - T_b}{\cosh \left(4 \frac{h_c}{d_j \cdot \lambda_w}\right)}\right) \quad (^\circ C) \quad (2.2)$$

Accurate values of the heat transfer coefficient, $h_c$, and the gas properties are clearly required to determine accurate radiation and conduction corrections. As an example, the corrections were calculated at a point on the centreline 500 mm downstream of the first flame, see section 3.2, using the heat transfer coefficient from the expression of Karmen, quoted by Hinze (1959);
\[ i.e. \ Nu = 0.42 \ Pr^{0.2} + 0.57 \ Pr^{0.32} \ Re^{0.5} \quad (2.3) \]

This expression holds for \(0.01 < Re < 10000\) and cross flow on a wire where:

\[
Re = \frac{\rho g U d_j}{\mu g} \quad \text{and} \quad Pr = \frac{\mu g C_p}{\lambda g} \quad (2.4)
\]

The gas velocity was assumed to be 15 m/s and the wire emissivity was taken equal to 0.3 as determined by Bradley et al (1961) for similar wires. The gas properties were evaluated with the help of the measured species concentrations using the method described by Brokaw (1961) in conjunction with data of species properties given by Svehla (1962).

The values of radiation and conduction errors estimated from the calculations were 240 °C and 10 °C respectively corresponding to a gas temperature of 1450 °C. This indicates that the radiation error can be large and represents 15.8% of the true mean temperature.

Thermocouples were also made from 40 and 80 \(\mu m\) diameter platinum and platinum-13% rhodium wires. The hot junctions were welded electrically by the technique described in detail by Moneib (1980) and the resulting bead diameters of approximately 45 and 90 \(\mu m\) respectively. The thermocouples were mounted on support wires of 500 \(\mu m\) diameter and of the same alloys. These were carried in twin-bore alumina insulator of external diameter 2.5 mm as shown in figure 2.2. The fine wires length between the hot junction and the supporting wires ensured a small temperature gradient along the wire and consequently minimized the conduction heat loss.
Since the radiation correction can be large, it was determined experimentally by compensating the radiated heat electrically, while the conduction error was estimated by calculation for both thermocouples as follows:

According to the analysis of Scadron and Warshawsky (1952), the time averaged conduction error, \( \Delta T_c \), is given by:

\[
\Delta T_c = \left( T_b - T_{th} \right) \frac{\psi_m}{(1 - \psi_m)} \quad (^{\circ}C)
\]

(2.5)

Where \( \psi_m = \text{sech} \left( \frac{v L}{2} \right) \) for the case where the junction is at the midpoint in a sensing wire of length, \( L \), and \( v^2 = 1/k \tau \).

For the platinum - platinum/13% rhodium thermocouple, the thermal diffusivity is given by \( k = 1.72 \times 10^{-5} \text{ m}^2/\text{s} \), the length is \( L = 15 \text{ mm} \) and a support wire temperature, \( T_b \), of \( 50 \text{ °C} \) is appropriate. The time constant, \( \tau \), was measured using the method described by Ballantyne et al (1976) and was found, approximately, to be 20 and 50 ms for the 40 and 80 \( \mu \text{m} \) thermocouples respectively 350 mm downstream of the first flame. The conduction error for the 40 \( \mu \text{m} \) thermocouple at a wire temperature of 1220 \( \text{°C} \) is:

\[
\Delta T_c = \frac{(50-1220) \times 5.59 \times 10^{-2}}{1 - 5.59 \times 10^{-2}} = -0.66 \times 10^{-2} \text{ °C}.
\]

And the conduction error for the 80 \( \mu \text{m} \) thermocouple at the same measuring point with a wire temperature of 1190 \( \text{°C} \) is:

\[
\Delta T_c = \frac{(50-1190) \times 6.14 \times 10^{-2}}{1 - 6.14 \times 10^{-2}} = -0.694 \text{ °C}.
\]

It is clear that the conduction error for these fine thermocouples is entirely negligible.
The magnitude of the radiation correction for the fine wire thermocouples was determined by compensating the radiated heat from the bead electrically. The technique is similar to that described by Holderness et al (1969), Odidi (1974) and Hayhurst et al (1977) and made use of the electrical circuit and the temperature equipment indicated in figure 2.3. The radiation loss calibration curve, which represents the power radiated by the probe at any temperature, was obtained using the high vacuum apparatus shown in plate 2.1. This was accomplished by electrically heating the thermocouple, which was located in the vacuum chamber at a pressure of approximately $10^{-2}$ torr, and simultaneously measuring the electrical power supplied to the hot junction and the related e.m.f. The resulting plots for the 80 µm and 40 µm thermocouples are shown in figures 2.4 and 2.5 respectively. It can be assumed that there is no loss of heat by conduction from the thermocouple under calibration conditions. In vacuum, convection losses are also eliminated and, hence, the power supplied to the probe is dissipated entirely by radiation.

While in the measuring position in the flames, the thermocouple was electrically heated and the heating power and the probe temperature were measured simultaneously. Plots of probe temperature versus electrical heating power were obtained at various positions along the axis of the two unconfined kerosene spray flames and the results are shown in figures 2.4 and 2.5. The temperature at the point of intersection of the curve obtained in the flame and the radiation loss calibration curve is the required probe temperature, corrected for the radiation loss. The experiment could not be performed with 40 µm thermocouples at temperatures above 1000 °C as the wire tended to break under the heating load. The results of figures 2.4 and 2.5 indicate that the maximum radiation correction
was 80 °C at 1400 °C and 35 °C at 1000 °C for the 80 μm and 40 μm thermocouples respectively.

The correction described above was based on the assumptions that the radiant energy received by the thermocouple bead from the gases of the flame and the combustion chamber walls is negligible and thus the average radiation temperature \( \left( \frac{T_g + T^*}{2} \right)^{\frac{4}{3}} \) is equal to \( \frac{T^4}{T_g} \). Holderness et al (1969) demonstrated that the first assumption will lead to over-compensation where the thermocouple bead receives a substantial amount of radiation from the flame gases and combustor hot walls. Therefore, it is expected that the measured correction will be reduced by a value that corresponds to the radiant energy to the thermocouple from incandescent carbon particles present in the flame. Unfortunately, it is difficult to quantify this reduction of the correction since there is not a simple method to obtain the radiation from the flame. However, the effect of the combustor-wall is insignificant as a result of the low wall-temperatures, see section 3.3. The effect of the second assumption depends on the magnitude of the temperature fluctuations, \( T^* \), and since these are unknown; the effect can not be exactly quantified. Lockwood et al (1975) assumed this error source, based on their temperature fluctuation measurements, and suggested that it is likely to be less than 1%.

2.2.2 Suction pyrometer

The suction pyrometer was similar to that described by Rounti, Hutchinson and Whitelaw (1980). The thermocouple was formed from 300 μm wire which was insulated by a twin-bore alumina sheath with an outside diameter of 1.5 mm; the sheath was covered by an
alumina shield of 3 and 6 mm inner and outer diameter respectively. The shield isolated the thermocouple junction from the radiation to the surrounding and was secured to a brass water cooled jacket. The configuration of the suction pyrometer is shown in figure 2.6.

The best location of the thermocouple junction inside the shield and the proper suction velocity to achieve the highest possible temperatures were determined experimentally. The gases were sucked through the pyrometer with a vacuum pump (Edwards ES 150 High Vacuum Pump) and the flow rate was measured by a rotameter. The output signal from the thermocouple was measured with the temperature circuit shown in figure 2.7, which included a D.C. amplifier, Disa type 52B30 integrator and Solartron DVM respectively.

The existence of soot in spray flames can cause reduction in the suction velocity and probe blockage. It was necessary, therefore, to remove deposited soot from the probe at each position of measurements and maintain the suction flow rate at a value between that required to maximize the convective heat transfer to the thermocouple bead and that which reduced the blockage by solid deposits or condensed vapours to satisfactory limits. The corresponding suction velocities were of the range 150 - 200 m/s.

The influence of the position of the hot junction inside the shield was examined and figures 2.8 and 2.9 present results obtained with distances of 2, 5 and 10 mm from the end of the shield in the two unconfined spray flames of chapter 3. The suction velocity was maintained at approximately 170 m/s and indicates that the 2 mm distance between the hot junction and the end of the shield leads to the highest recorded temperature for downstream distances greater than 250 mm. Measurements with the hot junction further than 2 mm
from the end of the shield indicate lower temperatures in this downstream region. Distances less than 1.5 mm also resulted in lower temperatures, in keeping with the expected effect of radiation heat transfer. The reduction in temperature with increase in distance from the end of the shield is due to reduced heat transfer between the gases and the hot junction and to the decrease in the shield temperature as a result of conduction heat losses through the pyrometer. In the upstream region and at small radii in the flame, where high concentrations of unburned hydrocarbons existed, there is a tendency for the measured temperature to increase with distance between the end of the shield and the hot junction. However, the magnitude of the increase is less than 20 °C and is probably due to several factors including droplet impingement on the thermocouple, increase in the shield temperature as a result of the surrounding gases temperature distribution as shown in figure 2.9 at X = 150 mm, and combustion of kerosene inside the shield. In this region of the flame, the tendency for probe blockage required that it be cleaned at frequent intervals as steady state values, corresponding to a uniform suction velocity were occasionally difficult to obtain.

The influence of the suction velocity on the temperature measured by the suction pyrometer was investigated and the results are shown in figure 2.10. The hot junction was located 2 mm from the end of the shield for this test which was carried out for the temperature range 700 - 1500 °C, where the heat losses were expected to be high. The suction velocity around the hot junction, $U_s'$, was calculated from the following expression and assuming constant pressure flow of gases through the pyrometer:

$$U_s' = \frac{(V_xT_a)}{A_xT_a} \quad \text{(m/s)}$$  \hspace{1cm} (2.6)
Where $V_m$ is the measured volume rate of the gases drawn at the ambient temperature, $T_a$; $T_m$ is the measured temperature of the flame and $A_s$ is the cross-sectional area of the shield. The figure indicates that an increase in the suction velocity of the gases around the thermocouple bead results in an increase in temperature. For example, at the centreline point 300 mm from the atomizer, an increase in the velocity from 50 to 170 m/s resulted in an increase in the mean temperature from 900 to 1130 °C and 700 to 930 °C in the first and second flame respectively. This is due to the increase in the heat transfer coefficient associated with the higher Reynolds number. It is also apparent that the temperature asymptotes to a constant value with increase in suction velocity. The suction velocity required to give the constant temperature should be greater than 200 m/s as shown in figure 2.10; a value in the range 150 - 200 m/s was used here to avoid probe blockage in too shorter time. At the highest temperatures, the use of this suction velocity implied an error of around 4.5% or 100 °C.

As the suction velocity of gases over the thermocouple bead, $U_s$, differs from the local gas velocities, $U_g$, the temperature indicated by the suction pyrometer, $T_m$, is different from the gas temperature, $T_g$, due to energy exchange within the boundary layer. Chedialle and Braud (1972) discussed this velocity error and introduced this equation:

$$\Delta T_v = T_g - T_m = (1 - a) \left( \frac{U_s - U_g}{2C_{p_g}} \right) \quad ^\circ C$$  \hspace{1cm} (2.7)

Taking the suction velocity, $U_s = 170$ m/s and the specific heat of the gases, $C_{p_g} = 1210 \text{ J/kg}^\circ C$ and assuming the gas velocity, $U_g = 15$ m/s
and the recovery factor, \( a = 0.09 \) for thermocouple parallel to the flow velocity, the resulting velocity error is 1.18 °C and clearly it is negligible.

The radiation heat loss from the thermocouple junction to the shield and to the surroundings represents another source of error in the measured temperature by the suction pyrometer and can be calculated from the expression:

\[
\Delta T_r = \left( \frac{\sigma e_w}{h_c} \right) \left\{ \left( T_{th}^4 - T_{sh}^4 \right) + \left( T_{th}^4 - T_a^4 \right) \right\} \quad (°C)
\]

(2.8)

The heat transfer coefficient was calculated from the following correlation for the heat transfer to a thermocouple bead placed parallel to the flow direction, see Moffat (1962):

\[
Nu = 0.094 \left( \frac{Re}{674} \right) \quad \text{and} \quad Re = \frac{\rho_g d_j U_s}{\mu_g}
\]

(2.9)

It was found this error represents around 45 °C at the high temperature.

The preceding discussion of the errors associated with the suction pyrometer reading suggests that, for example, at high flame temperature of 1500 °C, the uncertainty due to the lower suction velocity used is 100 °C. The error related to the radiation from the thermocouple junction represents 45 °C while the effect of the probe gas interference is negligible. Consequently, it can be concluded that the indicated suction pyrometer reading is associated with an error of around 145 °C at the high flame temperature.

2.2.3 Comparison of temperature measurements

This section compares the temperature measurements obtained
with the three different bare-wire thermocouples and the suction pyrometer described in subsections 2.2.1 and 2.2.2. The measurements were performed along the centre-line and along the radii at the three axial locations of \( X = 150, 300 \) and 500 mm from the fuel nozzle for the two unconfined flames described in chapter 3.

Figures 2.11 and 2.12 present the axial and radial distributions of temperature obtained with the three bare-wire thermocouples and the suction pyrometer with the 2 mm distance between the bead and the end of the shield. The 40 \( \mu \text{m} \) and 80 \( \mu \text{m} \) thermocouples results have been corrected for radiation effects as described in subsection 2.2.1. The results with the 300 \( \mu \text{m} \) thermocouple, with reference to those of the fine wires and the suction pyrometer, demonstrate clearly the magnitude of the radiation and conduction errors associated with a comparatively large (up to 15\%) as shown in figure 2.11 and 2.12. At temperatures around 1500 °C, the error can be large as 220 °C and even at low temperatures it can be 40 °C. The results of figures 2.11 and 2.12, also, indicate that the temperatures obtained from the two fine wire thermocouples are larger than those from the suction pyrometer with a maximum discrepancy of around 110 °C. The discrepancies increase to around 140 °C when the radiation correction is applied to the fine wire thermocouple readings. The main reasons for the differences between the results of the fine wire thermocouples and the suction pyrometer stem from the suction velocity used with the pyrometer and the losses by radiation from the thermocouple bead to the covering shield and the surrounding. The analysis of section 2.2.2 suggests that a correction of the order 145 °C should be applied to the suction pyrometer readings.

The temperature probes are subject to an energy balance
which may be written as:

\[ Q_c + Q_r + Q_k + Q_s = 0 \]  

(2.10)

where the subscripts refer respectively to the convective, radiative, conduction and stored energy. Since the conduction heat loss is negligible for these small thermocouples (<1 °C) as estimated in subsection 2.2.1, equation (2.10) can be written as:

\[ Q_c + Q_r + Q_s = 0 \]  

(2.11)

where: 

\[ Q_c = h_c A (T_g - T_w) \]  

(2.12)

\[ Q_r = -\varepsilon w A (T_w^4 - T_a^4) \]  

(2.13)

and 

\[ Q_s = -m_w C_w (dT_w/dt) \]  

(2.14)

Thus: 

\[ h_c A (T_g - T_w) - \varepsilon w A (T_w^4 - T_a^4) - m_w C_w (dT_w/dt) = 0 \]  

(2.15)

The instantaneous values of the gas temperature, \( T_g \), wire temperature, \( T_w \), and the heat transfer coefficient, \( h_c \), can be decomposed into a mean value and a fluctuating component, i.e.

\[ T_g = \bar{T}_g + T'_g \]  

(2.16)

\[ T_w = \bar{T}_w + T'_w \]  

(2.17)

\[ h_c = \bar{h}_c + h'_c \]  

(2.18)
where $\bar{T_g}$, $\bar{T_w}$ and $\bar{h}_c$ are the time averaging quantities and $T_g^\prime$, $T_w^\prime$ and $h_c^\prime$ are the fluctuating quantities.

Substitution of equations (2.16), (2.17) and (2.18) into equation (2.15) and taking time averages, equation (2.15) becomes:

$$\bar{T} = \bar{T_g} - \frac{\sigma_e}{\bar{h}_c} \cdot \left( \frac{T_g^4 - T_a^4}{T_w^4} \right) - \frac{\sigma_c}{\bar{h}_c} \cdot \left( \frac{T_w^4}{T^4} \right) \cdot \left\{ 6 \left( \frac{T_w^2}{T^2} \right)^2 + 4 \left( \frac{T_w^3}{T^3} \right) + \left( \frac{T_w^4}{T^4} \right) \right\}$$

radiation term temperature fluctuation term

$$+ \left\{ \frac{\bar{h}_c^\prime T_g^\prime}{\bar{h}_c} - \frac{\bar{h}_c^\prime T_w^\prime}{\bar{h}_c} \right\}$$

correlation term

(2.19)

It is clear that the correlation and the temperature fluctuation terms in equation (2.19) are difficult to quantify exactly, but their order of magnitude can be estimated by inspection of the present and related results. For an infinitely small hot junction, $T_g^\prime$ and $T_w^\prime$ are identical and the difference between the wire and gas temperature will be a function of the radiation heat loss and the temperature fluctuation terms. The former is insignificant in this case and the latter depends on the local fluctuation intensity, and it can be shown that it is smaller than the radiation error. Therefore, the thermocouple reading for this case can be regarded as the true time averaged gas temperature.

The terms on the right hand side of equation (2.19) are all dependant on the wire diameter and to conclude that the wire mean temperature, $\bar{T}_w$, represents the time averaged gas temperature, $\bar{T}_g$, the value of the terms involving the correlations and temperature fluctuations must be of similar magnitude to the radiation heat loss and with positive sign. This was concluded by Hayhurst and Kittelson (1977) who used 51, 25 and 13 μm thermocouples, Schenung and Mitchell (1979) who compared laser Raman spectroscopy and a 25 μm thermocouple
and Moneib (1980) who employed thermocouples of 40 µm and less: they all found identical temperatures to within ±30 °C. As the wire diameter increases, it can be seen from equation (2.19) that the temperature fluctuation term will be reduced and the correlation term will increase. This will, however, be balanced by an increase in the radiation heat loss and it can be expected that the uncorrected temperatures of figure 2.11 obtained with the 40 µm thermocouple are close to the time averaged gas temperature, \( \bar{T}_g \). Figure 2.11, also, indicates that an increase in the wire diameter from 40 µm to 80 µm is associated with a reduction in the uncorrected mean temperature by around 30 °C at the highest temperatures. Equation (2.19) suggests that the increase in radiation losses with wire diameter occurs at a higher rate than the increase in the magnitude of the correlation and fluctuating temperature terms. With further increase in the wire diameter, for example up to 300 µm, the results of figure 2.11 indicate a reduction in temperature of around 220 °C from the uncorrected mean temperature of 40 µm thermocouple. The previous estimate of radiation and conduction errors, see subsection 2.2.1, suggests a correction of the order of 250 °C. The term \( (h_c \bar{T}_g/h_c) \) which remains in equation (2.19) when applied to the 300 µm thermocouple will represent the difference (30 °C) between the corrected value of 300 µm thermocouple and uncorrected reading of 40 µm. The temperatures estimated with the suction pyrometer are 30 - 110 °C lower than those of the 40 µm thermocouple as shown in figure 2.11; the previous discussion of section 2.2.2 suggests a correction of the order 145 °C for the suction pyrometer readings. This difference between the corrected suction pyrometer reading and the 40 µm thermocouple probably represents the correlation term \( h_c \bar{T}_g/h_c \). If it is assumed that the value of the correlation and temperature
TABLE (2.1) SUMMARY OF RESULTS RELEVANT TO TEMPERATURE MEASUREMENTS.

<table>
<thead>
<tr>
<th></th>
<th>Bare-wire thermocouples</th>
<th>Suction pyrometer</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( d = 300 \mu m )</td>
<td>( d = 40, 80 \mu m )</td>
</tr>
<tr>
<td>* Operation.</td>
<td>* No difficulty</td>
<td>* Difficulty in handling and wire breaking</td>
</tr>
<tr>
<td></td>
<td>* Can be used for a very long period of time.</td>
<td>* Short life time</td>
</tr>
<tr>
<td>* Interference with the flame.</td>
<td>* Small.</td>
<td>* Negligible</td>
</tr>
<tr>
<td>* Radiation error.</td>
<td>* Significant (( \approx 200 - 250 ^\circ C ))</td>
<td>* Small (( \approx 30 - 80 ^\circ C ))</td>
</tr>
<tr>
<td>* Conduction error.</td>
<td>* Small (( \approx 10 ^\circ C ))</td>
<td>* Less than 1 ( ^\circ C ).</td>
</tr>
<tr>
<td>* Quantity measured (with radiation and conduction errors accounted for).</td>
<td>* Differ from the time mean average by ( \Delta T_g ).</td>
<td>* Can be used for suitable periods.</td>
</tr>
<tr>
<td></td>
<td>* 40 ( \mu m ): ( \approx ) time averaged gas temperature ( +40 ^\circ C ), and 80 ( \mu m ): differs from the 40 ( \mu m ) wire reading by ( \Delta T_{gf} ).</td>
<td>* Can be significant due to suction of gases</td>
</tr>
<tr>
<td></td>
<td></td>
<td>* Negligible</td>
</tr>
<tr>
<td></td>
<td></td>
<td>* Differs from time mean average by ( \Delta T_g ).</td>
</tr>
</tbody>
</table>

\[ \Delta T_f = \left( \frac{h_c T'_c}{h_c} \right) - \left( \frac{h''_c T''_c}{h_c} \right) - \left( \sigma \frac{T''_w}{T''_c} \right) \left( \frac{6(T''_w^2/T''_c^2) + 4(T''_w^3/T''_c^3) + (T''_w^4/T''_c^4)}{2} \right) \]

\[ \Delta T_g = \left( \frac{h_c T'_c}{h_c} \right) \]
fluctuation terms are negative, similar arguments suggest that equation (2.19) will represent a value for $\tilde{T}_g$ close to that of 40 µm thermocouple after correction for radiation.

Although there is insufficient direct evidence of the magnitude of the correlation terms in equation (2.19), it can be assumed that the use of fine-wire thermocouples of diameter 40 µm or less will allow measurements of the time-averaged gas temperature with an expected uncertainty of not more than $\pm 40^\circ C$. In case of larger thermocouples and suction pyrometer, provided that the fluctuation in bead temperature is zero and adding a term to represent the conduction error, equation (2.19) will be:

$$\tilde{T}_w = \tilde{T}_g + \left( \frac{h'_c T_g}{\tilde{h}_c} \right) - \Delta T_r - \Delta T_c$$  \hspace{1cm} (2.20)

It is obvious from equation (2.20) that the measured average temperature, $\tilde{T}_w$, corrected for radiation and conduction errors differs from the true time-averaged gas temperature, $\tilde{T}_g$, by the term $\left( \frac{h'_c T_g}{\tilde{h}_c} \right)$. Therefore, the corrected temperatures of the big thermocouples and suction pyrometer will be averaged by $\left( \frac{h'_c T_g}{\tilde{h}_c} \right)$ instead of $\tilde{T}_g$ as indicated by Bilger (1977).

Table 2.1 summarises the conclusions which can be drawn in relation to temperature measurements in kerosene spray flames.

2.3 Concentration measurements

Measurements of the gas concentrations are most valuable of the various sources of information for the study of the combustion field. Gas samples are readily extracted from flames by sampling probes and analysed, but uncertainties exist arising from
difficulties in ensuring conservation of the initial characteristics of the sample, interference of the sampling probe with the flow pattern upstream of the sampling nozzle and from the need to provide rapid analysis of very complex gaseous mixtures. These difficulties have been considered by many research workers, for example, Westenberg et al (1957), Kent et al (1973), Beck et al (1975), Yanagi (1972,1977), Kramlich et al (1978), Clark and Mellor (1980) and Colket et al (1982), in their efforts to minimize the resulting error in the concentration measurements. However, there is no guarantee that these problems can be overcome, the main sources of errors related to the measured species concentrations are analysed in this section. The following subsection gives a brief description of the sampling system together with the analysis instrumentation used in this work. A discussion of the factors which affect the accuracy of the measured species concentration is presented in subsection 2.3.2.

2.3.1 Sampling and analysis instrumentations

Measurements of the concentrations of carbon monoxide, carbon dioxide, oxygen and unburned hydrocarbon were obtained in the unconfined and confined kerosene spray flames with the stainless-steel water-cooled probe shown in figure 2.13, which is similar to that described by other investigators, including for example, Toral and Whitelaw (1982), El-Banhawy and Whitelaw (1981), Noyce et al (1981), Kmiyama et al (1976) and Onuma et al (1976). The sample tube had an inner diameter of 2.05 mm and was surrounded by a cooling water arrangement comprising two-concentric tubes of 4.25 and 6.06 mm diameter respectively. The slender-nosed probe tip had a diameter of 4 mm at the measuring point. The sample was sucked through the probe by a 100% oil-free diaphragm compressor (Dawson McDonald and Dawson
The sample was passed through a water separator, filter and magnesium perchlorate to condense and remove the hydrocarbon and water vapours from the sampling line of carbon monoxide, carbon dioxide and oxygen. The sampling line connected to the unburned hydrocarbon analyser was heated electrically to 150 °C to prevent the condensation of the hydrocarbons in the sampling line. Therefore, the concentrations of the carbon monoxide, carbon dioxide and oxygen species are based on the dry analysis and the unburned hydrocarbon is based on wet analysis. The sample flow rate was measured by a rotameter. The probe was cleaned frequently, by blow back with high pressure air to maintain a constant suction flow rate. The carbon monoxide and carbon dioxide were measured using infra-red gas analysers (Analytic Developement Co. types 336 and 483 for CO and CO₂ respectively). The oxygen was analysed by a paramagnetic analyser (Tylor inst. Analytics Ltd., model Servomex OA. 540). The unburned hydrocarbons was measured by a high temperature flame ionization detector (Analysis Automation Ltd., model 523 II). Figure 2.14 shows a schematic diagram of the sampling system and gas analysis equipment.

2.3.2 Accuracy of concentration measurements

The factors influencing the accuracy of measurement are normally grouped into two categories: the first includes those resulting from probe gas interference and quenching of chemical reaction within the probe and the second is associated with sample handling and analysis.

Due to insufficient quenching, it is likely that the active species oxidize the carbon monoxide inside the sampling tube according to the reactions indicated by Malte and Kramlich (1980) and
Samuelsen et al (1979):

\[ \begin{align*}
\text{CO} + \text{OH} & \rightarrow \text{CO}_2 + \text{H} \quad (2.21a) \\
\text{CO} + \text{NO}_2 & \rightarrow \text{CO}_2 + \text{NO} \quad (2.21b)
\end{align*} \]

This oxidation of \( \text{CO} \) in probes is important, since it can lead to under-estimation of actual \( \text{CO} \) and uncertainties in the carbon dioxide. In the present work, quenching of chemical reaction at the probe tip was achieved by cooling the internal tube with a high pressure jet of water, as shown in figure 2.13, which gave substantial cooling on the outside of this tube within 5 mm of the tip. Sample results obtained within the unconfined flames confirmed that the quenching of the sample was sufficiently high. The results of figure 2.15 show that the concentration of both \( \text{CO} \) and \( \text{CO}_2 \) are independent of the sample cooling water flow rate in the range from 500 to 900 ml/min. The results also indicate that flow rates of less than 500 ml/min give rise to higher \( \text{CO}_2 \) and lower \( \text{CO} \) concentrations which means that the reaction expressed by equation (2.21) takes place. A value of 900 ml/min was used for the results of the following chapter. The quenching rate was estimated from the expression given by Ramshaw (1968):

\[
\frac{dT_g}{dt} = 4.3 h_c (T_g - T_{sp})/C_p \rho D_{sp} \quad (^{\circ}\text{K}/\text{s}) \tag{2.22}
\]

Where \( h_c \) is the heat transfer coefficient inside the suction tube, \( T_g \) and \( T_{sp} \) are the gas and probe tip wall temperature respectively. A gas temperature of 1800 \( ^{\circ}\text{K} \), probe wall temperature of 800 \( ^{\circ}\text{K} \) and a heat transfer coefficient of 200 Kcal/hr.m.\( ^{\circ}\text{K} \) calculated from the appropriate correlation for turbulent flow inside a tube.
\[ \text{Nu} = 0.02 \times \text{Re}^{0.8}, \quad (2.23) \]

gives rise to a quenching rate \( \text{dTg/dt} \) of \( 2.27 \times 10^6 \text{ K/s} \). These estimated values are in close agreement with those quoted by Ramshaw (1968), Bilger (1977), El-Banhawy and Whitelaw (1981) and Colket et al (1982).

Probe measurements in combusting flows are usually associated with uncertainties in the measured quantities due to the expected thermal and aerodynamic interference between the probe and surrounding gas. The degree of interference also changes according to the suction rate and direction of the probe with respect to the gas flow. Fristrom et al (1965), Yanagi (1972,1977) and Tine (1961) considered the problem of probe effects on concentration profiles in laminar flows, where they are likely to be greatest. The distortion of profiles decreased with the probe size and with the use of isokinetic sampling. In turbulent combustion flows, however, probe-gas interference effects are more difficult to quantify due to the fluctuating nature of the measured quantities and large variations in the flow conditions from point to point within a turbulent flame. Attempts to use isokinetic sampling in turbulent flames have been described by Kent and Bilger (1973), Vranos et al (1969), Bowman (1977), Oven et al (1978) and Mizutani et al (1976), but errors associated with departures from the isokinetic conditions were not quantified in detail. As indicated by Tine (1961) and Goulard et al (1976), it is difficult to guarantee isokinetic-sampling in a turbulent flow and, due to the complexity of the present flow conditions and the uncertainties associated with achieving isokinetic sampling, no attempt has been made to use this principle. Systematic
checks on the influence of changes in probe suction velocity and orientation, however, were made to provide an estimate of the likely errors. Visual observation indicated that, when the probe was placed in upstream and very near to the atomizer of the unconfined flame or the axisymmetric combustor, the probe adversely affected combustion stability. Concentration data for this region are not, as a consequence, presented. When the probe was moved further downstream, obvious probe distortion effects disappeared and the effect of flame cooling by the probe was negligible as indicated by the insignificant amount of heat transferred to the probe (the maximum rise in the probe cooling water temperature was less than 5°C).

Figure 2.16 shows the results of tests performed to study the influence of suction velocity on concentration measurements. The results correspond to points within the unconfined spray flames which were selected to coincide with regions of different concentration gradients. In general the results show that the concentrations of carbon monoxide and carbon dioxide increase by around 0.7% and 1% respectively with suction velocities from around 25.0 to 190.0 m/s. This increase is unlikely to result from probe sampling from a finite region surrounding the point of measurements as can be interpreted from the results obtained at points which were surrounded by comparatively lower concentrations. It is probable that it results from either segregation effects of the type described, for example, by Bilger (1977) or enhancement of chemical reaction at the points of measurements as a result of the higher suction velocities. Bilger (1977) indicated that suction of the gases at velocity different from the local gas velocities introduces the possibility that the probe segregates between gas pockets of low and high density especially in
regions where the size of these pockets is smaller than the probe size and where large density fluctuations exist. Higher suction velocities may thus be associated with a bias towards lower density. As gas pockets of low density (high temperature) are expected to be associated with high concentrations of combustion products, it is likely that the observed increase in both carbon monoxide and carbon dioxide can be related to this effect. The magnitude of the influence of suction velocity at locations off the centreline and near to the flame boundary are slightly higher than those obtained at other points and this can be related to the higher oxygen concentrations. The suction velocities, $U_{sp}$, indicated in figure 2.16 were calculated from the following expression and assuming a constant pressure along the sampling line, see Tine (1961):

$$U_{sp} = (V_{sp}/A_{sp}) \cdot (T_g/T_{a}) \cdot (P_a/P_g) \quad (m/s) \quad (2.24)$$

The present results were obtained at a sample flow rate ($V_{sp}$) of around 2000 ml/min with consequent suction velocities in the order of 70 m/s at the higher temperatures ($\approx 1600 \, ^\circ C$) and 20 m/s at temperatures of around 100°C. The gas velocities within the flames are in the velocity range 5 - 27 m/s as shown in chapter 3 and the results of figure 2.16, therefore, suggest an uncertainty in the measured concentrations of approximately $\pm 0.5$ percent volume fraction.

The results of figure 2.16 were obtained with the probe at a 90 degree angle to the axis of the flame, which implies that the samples were not extracted in the direction of the mean velocities and consequent bias could be present, as interpreted by Goulard et al (1976). Tests were conducted to attempt to quantify the uncertainties
and figure 2.17 shows the results of CO and CO₂ concentrations obtained with different angles of the probe axis to the flame axis and at different points within the flame. It is clear that there is a tendency for the measured CO₂ concentrations to increase as the probe angle decreased by an amount which is in the range 0.2 - 0.4%. On the other hand, the concentrations of CO show a slight reduction of around 0.25 - 0.45% volume fraction in the measured values with decrease of the probe angle. The trend of these variations in CO and CO₂ is probably influenced by the changes in the relative suction velocity due to the probe rotation and by the possibility of changes in the combustion rate at the point of measurements due to probe-gas interaction. Gas velocities in the direction of the flame axis are expected to be higher than the radial velocity and, consequently, the relative suction velocity will be smaller for small probe orientation angles, according to the results of figure 2.16 and those of figure 2.18, which show the effect of suction velocity for probe angles of 90, 45 and 30 degrees. The results of figure 2.17 indicate a reduction in the concentration of CO and not CO₂; it is possible, therefore, that the probe angle to the main direction of the flow influences the chemical reaction close to the probe tip, for example, by reducing the reaction at large angles or decreasing it at small angles.

Continuous analysis instrumentation used in these measurements have a response time less than 30 sec and, accordingly, the mean measured species concentrations were taken for a sampling duration of 3 - 5 min at each point. The range of the infra-red analysers for CO and CO₂ species measurements was adjusted automatically or by using standard gas-cylinders filled with a mixture of 12% CO₂, 4% CO and 84% N₂. The accuracy of the analysers
was specified to be better than ±1% of full scale reading (15% for 
CO₂ and 25% for CO). The accuracy of the paramagnetic oxygen analyser 
was also specified as ±1% of full scale deflection or ±0.02 oxygen, 
whichever is greater and oxygen gas was used to calibrate its range. 
The range of the high temperature flame ionization detector 
hydrocarbon analyser was adjusted by using standard gas cylinders 
contains 545 ppm propane in nitrogen and its accuracy was specified 
as ±1.5%. Standard nitrogen cylinder was used to adjust the zero for 
all analysers. The tolerance specified by the supplier of the gas 
cylinders used in the calibration of the analysers was ±5% and 
implies corresponding uncertainty in the measured values.

The measured values of species concentrations obtained in 
the present work are subject to uncertainty due to the influence of 
the fluctuations in the various properties on the type of averaging 
produced by the probe. Bilger (1977) indicated that if the sampling 
is unbiased in terms of collecting a representative sample of the gas 
volumes sweeping past the probe and using isokinetic sample 
collection, the resulting composition analysed at room temperature 
will not represent the true time mean composition, but should be the 
Favre mean: \( \bar{Y}_i = \frac{\bar{\rho} Y_i}{\bar{\rho}} \). Therefore, it can be concluded that the 
measured species concentrations are close to the density-average with 
an uncertainty due to the reasons discussed earlier; these errors are 
summarized below.

The preceding discussion of the factors influencing the 
accuracy of the concentration measurements suggests that the 
uncertainties associated with quenching of the chemical reaction are 
insignificant. Those associated with the probe gas interference as 
quantified by the effect of the probe suction velocity and 
orientation angle were found to be in the range ±3 - 7%. The
instrumentation errors are small and are anticipated to be within $\pm 1.5\%$. Consequently, it can be concluded that the measured species concentrations were obtained to an accuracy within $\pm 9\%$ of their Favre average.

2.4 Velocity measurements

This section deals with the laser Doppler anemometer and provides a brief description of the arrangement used in the measurements of this work. In general, the anemometer consists of two main parts: the optical arrangement and the signal processing system. The optical arrangement made use of forward scattered light in the unconfined flames and backward scattered light in the axisymmetric combustor. The two optical arrangements together with signal processing system are described in the following two subsections. The section ends with an analysis of the errors involved in the measurements.

2.4.1 Optical arrangement with forward-scattered light

The anemometer arrangement is indicated in figure 2.19 and made use of an argon laser (Spectra Physics Model 164) operating at a wavelength of 514 nm and around 120 mW power. The laser beam was passed through a water filled acoustic-optic cell exploiting the Debye-Sears effect to provide a frequency shift of 21.00 MHZ corresponding to ultrasonic waves travelling in water. It was necessary to shift the frequency particularly in case of the swirling flows described in section 3.3, see for example Whitelaw (1981), Crabb, Durão and Whitelaw (1980) and Durão and Whitelaw (1978), where
positive and negative velocities existed. Full details of the design of the cell are given by Durao (1976). The beams were separated by 71.7 mm, made parallel and focused to their intersection by a lens of 300 mm focal length and focused to a pin-hole in front of photomultiplier cathode (EMI 9815B). The laser and the optical components were secured to a fixed optical bench and the burner was located on a traverse mechanism which allowed translation in three directions with a precision of better than 0.1 mm. Table (2.2) summarises the principle characteristics of the optical arrangement.

### TABLE (2.2) PRINCIPLE CHARACTERISTICS OF OPTICAL COMPONENTS
(FORWARD SCATTER MODE)

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Half-angle between incident beam (ψ), (degrees)</td>
<td>6.78</td>
</tr>
<tr>
<td>Frequency difference (MHz)</td>
<td>21.0</td>
</tr>
<tr>
<td>Fringe spacing (line pair spacing), (μm)</td>
<td>2.15</td>
</tr>
<tr>
<td>Length of intersection volume at e⁻² location, (mm)</td>
<td>0.82</td>
</tr>
<tr>
<td>Diameter of part of intersection volume observed by photomultiplier (mm)</td>
<td>0.097</td>
</tr>
<tr>
<td>Calculated number of (stationary) fringes within e⁻¹ intensity level</td>
<td>57.0</td>
</tr>
<tr>
<td>Magnification of light collection arrangement, M</td>
<td>1.0</td>
</tr>
<tr>
<td>Velocimeter transfer constant, K⁻¹ (MHz/ms)</td>
<td>0.465</td>
</tr>
</tbody>
</table>
The Doppler signals were detected by the photomultiplier and demodulated by a swept-tuned spectrum analyser (Hewlett-Packard model 141 T/8552A/8553B) which used a system described by Durao, Laker and Whitelaw (1978). The spectrum analyser resolved the frequency components of the Doppler signal and could be used up to about 100 MHz Doppler frequency; poor signal-to-noise ratio could be accepted with low and high particle concentrations, but data acquisition and reduction were slow (see Melling 1975). The spectrum analyser measures the probability of the frequency (the number of bursts over same triggering level at certain frequency) at N frequencies equally spaced between the extreme lower and upper frequencies of the Doppler spectrum.

A microprocessor (Intersil model IM 6101) was used to evaluate the probability distribution of the Doppler frequency and calculate the mean and rms frequencies \( \bar{v}_D \) and \( \hat{v}_D \). The probability distribution was formed with one pulse per Doppler burst; on recognition of the arrival of a burst a flag is raised and to be reset by the microprocessor after a maximum time delay of 42 \( \mu \)s which is the sampling rate of the microprocessor. The mean and rms velocities were calculated from the first two moments \( \bar{v}_D \) and \( \hat{v}_D \) according to these relations, see Durst, Melling and Whitelaw (1976):

\[
\bar{U} = \lambda^* (\bar{v}_D - v_S) \quad (\text{m/s})
\]  

(2.25) 

\[
\hat{u} = \lambda^* (\hat{v}_D) \quad (\text{m/s})
\]  

(2.26)

Where \( v_S \) is the frequency shift of the Bragg cell and \( \lambda^* \) is the Doppler frequency/velocity conservation factor defined as: 

\( \lambda^* = (\lambda^* / 2 \sin \alpha) \). The microprocessor calculates the mean and rms
frequencies with the equations:

\[
\bar{v}_D = \frac{\sum_{i=1}^{N} P(v_i) \cdot v_i}{\sum_{i=1}^{N} P(v_i)} \quad (2.27)
\]

\[
\bar{v}_D = \left( \frac{\sum_{i=1}^{N} (v_i - \bar{v}_D)^2 \cdot P(v_i)}{\sum_{i=1}^{N} P(v_i)} \right)^{0.5} \quad (2.28)
\]

The time averaged number of droplets passing through the laser Doppler anemometer control volume was measured using a digital counter (type TSA 663612). The signal from the photomultiplier was passed to a high pass filter and to the oscilloscope. The high pass filter ensured the removal of the pedestal signal. The filter characteristics were known and did not influence the Doppler signal. The counter was connected to the delayed output of the oscilloscope, see figure 2.19. The threshold level of the oscilloscope was kept constant during an axial or radial traverse and provided a constant base for the counting procedure.

This arrangement of the laser doppler anemometry allowed the measurements of droplet velocity and droplet number-density in the unconfined spray flames in the initial region and up to approximately 240 mm downstream where the number of droplets provided good probability distributions with minimum error. Measurements could not be obtained in the immediate vicinity of the fuel nozzle, i.e. \(X < 30\) mm, due to dense population of fuel droplets. The droplet velocities, especially, near the nozzle location, were influenced also by their size and injection velocities, gas and droplet properties and processes of collision and break up. The anemometer measured the droplet velocities which were the same as the gas velocity only when the droplet was sufficiently small to follow the gas velocities.
2.4.2 Optical arrangement with backward-scattered light

Due to the projection of the rotating cup atomizer inside the combustion chamber, see chapter 3, backward scattered light was used to measure the droplet and isothermal flow velocities inside the combustor as shown in figure 2.20, see also Durst, Melling and Whitelaw (1976) and El-Banhai (1979). The laser power was increased to around 800 mW to overcome the low intensity of the scattered light in the backward direction. The transmitting optical components were the same as used with forward scattered light, but a focusing lens of 500 mm focal length was used for the tangential velocity measurements because the height of the observation window did not allow measurement near the centreline with the 300 mm focal length lens. The receiving optical components comprised a mirror box, equipped with two parallel mirrors inclined 45 degrees to the laser beam direction, a lens of 300 mm focal length and a photomultiplier (EMI model 7815B). The laser and optical components were secured to an optical bench locating on a milling table which allowed translation in two horizontal directions with a precision of better than 0.1 mm. The milling table also allowed a small range of translation in the vertical direction. Table 2.3 indicates the main characteristics of the optical arrangement.

The signal processing system was similar to that described in connection with the forward scatter mode.

Droplet velocity measurements were obtained for the preheated flames described in chapter 3 and were confined to the spray region. It was impossible to measure velocities in the unpreheated flames due to the soot layer which formed on the quartz windows. At locations away from the spray boundaries, velocity
measurements were not possible due to the significantly reduced number and size of fuel droplets resulting from the high evaporation and combustion rates.

Isothermal air flow velocity measurements within the combustion chamber, for air swirl numbers of 1.4, 0.81 and 0.47, were also obtained to aid understanding of the main aerodynamic features of the flow. They were also used in the validation study of the calculation procedure presented in chapter 4. Corn-oil seeding particles were used in connection with this set of isothermal flow velocity measurements because it produced a convenient particle concentrations with such high flow rates and swirl.

TABLE (2.3) PRINCIPLE CHARACTERISTICS OF OPTICAL COMPONENTS (BACKWARD SCATTER MODE).

<table>
<thead>
<tr>
<th></th>
<th>Focusing lens diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>300 mm</td>
</tr>
<tr>
<td>Half-angle between incident beam (α), (degrees)</td>
<td>6.54</td>
</tr>
<tr>
<td>Frequency difference, (MHz)</td>
<td>21.0</td>
</tr>
<tr>
<td>Fringe spacing (line pair spacing), (μm)</td>
<td>2.21</td>
</tr>
<tr>
<td>Length of intersection volume at e^2 location, (mm)</td>
<td>0.565</td>
</tr>
<tr>
<td>Diameter of part of intersection volume observed by photomultiplier, (mm)</td>
<td>0.065</td>
</tr>
<tr>
<td>Calculated number of (stationary) fringes within e^2 intensity level</td>
<td>37.0</td>
</tr>
<tr>
<td>Velocimeter transfer constant, K^{-1} (MHz/ms)</td>
<td>0.452</td>
</tr>
</tbody>
</table>
2.4.3 Error analysis

Uncertainties in the velocity measurements can result from:
optical alignment, signal processing, gas medium and scattering
data. They have been discussed, for example, by Self and
Whitelaw (1976), Durst, melling and Whitelaw (1976) and Pitz et al
(1981). In the following paragraph the important errors related to
this work are discussed.

The signal processing based on averaging of a number of
individual frequency measurements may be subject to errors due to
gradient and particle broadening, the number of signals used to form
the average and to non zero correlation between number of particles
and flow velocity. The dimensions of the scattering volume were
small, therefore its alignment with the predominant velocity gradient
was not long. This led to the neglect of the first source of error.
The second error is caused by the use of discrete representation of
the probability density function of the Doppler frequency to
calculate the mean and variance frequencies. This error was
quantified by Yanta (1973), for the mean and rms velocities, as given
by the following formulae:

\[ e_\mu = \frac{2}{\sqrt{N}} \cdot \left( \frac{\bar{v}_D}{\bar{v}_D} \right) \]  
(2.29)

\[ e_\sigma = \sqrt{\frac{2}{N}} \]  
(2.30)

where \( e_\mu \) and \( e_\sigma \) are the error in the mean and rms values
respectively, and \( \bar{v}_D \) and \( \bar{v}_D \) are the mean and rms values of the
Doppler frequency. In the present measurements the number of the
samples, \( N \), used to form the averages ranged from 2500 to 73000 and
results in uncertainties of less than 1.7% in the mean velocity and
2.8% in the rms values at the maximum turbulence intensity recorded.

The third source of error is velocity bias, which arises because the rate at which particles arrive in the measuring volume, \( \dot{R} \), and is proportional to the volumetric flux of the fluid through the measuring volume, see Taylor (1981):

\[
\dot{R} = \mu \cdot A_v \cdot [U(t)]
\]

(2.31)

where \( \mu \) is the instantaneous scattering-particle density \( (m^3) \) and \( A_v \) is the cross-sectional area of the measuring volume projected in the direction of instantaneous velocity vector, \( U(t) \).

As analysed by MacLaughlin and Tiederman (1973), Dimotakis (1976), Buchhave (1976) and Wilson (1978), the probability density function of the individual velocity measurements may be biased towards higher velocity because the rate of realizations depends on the instantaneous velocity. However, it was shown by Durão (1976), Tiederman (1977) and Durão and Whitelaw (1979) that, if the signal validation criterion depends on the signal amplitude, a "reverse" bias exists. The effect depends on the laser power, the speed of response of the photomultiplier tube and the presence or absence of the frequency shift. Durão, Laker and Whitelaw (1980) suggested that this bias tends to cancel that of the velocity-particle density correlation and the probability distribution formed with one pulse per Doppler burst will lead to the correct values of mean velocities for arrangements with turbulent, instrumentation and particle time scales. The results of Durão and Whitelaw (1979), El-Banhawy and Whitelaw (1981), Habib and Whitelaw (1979) and Taylor and Whitelaw (1980), using the same instruments as described in this section, suggest that the velocity biasing is cancelled.
In the turbulent reacting flows, the variation in the refractive index with position and time can lead to dispersion of the laser light and movement of the focal region in space and in time. This can result in reduced signal quality, reduced number of signals per unit time and a large control volume than would apply in a non-reacting flows. El-Banhawy and Whitelaw (1981), Taylor and Whitelaw (1980) and Pitz et al (1981) found this effect to be negligible.

2.5 Summary and concluding remarks

In this chapter the measurement techniques were described and investigated for the use in kerosene spray flames with a quantification of their possible errors. The temperature results for the unconfined spray flames were obtained by bare-wire thermocouples of 300, 80 and 40 μm wire diameter and a suction pyrometer. The results allowed an assessment of the accuracy of each method and what sort of averaging was produced. The temperature distributions obtained by the suction pyrometer indicated that the 2 mm location from the end of the pyrometer shield leads to the highest values compared to 5 and 10 mm locations. This position was realistic for minimizing the radiation error and allowing high heat transfer between the gases and the hot junction. It was also shown that the suction velocities in the suction pyrometer around the hot junction must be very high (> 250 m/s) at high temperatures to compensate the errors associated with the thermocouple. The comparison of the results obtained by the different bare-wire thermocouples and the suction pyrometer indicated that the fine bare-wire thermocouple measures the higher values of gas temperatures with a small uncertainty. It was indicated, also, that the fine wire thermocouples
measured the time-mean gas temperature to an accuracy of within $\pm 2\%$
while the large thermocouple and the suction pyrometer recorded a
different mean represented by: $\overline{\left(\frac{h_{c}T_{q}}{h_{c}}\right)}$.

A water cooled stainless-steel probe was used to collect
gas samples from the flames for analysis and it is known that the
quenching of the chemical reaction at the probe tip and isokinetic
sampling are important to draw a representative sample. The
investigation of these factors implied that the error in the measured
species concentrations due to quenching is insignificant and the
error due to the non-isokinetic sampling represent approximately $+3 - 7\%$. It was estimated that the Favre averaged species concentrations
could be measured to an accuracy of within $+9\%$.

The laser Doppler anemometer was used in two different
optical arrangements according to the flow configurations. A brief
description for these arrangements was presented with a discussion of
the important sources of errors involved in the results. It was shown
that the uncertainties in the mean velocity is less than 1.7\% and
2.8\% in the rms values at the maximum turbulence intensity recorded.
Fig. 2.1  300 μm bare thermocouple.

Fig. 2.2  40 μm and 80 μm fine wire thermocouples.
Fig. 2.3 Electrical circuit and temperature equipment.
Plate (2.1) High vacuum apparatus used for radiation correction.
Fig. 2.4 Radiation error calibration; 80 μm thermocouple.
Fig. 2.5 Radiation error calibration: 40 μm thermocouple.
Fig. 2.6. Diagram of the suction pyrometer.
Fig. 2.7. Arrangement for temperature measurement by the suction pyrometer.
Fig. 2.8. Effect of the bead position inside the shield on the centre-line temperatures.
Fig. 2.9. Effect of the bead position inside the shield on the radial temperature profiles.
Fig. 2.10. Effect of the suction velocity on the measured temperatures.
Fig. 2.11a. Comparison of the centre-line temperature distributions measured in flame 1.
Fig. 2.1b. Comparison of centre-line temperature distributions measured in flame 2.
Fig. 2.12. Comparison of radial temperature distributions measured in flame 1 and 2.
Fig. 2.13. Water-cooled gas sampling probe.
Fig. 2.14. Arrangement of the sampling system and gas analysis equipments.
Fig. 2.15. Effect of probe cooling water on quenching of the gas sample.
Fig. 2.16. Effect of suction velocity on species concentration (θ=90°)
Fig. 2.17. Effect of probe inclination on species concentration.

Fig. 2.18. Effect of suction velocity on CO₂ concentration
Fig. 2.19. Laser Doppler anemometry arranged to collect forward-scattered light.
Fig. 2.20. Laser Doppler anemometry arranged to collect backward-scattered light.
CHAPTER 3

EXPERIMENTS IN CONFINED AND UNCONFINED KEROSENE SPRAY FLAMES

3.1 Introduction

Many practical combustion devices, ranging from gas turbines to utility boilers, involve the burning of liquid fuel which is sprayed directly into the primary combustion region. The injected liquid shatters into droplets which move relative to, and exchange momentum with, the combustion air and already burnt gases. The droplets heat up, vaporize and decelerate. Since the fuel and air enter the combustor separately, the fuel vapour and air must mix to within combustible limits by a turbulent mixing process before the fuel will burn. Quantification of the extent to which the spray characteristics, including evaporation rate, trajectories and velocities of the fuel droplets and droplet size, contribute to the overall combustion process represents the major goal in the current research. This work provides new information of temperature, species concentrations and droplet velocity distributions, for a range of confined and unconfined kerosene spray flames, which improve the understanding of the spray characteristics and combustion. In addition, the results aid the evaluation of the calculation methods with their assumed models for turbulence, combustion, radiation and sprays. The physical properties of the kerosene fuel used in this study are given in table 3.1.

The work presented in this chapter describes the measurements which were achieved in free jet kerosene spray flames and an axisymmetric combustor. For the unconfined flame experiments,
### TABLE (3.1) PHYSICAL PROPERTIES OF KEROSENE FUEL.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical formula</td>
<td>( \text{C}<em>{12}\text{H}</em>{26} ) approx.</td>
</tr>
<tr>
<td>% composition by mass (ultimate)</td>
<td></td>
</tr>
<tr>
<td>Carbon</td>
<td>85.8</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>14.1</td>
</tr>
<tr>
<td>Sulpher</td>
<td>0.1</td>
</tr>
<tr>
<td>Density at 1 atm and 15.6 °C (Kg/m(^3))</td>
<td>780.0</td>
</tr>
<tr>
<td>ASTM distillation</td>
<td></td>
</tr>
<tr>
<td>Initial boiling point</td>
<td>165.0 °C</td>
</tr>
<tr>
<td>10% volume</td>
<td>175.0 °C</td>
</tr>
<tr>
<td>50% volume</td>
<td>200.0 °C</td>
</tr>
<tr>
<td>90% volume</td>
<td>230.0 °C</td>
</tr>
<tr>
<td>Final boiling point</td>
<td>255.0 °C</td>
</tr>
<tr>
<td>Mean specific heat capacity between 0 - 100 °C (KJ/Kg K)</td>
<td>2.09</td>
</tr>
<tr>
<td>Specific latent heat at mid boiling point (KJ/Kg)</td>
<td>226.0</td>
</tr>
<tr>
<td>Calorific value at 15 °C (MJ/Kg)</td>
<td></td>
</tr>
<tr>
<td>Gross</td>
<td>46.5</td>
</tr>
<tr>
<td>Net</td>
<td>43.5</td>
</tr>
<tr>
<td>Kinematic viscosity (mm(^2)/s)</td>
<td></td>
</tr>
<tr>
<td>at 15.6 °C</td>
<td>2.1</td>
</tr>
<tr>
<td>at 37.8 °C</td>
<td>1.45</td>
</tr>
<tr>
<td>Stoichiometric air requirements</td>
<td></td>
</tr>
<tr>
<td>Mass of dry air/mass of fuel</td>
<td>14.96</td>
</tr>
<tr>
<td>Volume of dry air (1 atm, 0 °C)/ mass of fuel (m(^3)/Kg)</td>
<td>11.36</td>
</tr>
</tbody>
</table>
a twin-fluid atomizer was used with kerosene fuel to generate the flames which burned in still air with vertical axes. Measurements of droplet mean velocity and number-density, mean temperature and concentrations of oxygen, carbon monoxide, carbon dioxide and unburned hydrocarbon, were obtained in two flames with droplets characterized by Sauter mean diameters of 80 and 90 μm. The atomizer used is directly relevant to gas turbine combustors and the results are intended to allow conclusions about the influences of droplets in diffusion-flame combustion. Experiments were then conducted in an axisymmetric combustor. A spinning cup-atomizer, located axially, generated symmetrical sprays with near mono-disperse mean-droplet diameters which depended on the rotational speed and the fuel mass flow rate. The combustion air was preheated to different levels and supplied to the combustor through an annular, and interchangeable, vane swirler. The measurements quantify the influences of the spray mean-droplet diameter and combustion air swirl and inlet temperature on the flame characteristics and performance.

This chapter begins with a description of the unconfined flames experiment together with an analysis of related results, section 3.2. The details of the measurements carried out in the axisymmetric combustor are presented and discussed in section 3.3. Each of the two sections is followed by a summary of main conclusions.

3.2 Measurements in unconfined spray flames

This section provides a description of the experimental study of unconfined flames. The burner configuration, flow conditions and the experimental program are described in subsection 3.2.1 and
the results are presented and discussed in subsection 3.2.2. Subsection 3.2.3 describes the gas temperatures inferred from the enthalpy balance, local mixture fraction and air to fuel ratio based on the measured species concentration and the section ends with a brief statements of the more important conclusions.

3.2.1 Burner configuration and operating conditions

The twin-fluid atomizer used to generate the fuel sprays is shown in figure 3.1 and is similar to that described by Pounti, Hutchinson and Whitelaw (1980), Khalil and Whitelaw (1977), Chigier et al (1972) and Fraster et al (1957). The liquid kerosene was supplied to the atomizer at a gauge pressure of approximately 0.68 bar through three 4 mm diameter pipes located radially 120 degrees apart and discharged vertically through a 0.5 mm centrally located fuel nozzle. The atomization air, at the pressures shown in table 3.2, was supplied to an annular shroud, through three 6.3 mm diameter pipes located axially and 120 degrees apart, and passed through the shroud to impinge on the fuel jet and cause atomization. The result was a solid spray with a cone angle of approximately 20 degrees. The flow rates of the atomization air and kerosene were measured with rotameters and the pressures with standard pressure gauges. The flow rate of each fluid was regulated by means of a valve and could be varied independently. Flame stability was ensured by a 300 mm diameter disc located 2 mm above the exit plane of the nozzle. Figure 3.2 presents a line diagram of the flow system and the arrangement of the unconfined kerosene spray flame experiment.

The measurements were obtained with the operating conditions given in table 3.2. The Sauter mean diameter, SMD, of the
droplets was estimated from the correlation of Fraster et al (1957) and depended on the mass ratio of atomization air to kerosene.

### TABLE 3.2 OPERATING CONDITIONS OF UNCONFINED SPRAY FLAMES.

<table>
<thead>
<tr>
<th>Flame</th>
<th>(\dot{m}_f) (Kg/h)</th>
<th>(\Delta P_f) (bar)</th>
<th>(\dot{m}_a) (Kg/h)</th>
<th>(\Delta P_a) (bar)</th>
<th>SMD ((\mu)m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flame 1</td>
<td>3.82</td>
<td>0.68</td>
<td>0.725</td>
<td>4.08</td>
<td>80</td>
</tr>
<tr>
<td>Flame 2</td>
<td>3.82</td>
<td>0.68</td>
<td>0.654</td>
<td>3.40</td>
<td>90</td>
</tr>
</tbody>
</table>

\(\dot{m}_f\), \(\Delta P_f\) and \(\dot{m}_a\) and \(\Delta P_a\) are the mass flow rate and pressure drop for the kerosene fuel and atomization air respectively.

The experimental program was directed towards quantifying the performance and accuracy of temperature and concentration measurements in the kerosene spray flames, as explained in chapter 2 and, at the same time, the provision of new information of unconfined diffusion kerosene spray flames. Flame characteristics are interpreted from the measured droplet velocity, droplet number intensity, mean temperature and concentration of different species \(\text{O}_2\), \text{CO}, \text{CO}_2\,\text{and UHC}\). The measured mean temperatures are compared with those obtained from mean energy balance with a discussion of the reasons for the discrepancies.

#### 3.2.2 Presentation and discussion of results

In this subsection the droplet characteristics are considered in addition to the presentation and discussion of the temperature and species concentration results.
(i) Droplet number density and velocity

Measurements of time averaged droplet number-density were possible in the spray region between axial locations of 75 and 270 mm from the exit plane of the nozzle for both flames. Far downstream the droplet concentrations were negligibly small and, in the upstream region, the very high concentrations of droplets made measurements difficult and in the immediate vicinity of the spray exit, impossible. The centreline distributions of the normalized droplet-number density \( \frac{N}{N_0} \) of the two flames are shown in figure 3.3 and indicate the high droplet concentration at the first point of measurements, \( X = 70 \) mm, and the subsequent rapid decrease. The radial profiles of figure 3.4 show the high concentration in the core of the spray and the expected decay towards the edge. The rapid gradients reduce with downstream distances and large radii as the evaporation and combustion begin to exert their influences and the spray spreads. This accords with physical descriptions for the structure of the spray flames, see for example, McCreath et al (1973), Chigier (1976), Onuma et al (1975,1977) and Tuttle et al (1976). The results are consistent with the temperature and species concentration distributions of figures 3.7 and 3.8 which show lower temperature values of around 150 °C and negligible combustion gases at the upstream region and higher values elsewhere. Further downstream and at larger radii, the droplet density continues to decrease to zero on complete evaporation and the main reaction zone will move from the boundary of the flame, at the upstream, to the centreline as indicated in figure 3.9 and 3.10. It is also evident from figure 3.4 that the second flame (SMD = 90 \( \mu \)m) is broader than the first (SMD = 80 \( \mu \)m) due to the lower atomization air velocity and bigger droplet size of the second flame which causes lower ratios of drag to
inertia and longer evaporation time as indicated by Tuttle et al (1976) and Mellor (1976).

The mean axial velocities and rms of the fluctuations of the droplet were measured with the laser Doppler anemometer, in the spray region where the droplet population yielded good probability distributions; this was possible at locations up to an axial distance of 240 mm. Measurements were difficult in the immediate vicinity of the fuel nozzle, i.e. X < 30 mm, due to the dense population of fuel droplets. The droplet velocities, especially at near nozzle locations, were influenced by their size, injection velocities, gas and droplet properties and process of collision and break up. The anemometer measured the droplet velocities which were the same as the gas velocities only when the droplets were sufficiently small to follow the gas flow and this could only happen at downstream locations and large radii. Figure 3.5 shows the centreline distributions of time averaged mean axial velocity, rms of the corresponding fluctuations and the turbulence intensity for the droplets of the two flames. The mean velocity rises rapidly to a maximum at the upstream locations of X = 30 mm, probably caused by the acceleration of the smaller diameter droplets due to the high injection velocity of the atomization air. The mean velocity decays at a higher rate over a short axial distance from the fuel nozzle (i.e. up to X = 80 mm), beyond which the deceleration of the droplets becomes very slow. The reasons for the high rate of change of velocity upstream and the slower one downstream can be explained when consideration is given to the equation of motion of the droplets. This can be written in one-dimensional form as:

\[
\frac{dU_p}{dt} = - \left( 3C_p \rho_p / 4 \rho_g D_p \right) (U_p - U_g) \cdot |(U_p - U_g)| - G \quad (3.1)
\]
where $U_p$, $\rho_p$, $U_g$, and $\rho_g$ are the velocity and density of the droplet and gas respectively, $C_D$ is the drag coefficient, $D_p$ is the droplet diameter and $G$ is the acceleration due to gravity. In the above equation, the first term on the right hand side is negative or positive depending on whether the droplet velocity is greater than or less than the gas velocity while the second term is negative. Chigier et al (1972) indicated that the gas velocity decays faster than the droplet velocity and it is likely that for $X < 90$ mm, where the temperatures are low and the evaporation is absent, the gas velocity achieves lower values than the droplet velocity which results in a negative sign for the drag term with a consequent high droplet deceleration. However, further downstream, the temperature rise due to evaporation and combustion causes an increase in the gas velocity which reduces the first term on the right hand side of equation (3.1) or makes it positive to yield very low deceleration. Comparison between the centreline distributions of mean axial velocity of the two flames, as shown in figure 3.5, indicates a relatively large differences in the upstream region with a common trend of increased velocity with decrease in Sauter mean diameter. This could be attributed to the presence of smaller droplets at near nozzle locations with their lower ratio of inertia to drag which results in higher velocity values. Far downstream, differences are expected due to the increased evaporation rate and earlier combustion associated with the smaller droplet size flame.

The centreline rms of velocity fluctuations and the corresponding intensity distributions of figure 3.5 show that relatively high values are attained at near nozzle locations. These are associated with the process of droplet collision and break up,
spreading of the droplet size distribution and the difference in their injection velocities and the turbulent gas flow. The location of the maximum rms velocities occur at an axial location of around 40 mm for both flames and correspond to the points of inflection in the mean velocity profiles. The increase in the centreline rms values with decreased droplet mean diameter may be attributed to the increase in the combustion intensity and the expected reduction in the fuel spray spreading angle. The turbulence intensity for both flames varies from 0.15 to 0.21 along their centrelines.

Radial profiles of the axial mean velocity, rms of fluctuation velocity and their turbulence intensity at different axial stations for both flames are presented in figure 3.6. In general, the mean velocities of the first flame are higher than those of the second for the reasons discussed above. At X = 50, 75 and 100 mm, the velocities are high in the core of the spray and suddenly decrease towards the edge of the flame due to the high drag forces acting on the droplets and the onset of combustion at the spray boundaries. At downstream stations, where evaporation and combustion take place at all radii with consequent further reduction in the drag force, the rate of the deceleration of the droplet mean velocity reduces and the velocity becomes near constant. The locations of the maximum rate of change of the mean velocities (dU/dr) occur at radii of approximately 5 and 7 mm for the first and second flame respectively and are consistent with the maximum values of the rms velocity fluctuations. At the upstream stations, the turbulence intensities are higher at the flame boundaries, ~ 60% and 40% for the first and second flame respectively and, probably, stem from the instabilities caused by the high pressure atomization air and the spread of the droplet size distributions. The rms of velocity
fluctuations and the turbulence intensities at the downstream stations have lower values and generally increase outwards.

(ii) Mean temperature and species concentration

The centreline distributions of the mean temperature and species concentration of oxygen, carbon monoxide, carbon dioxide and unburned hydrocarbon expressed as $C_3H_8$ are presented in figures 3.7 and 3.8 for the first and second flame respectively. As can be seen the temperatures are very low and less than 150°C and the hydrocarbons are high near the fuel nozzle; this is due to incomplete atomization, the density of the spray and the absence of the chemical reaction as indicated by the droplet number density profiles of figures 3.3 and 3.4. Consistent with the non-existance of combustion, the oxygen and hydrocarbons have their highest values, while the carbon monoxide and carbon dioxide concentrations are very small (i.e. < 0.6% at $X = 100$ mm); these small values of the combustion products are likely to be entrained from the reaction zone at the spray boundary. As the evaporation of the droplets progresses with downstream distance, the combustion occurs at the centreline and the temperature increases rapidly: correspondingly, the oxygen and hydrocarbon concentrations decrease and the carbon dioxide and carbon monoxide concentrations increase. The carbon monoxide reaches its highest value of 10.5% at a temperature of the order of 1250°C while the oxygen decreases to a minimum value of 0.3%. At this stage, the combustion is intense, the temperatures are high and the reaction zone has moved towards the axis of the flame. It is obvious that the maximum values for carbon monoxide concentration and the minimum values for oxygen have moved slightly downstream for the lower atomization air to fuel ratio flame (flame 2) due to the reduction in the evaporation rate associated with larger fuel droplets. Further
downstream, the carbon monoxide decreases due to oxidation to carbon
dioxide, hydrocarbons decrease to their lowest values while the
carbon dioxide and temperature increase to their highest values of
around 8.5\% and 1500 °C respectively. The very slight increase in the
oxygen concentration at X = 500 mm is due to the entrainment of the
surrounding air. It is expected that the temperature and carbon
dioxide, at the furthest downstream for axial distances larger than
500 mm, will decrease again due to the dilution by entrainment of the
surrounding air at the edge of the flame and the oxygen concentration
will increase steadily: this is confirmed by the radial profiles
distributions of figures 3.9 and 3.10.

Figures 3.9 and 3.10 present the radial distributions of
mean temperature and concentrations of carbon dioxide, carbon
monoxide, oxygen and unburned hydrocarbon at the axial stations 150,
300 and 500 mm from the fuel nozzle for the two flames. At
X = 150 mm, combustion is still in its initial stages with the main
reaction zone at the spray boundary; the temperature has achieved a
centreline value of 400 °C and increase towards larger radii so that
heat is transferred towards the centreline to aid vaporization of the
droplets and decomposition of the hydrocarbons. The concentration of
oxygen and hydrocarbons have high values at the centreline
(i.e. O_2 ≈ 15\% and UHC ≈ 10\%), whereas the temperature and CO_2 and CO
concentrations have correspondingly small values. The high values of
oxygen concentration in the central region of the flame are mainly
due to air from the atomizer and the absence of combustion. Away from
the centreline, with the combustion taking place at the spray
boundary, the oxygen concentration decreases progressively to a
minimum value of ≈ 2\% at R = 17.5 mm. The carbon monoxide increases
to attain a peak value of approximately 7\% followed by a rapid
increase in the temperature, and indicates that reaction is very strong in this region. At these high temperatures, the carbon monoxide reacts with oxygen to form carbon dioxide and the hydrocarbon concentration continues to decrease. Near to the edge of the flame, the oxygen concentration increases rapidly due to entrainment of the surrounding air and the temperature and concentrations of carbon monoxide and carbon dioxide decrease for the same reason. At $X = 300$ mm, the evaporation of the droplets is complete, as shown by the higher values of the carbon monoxide and low hydrocarbon values at the centreline, and the reaction zone has moved from the boundary to the core of the flame. The concentration of carbon monoxide shows a nearly constant value in the region $0 < R < 12$ mm because the corresponding temperatures are lower than those required for the oxidation of carbon monoxide. At larger radii, the temperature rises and permits oxidation of CO to CO$_2$ with consequent rise in temperature and carbon dioxide towards their peak values. At $X = 500$ mm, the temperature has a maximum value of the order 1500 °C at the centreline and the carbon dioxide has a near uniform value close to the centreline which may be attributed to the balance between the CO$_2$ formed from the oxidation of CO, which increases the value of CO$_2$, and the dilution of the flame by the entrainment of air. This is consistent with the low values of CO in this region, especially with the lower atomization air flame.

Visual observation of the two flames showed that the length of the first flame was of the order of 650 mm and 100 mm in diameter, while the second flame was slightly longer and wider; this is consistent with the delayed evaporation, mixing and combustion associated with the bigger droplets. Both flames were lifted about 50 mm above the fuel nozzle. Comparison of the measurements for the
two flames has shown that, as expected, the smaller initial droplet size results in a more rapid rise in temperature and in the generation of the products of combustion. Figures 3.7 and 3.8, for example, show that the centreline temperatures begin to differ by 10 mm from the exit of the nozzle and that, 500 mm downstream, the first flame has a temperature more than 100 °C greater than the second flame. Consistent with this result, the concentration of UHC has smaller values for the first flame and the CO and CO₂ are formed more rapidly due to the shorter times of fuel evaporation, turbulent mixing and chemical reaction associated with the smaller droplet diameter. The first flame has a higher rate of CO oxidation corresponding to the very rapid achievement of high temperatures. Similar trends are revealed by the radial distributions of figures 3.9 and 3.10.

The addition of preheat, as in gas turbine, will lead to more rapid evaporation, see section 3.3, smaller flames and probably to smaller differences between the temperature and concentration characteristics of the flames downstream of the burner with the operating conditions of table 3.2.

3.2.3 Computation of temperature, mixture fraction and air/fuel ratio from gas analysis

This subsection describes the method used to calculate the local mean temperature, mixture fraction and air to fuel ratio for the previous two flames, see table 3.2, based on the measured species concentration, together with presentation of these results. The calculations aid understanding of solutions to differential equations, with their combustion models which often assume fast chemical reaction or chemical equilibrium. The calculations also
evaluate the temperatures inferred by local enthalpy balance based on density-weighted gas analysis for liquid fuel combustion.

The calculations were based on the assumption that the fuel injected at each point is liquid kerosene ($C_{12}H_{26}$) and the unburned hydrocarbons exist in this form. However, the hydrogen concentration has not been measured and can influence the calculated results as demonstrated by Noyce et al (1981), see also equation (3.4); the hydrogen concentration was estimated from the measured CO concentration using these empirical correlations given by Lyon et al (1975); i.e.

\[
R_{H_2} = R_{CO}[0.2(100. R_{CO})^{0.16}] \quad (3.2a)
\]

for $R_{CO} \leq 0.091022$

\[
R_{H_2} = R_{CO}[0.0147(100.R_{CO})^{1.342}] \quad (3.2b)
\]

for $R_{CO} > 0.091022$

The $H_2O$ concentration in the sample was obtained from the conservation of hydrogen atoms and knowledge of carbon to hydrogen ratio as explained by Clifford et al (1977). Since all carbon containing compounds (CO, CO$_2$, UHC) are analysed and the hydrogen concentration is calculated, the remaining hydrogen can be assigned as water. This method yields a simple quadratic equation for the "wet" $H_2O$ concentration of the form:

\[
R_{H_2O} = \frac{\sqrt[3]{(100 + 4[(R_{CO} + R_{CO_2})(13/12) - R_{H_2}])} - 50}{5} \quad (3.3)
\]

This value of "wet" water, $R_{H_2O}$, was then used to convert all of the dried species to the wet. This method of calculation based on the
above assumptions implied uncertainties of around 10% in the overall wet sample analysis.

The theoretical mean temperature was calculated, iteratively, from the heat balance between the reactants and products neglecting the radiation heat losses from the flames and radical concentrations using the equation:

\[ C_{bf} \cdot H_{fu} = \sum_{i=1}^{n} C_i (\Delta h_i) + C_{H_2} \cdot H_{H_2} + C_{CO} \cdot H_{CO} \]  \( (3.4) \)

where \( C_{bf} \) is the burnt fuel (Kmol), \( H_{fu} \) is the net calorific value of the fuel (MJ/Kmol), \( C_i \) and \( \Delta h_i \) are the concentration (Kmol) and enthalpy difference (MJ/Kmol) respectively for the product species (CO, CO\(_2\), O\(_2\), N\(_2\), H\(_2\)O and H\(_2\)) and \( C_{H_2} \), \( C_{CO} \), \( H_{H_2} \) and \( H_{CO} \) are the concentration and calorific value of hydrogen and carbon monoxide respectively. The reference temperature for the enthalpy balance equation (3.4) was 25 °C and the enthalpies at different temperatures were taken from Haywood's (1972) thermodynamic tables. The local values of mixture fraction (\( \tilde{f} \)), defined as the mass fraction of the fuel both burnt and unburnt, and the air to fuel ratios were calculated from the analysis of the fuel and oxygen in the products.

Figures 3.11 and 3.12 present axial and radial distributions of the calculated mixture fraction, air/fuel ratio, temperature and the measured temperature distributions for the two flames. It should be noted that the computed values shown in these figures are density-weighted mean values since they are based on density-weighted concentrations while the measured temperatures are close to the time mean average as explained in the previous chapter. It is seen from the axial distributions of the calculated and measured temperatures of the two flames (figure 3.11) that, in the
upstream region where the temperatures are very low, the discrepancies are small (≈ 40 °C) and increase gradually to achieve maximum values of around 220 °C. The radial profiles shown in figure 3.12 similarly indicate high differences at downstream stations. An important reason for these discrepancies is the radiation heat transferred from the flame to the surrounding, which is proportional to the fourth power of the temperature, and may be expected to increase as the flame temperatures rise. The second reason is due to the different forms of averaging the two quantities which are related approximately through this expression:

$$\tilde{T}_C = \bar{T}_m + \rho \bar{T}' / \bar{\rho}$$

where $\tilde{T}_C$ and $\bar{T}_m$ are the calculated and measured temperature, $\rho'$ and $T'$ are the fluctuations of the density and temperature respectively and $\bar{\rho}$ is the conventional average of density. The last reason is that any attempt to calculate the properties of a flame without consideration of the fluctuations in the temperature and species concentrations is unlikely to provide accurate answers. It is well known, see for example Jones and Whitelaw (1981,1982), that the calculation of reacting flows, based on the solution of differential equations, requires consideration of the magnitude and form of the fluctuations of at least one scaler property and that the links between properties, even in diffusion flames, requires knowledge of a single or joint probability density function and equations of the form:

$$\tilde{\psi} = \int \psi(f) \, \mathcal{P}(f) \, df$$

where $\tilde{\psi}$ is the density-weighted value of concentration or enthalpy,
$\Phi(f)$ is the variation of the property with mixture fraction and $P(f)$ is the probability distribution.

The calculated density-weighted mixture fraction and air/fuel ratio shown in figure 3.11 indicate that the mixture fraction is high along the centreline and decreases at downstream axial distances of around 400 mm. The air/fuel ratio shows high values at the upstream followed by a decrease and subsequent increase at downstream distances of around 350 mm. The downstream radial profile stations of $X = 300$ and 500 mm, figure 3.12, indicate high mixture fractions at the centreline and decrease outwards in contrast to the air/fuel ratio. The same trends are shown at larger radii at the upstream station of $X = 150$ mm while the core of the flame has opposite trends consistent with those shown at the upstream along the centreline.

3.2.4 Concluding remarks

The following important findings and conclusions may be extracted from this section:

1. The twin fluid atomized spray is found to consist of a cold core characterized by high concentrations of droplets, high velocities, low temperatures and low concentration of combustion species. Near the periphery, however, evaporation is accelerated, velocities are low and the primary reaction zone is situated near the spray boundary.

2. High values of the rms of velocity fluctuations coincide with the initial spray regions and inflections in the mean velocity profiles. The former is due partly to the effects of droplet collision, break up and turbulent gas flow and
partly to the spread in velocity associated with the range of droplet sizes.

3. Burning occurs preferentially in regions of high temperature and advantageous fuel air ratio. There are two main reaction zones; one off the centre-line near to the flame root and another at greater axial distances and more extensive. This is demonstrated by high temperatures, maximum carbon monoxide, minimum oxygen and low unburned hydrocarbon values which occur on the centreline in downstream regions and off the centreline in upstream regions.

4. The temperature and concentration measurements show changes as a result of vaporization, which proceeds chemical reaction, and initial entrainment of hot gases from the flame front followed by subsequent entrainment of the surrounding air.

5. The increase of the atomization air flow rate results in spray with smaller droplet size and less spreading and a shorter flame. The former is due to increased interaction between the atomizing air and fuel droplets, while the latter stems from the shorter time of evaporation and combustion associated with smaller droplets.

6. The flame with the smaller initial droplet size has the more rapid temperature rise and generation of combustion products due probably to the accelerated evaporation, increased turbulent mixing and chemical reaction.

7. The discrepancies between the measured temperatures and those calculated from the enthalpy balance equation based on measured species are satisfactory.
Fig. 3.1. The twin fluid atomizer used for the unconfined flame experiment.
Fig. 3.2. Flow diagram and arrangement of equipment for the unconfined flame experiment.
Fig. 3.3 Axial distributions of normalized particle-number density.

(N₀ = number of droplets at first point of measurements.)
Fig. 3.4. Radial distributions of normalized particle number density. 
\( N_0 = \text{number of droplets at the centreline.} \)
Fig. 3.5. Axial distribution of mean axial velocity, corresponding normal stress and turbulence intensity for flame 1 & 2.
Fig. 3.6. Radial profiles of mean axial velocity, rms of velocity fluctuation and turbulence intensity for flame 1 & 2.
Fig. 3.7. Axial distribution of mean temperature and concentrations of CO$_2$, CO, O$_2$ and UHC for flame 1.
Fig. 3.8. Axial distribution of mean temperature and concentrations of $CO_2$, $CO$, $O_2$ and UHC for flame 2.
Fig. 3.9. Radial profiles of mean temperature and concentrations of \( \text{CO}_2 \), \( \text{CO} \), \( \text{O}_2 \) and UHC for flame 1.
Fig. 3.10. Radial profiles of mean temperature and concentrations of CO₂, CO, O₂ and UHC for flame 2.
Fig. 3.11. Axial distributions of measured temperature and calculated temperature, mixture fraction and air/fuel ratio based on gas analysis.
Fig. 3.12a. Radial profiles of measured temperature and calculated temperature, mixture fraction and air/fuel ratio based gas analysis, flame 1.
Fig. 3.12b. Radial profiles of measured temperature and calculated temperature, mixture fraction and air/fuel ratio based on gas analysis, flame 2.
3.3 Measurements in the axisymmetric combustor

This section describes the details of the confined flame experiments and begins with a description of the design of the combustor, subsection 3.3.1. The experimental program and the flow conditions are given in subsection 3.3.2 and the velocity results for the isothermal flow field are presented and analysed in subsection 3.3.3. Subsection 3.3.4 presents the results of the combusting flow experiments and quantifies the different influences of droplet size, degree of swirl and air inlet temperature on the flame structure and spray flame performance and ends with more general discussion of these results. The section ends with a brief summary of the main conclusions and findings.

3.3.1 Flow configuration

The combustion chamber was manufactured from a 900 mm length of mild steel pipe of 206 mm internal diameter and 5 mm thickness and is shown schematically in figure 3.13 and plate 3.1 with its axis of symmetry in the vertical plane (see figure 3.13a-b). The pipe was circular to within about 1.5 mm except in the vicinity of the observation windows, as shown in figure 3.13b, which reduced the local diameter by up to 3 mm. The combustor wall had four water cooled sections: each of the first two comprised of two sector-shaped annuli due to the observation window and the third and fourth are complete annuli, see figure 3.13a-b. Along the top side of combustion chamber, twelve 200 μm diameter chromel-alumel thermocouples, 40 mm apart, were embodied in the wall of the combustor as indicated in figure 3.13d. Along the horizontal plane of the combustor, observation ports of 35x400 mm was cut and fitted with a vitreous silica plate window of 90 mm length and steel inserts for
the remainder. As a result, axial profiles of velocity could be made over lengths of 50 mm at a time without disturbing the flow and by repositioning the window. The 90 μm diameter thermocouple and the water-cooled sampling probe described in chapter 2, were passed through the observation ports as shown in figure 3.14. The probe was screwed to a 3-D transverse which allowed translation along the radial and axial directions of the combustor with a precision of better than 0.2 mm.

The combustion chamber was equipped with a rotating cup atomizer, capable of producing sprays with near mono-disperse controlled droplet size. The generated flat sprays had two droplet-size ranges, main and satellite, which were controlled by the rotational speed and fuel flow rate: Macfarlane et al (1969) indicated that the contribution of the satellites to the mass mean-diameter of the spray is insignificant. Variation in the spray mean-droplet diameter was possible, without altering the fuel mass flow rate, by changing the rotational speed of the cup. The driving variable-speed motor of the atomizer (SPINTEC type VM10, air cooled) provided rotational speed between 11,450 and 72,000 rpm. The cooling air velocity can affect the spray angle and mean-droplet diameter. Visual observation of the spray with and without cooling air indicated that the spray angle was slightly less than 180 degrees. The detailed information of Macfarlane (1982), Macfarlane (1971), Macfarlane et al (1969) and Fraster et al (1963) for rotary cup blast atomizers and the discussions with Coats (1982) suggest that the effect of the cooling air velocity used here on the spray mean-droplet diameter is likely to be negligible. In this work, the spray mean-droplet diameter was estimated from the data given by Macfarlane et al (1969) for the same atomizer according to the rotational speed
of the motor. Due to the flatness characteristics of the sprays of this type, the atomizer was located 22 mm inside the combustion chamber as shown in figure 3.13c. Figure 3.15 and plate 3.2 show the arrangement of the equipment and the flow system. The combustion air was supplied from a compressor, at nearly atmospheric pressure, to the combustor through a changeable axial guide vane-cascade swirler assembled co-axially with the fuel atomizer as shown in figures 3.13 and 3.15; the geometric details of the three vane swirlers used throughout this study are given in Appendix A.I. The swirling of the combustion air was necessary to stabilize the flames and to allow the investigation of the swirl-degree on the structure and performance of the kerosene spray flames. The degree of swirl is usually characterized by the swirl number, \( S \), which is a non-dimensional number representing axial flux of swirl momentum of the air flow through the swirler divided by axial flux of axial momentum multiplied by a characteristic radius and can be expressed as:

\[
S = \frac{\int_{r_1}^{r_o} \rho \bar{U} \bar{W} r^2 dr}{\int_{r_1}^{r_o} \rho \bar{U} r dr} = \frac{1 - (r_o/r_1)^3}{1 - (r_o/r_1)^2} \left\{ \frac{2}{3} \right\} \tan \theta
\]  

where \( r_1 \) and \( r_o \) are the inner and outer radii of the swirler, \( \bar{U} \) and \( \bar{W} \) are the axial and tangential mean velocity components respectively and \( \theta \) is the vane angle.

Each of the vane-cascade swirlers used in this work has ten blades inclined to the main flow at 60, 45 and 30 degrees, corresponding to swirl numbers of 1.4, 0.81 and 0.47 respectively. The most significant and useful phenomenon of the flow in the wake of the swirl vanes is the recirculation zone generated centrally for
supercritical numbers, $S > 0.6$ (see for example Beer et al (1972)). The recirculation zone plays an important role in flame stabilization by providing a hot flow of recirculated combustion products and a reduced velocity region where flame speed and flow velocity can be matched.

The flow rate of the combustion air was measured with an orifice plate constructed in accordance with B.S.1402. A voltage-controlled 22.5 kW electric heater was installed in the combustion air line after the orifice plate, as shown in figure 3.14, to preheat the combustion air. The combustion air line between the heater and the combustor entrance was thermally insulated. The combustion air temperature was measured immediately before the swirler using a chrome-alumel thermocouple.

The atomizer cooling air was supplied from laboratory compressed air and its flow was measured with a rotameter. The kerosene fuel was pumped to the atomizer and the flow rate also was measured with a rotameter.

3.3.2 The experimental program and operating conditions

The motivation and objectives of this work are to investigate the influences of the different parameters which affect the mixing, evaporation and combustion in the spray flames including droplet size and combustion air swirl and inlet temperature. An additional purpose is to provide detailed information to help the development and evaluation of computational methods for the axisymmetric combustors so as to allow the calculation of gas-turbine flow properties starting from a knowledge of the spray
characteristics, mass flow rates of fuel and air and the boundary conditions. To achieve these purposes, detailed measurements of mean temperature, volume concentrations of carbon monoxide, carbon dioxide, oxygen and unburned hydrocarbons, axial and tangential droplet velocities and wall temperatures were obtained for the operating conditions summarised in table 3.3. Starting 25 mm downstream of the swirler, gas concentration and temperature data were collected at ten axial positions spaced at 25 mm increments. Radial probe positions started from the combustor axis and were spaced at 10 mm increments. The droplet velocities were measured where the spray existed. The following paragraph summarizes the arrangements of the different flames to satisfy the above objectives.

At the beginning the isothermal flow velocities (without spray) in the combustor corresponding to the three degrees of swirl used, 1.4, 0.81 and 0.47, were obtained to determine the aerodynamic properties of the flow field and for reference purposes. The operating conditions for these measurements correspond to runs 1, 2 and 3 of table 3.3. In the three flames corresponding to runs 4, 5 and 6 of table 3.3, the spray mean-droplet diameters were 54, 77 and 115 μm and a 60 degrees combustion air swirler angle was used. The combustion air swirler angle was then changed to 45 degrees and 30 degrees and the two sets of flames corresponding to runs 7 to 9 and 10 to 12 were obtained: these flames with the set 4 - 6 allowed the study of the influence of the spray mean-droplet diameter on the combustion characteristics and the flame structure at three levels of turbulence. Flames 4, 7 and 10, 5, 8 and 11 and 6, 9 and 12 provided the data concerning the effect of combustion air degree of swirl on the interaction between the fuel spray and the combustion air and the resulting flame properties for 54, 77, 115 μm spray mean-droplet
### TABLE (3.3) OPERATING CONDITIONS OF THE CONFINED FLAME EXPERIMENTS.

<table>
<thead>
<tr>
<th>Run No.</th>
<th>$S$</th>
<th>Combustion air rate (Kg/h)</th>
<th>Combustion air inlet temp. ($^\circ$C)</th>
<th>Atomizer cooling air rate (Kg/h)</th>
<th>Kerosene flow rate (Kg/h)</th>
<th>Atomizer rpm</th>
<th>Droplet mean-diam. (μm)</th>
<th>Measurements</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.4</td>
<td>266.0</td>
<td>-</td>
<td>12.8</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>$\bar{u}, \bar{W} &amp; \bar{u}$</td>
</tr>
<tr>
<td>2</td>
<td>0.81</td>
<td>266.0</td>
<td>-</td>
<td>12.8</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>$\bar{u} &amp; \bar{u}$</td>
</tr>
<tr>
<td>3</td>
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<td>266.0</td>
<td>-</td>
<td>12.8</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>$\bar{u} &amp; \bar{u}$</td>
</tr>
<tr>
<td>4</td>
<td>1.4</td>
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<td>20</td>
<td>12.8</td>
<td>8.66</td>
<td>26,240</td>
<td>54</td>
<td>$T, CO_2, CO, O_2, UHC &amp; T_w$</td>
</tr>
<tr>
<td>5</td>
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<td>266.0</td>
<td>20</td>
<td>12.8</td>
<td>8.66</td>
<td>17,130</td>
<td>77</td>
<td>$T, CO_2, CO, O_2, UHC &amp; T_w$</td>
</tr>
<tr>
<td>6</td>
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<td>20</td>
<td>12.8</td>
<td>8.66</td>
<td>11,450</td>
<td>115</td>
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<td>26,240</td>
<td>54</td>
<td>$T, CO_2, CO, O_2, UHC &amp; T_w$</td>
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<tr>
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<td>266.0</td>
<td>20</td>
<td>12.8</td>
<td>8.66</td>
<td>17,130</td>
<td>77</td>
<td>$T$</td>
</tr>
<tr>
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<td>266.0</td>
<td>20</td>
<td>12.8</td>
<td>8.66</td>
<td>11,450</td>
<td>115</td>
<td>$T$</td>
</tr>
<tr>
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<td>12.8</td>
<td>8.66</td>
<td>26,240</td>
<td>54</td>
<td>$T, CO_2, CO, O_2, UHC &amp; T_w$</td>
</tr>
<tr>
<td>11</td>
<td>0.47</td>
<td>266.0</td>
<td>20</td>
<td>12.8</td>
<td>8.66</td>
<td>17,130</td>
<td>77</td>
<td>$T$</td>
</tr>
<tr>
<td>12</td>
<td>0.47</td>
<td>266.0</td>
<td>20</td>
<td>12.8</td>
<td>8.66</td>
<td>11,450</td>
<td>115</td>
<td>$T$</td>
</tr>
<tr>
<td>Run No.</td>
<td>S</td>
<td>Combustion air rate (Kg/h)</td>
<td>Combustion air inlet temp. (°C)</td>
<td>Atomizer cooling air rate (Kg/h)</td>
<td>Kerosene flow rate (Kg/h)</td>
<td>Atomizer rpm</td>
<td>Droplet mean-diam. (µm)</td>
<td>Measurements</td>
</tr>
<tr>
<td>---------</td>
<td>-----</td>
<td>-----------------------------</td>
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<td>-------------------------</td>
<td>----------------------------------</td>
</tr>
<tr>
<td>13</td>
<td>1.4</td>
<td>266.0</td>
<td>125</td>
<td>12.8</td>
<td>8.66</td>
<td>26,240</td>
<td>54</td>
<td>T, CO₂, CO, O₂, UHC,</td>
</tr>
<tr>
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<td>266.0</td>
<td>225</td>
<td>12.8</td>
<td>8.66</td>
<td>26,240</td>
<td>54</td>
<td>ū, w̄, u, w &amp; T</td>
</tr>
<tr>
<td>15</td>
<td>1.4</td>
<td>266.0</td>
<td>125</td>
<td>12.8</td>
<td>8.66</td>
<td>11,450</td>
<td>115</td>
<td>T, CO₂, CO, O₂,</td>
</tr>
<tr>
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<td>1.4</td>
<td>266.0</td>
<td>225</td>
<td>12.8</td>
<td>8.66</td>
<td>11,450</td>
<td>115</td>
<td>UHC &amp; T_w</td>
</tr>
</tbody>
</table>

T : Mean gas temperature.
CO₂ : Mean carbon dioxide concentration.
CO : Mean carbon monoxide concentration.
O₂ : Mean oxygen concentration.
UHC : Mean unburned hydrocarbon concentration.
T_w : Wall-temperature.

\[ \bar{U} \text{ and } \bar{W} \] : Mean axial and tangential velocity of the isothermal flow.
\[ \tilde{u} \] : rms of axial velocity fluctuation.
\[ \bar{U}_P \text{ and } \bar{W}_P \] : Mean axial and tangential velocity of the droplets.
\[ \tilde{u}_P \text{ and } \tilde{W}_P \] : rms of axial and tangential velocity fluctuation of the droplets.
diameter respectively. The combustion air was preheated, for the 60 degree swirl case, to 125 and 225 °C for the two flames having spray mean-droplet diameter of 54 and 115 μm and corresponding to runs 13 through 16. Flames 4, 13 and 14 and 6, 15 and 16 permitted the study of the influences of the inlet temperature of the combustion air on the spray characteristics and the flame properties.

3.3.3 Velocity characteristics of the isothermal flow in the combustor

Swirl of the combustion air stream results in much of desirable features including enhanced flame stability and combustion intensity and improves the turbulent mixing throughout the combustor. Therefore, the axial mean velocity and the corresponding rms of velocity fluctuations for swirl numbers of 1.4, 0.81 and 0.47 were obtained for the same mass flow rates of the atomizer cooling air and combustion air, but without injection of the fuel, to understand the aerodynamics of the flow. Measurements were not possible close to the swirler exit due to the projection of the atomizer which caused reflection of the laser beams.

The radial profiles of the mean axial velocity and the corresponding normal stress for the highest swirl number case of run 1 (S = 1.4) are shown in figure 3.16 and 3.17. It is clear from the radial profile distributions that the recirculation zone is off the centreline, at the first station of measurements (X = 25 mm), and extends from R = 18 mm to R = 51 mm. Further downstream, the zero velocity moves radially outwards and the recirculation zone covers a significant portion of the enclosure radius reaching its maximum extent, R = 24 to 74 mm, at X = 75 mm. The maximum negative mean axial velocities are around -4 m/s at X = 50 mm. Further downstream,
the recirculation zone diminishes gradually to \( X = 150 \) mm beyond which it will disappear. The maximum positive velocity shown off the centreline has moved to larger radii with a reduction in its value at downstream stations. In the centreline region, there are relatively high positive velocities (see radial profiles at \( X = 50 \) and \( 75 \) mm) and this is attributed to the injection of the atomizer cooling air from an annulus around the atomizer with an induced angle of 13 degrees at \( X = 17 \) mm.

The radial profiles of the rms of velocity fluctuation are presented in figure 3.17 for the same axial locations as the mean velocity. They indicate that the maxima of the normal stresses profiles at upstream sections are close to the maximum gradients of the axial mean velocity. The generation of \( \overline{u^2} \) is connected to the shear stress \( \overline{uv} \) and the rate of strain \( (\partial \overline{U}/\partial y) \) and can be expressed by:

\[
\frac{G_{\overline{u^2}}}{u^2} = -2\rho \overline{u^2} \left( \frac{\partial \overline{U}}{\partial x} \right) + \overline{uv} \left( \frac{\partial \overline{U}}{\partial y} \right)
\]

(3.8)

It is expected, therefore, that the location of the maximum normal stress will be related to the maximum rate of strain, \( (\partial \overline{U}/\partial y) \), in the region of high shear. Moving to the downstream stations, the two peaks of the normal stress profiles diffuse, one to the centreline and the other outwards, and decay gradually. At \( X = 150 \) mm, the normal stress profile is nearly uniform and, as indicated by Habib and Whitelaw (1980), this suggests strong turbulent diffusion.

Figures 3.18 and 3.19 represent the mean velocities and their corresponding normal stresses for the lower swirl number of 0.81 (run 2). The development of the mean axial velocity across the
centreline, figure 3.18, indicates that an off-centreline recirculation zone is again created with a significant reduction in the reversed velocities and width and starts at larger radii; the maximum negative velocity is 2.5 m/s and, for example at the first station of measurements, it extends from \( R = 23 \) to 37 mm. The observed reduction in size and strength of the recirculation zone are caused by the reduction in the angular to axial momentum ratio induced by the decrease of the swirl number. With the reduction in the size of the recirculation zone, the off-centreline maximum mean velocity diffused to the centreline with little increase in magnitude. Very far downstream, the recirculation zone decays and tends to disappear. The reduced spreading of the incoming air stream, again induced by the reduction of the swirl number, has again resulted in negative velocities near to the combustor wall at the first station of measurements. At the following station \( (X = 50 \text{ mm}) \), the small wall-recirculation zone has disappeared. As described earlier, the maximum normal stress of figure 3.19 is associated with the maximum velocity gradient, \((\partial \bar{U}/\partial y)\), as can be seen by the radial profiles at \( X = 25, 50 \) and 75 mm. This link breaks down at the downstream stations, \( X = 100 \) and 125 mm, where the normal stress has one peak value at the centreline. It is likely that the maximum on the centreline can be attributed to the axial rate of strain, \((\partial \bar{U}/\partial x)\), since the rate of change of \( \bar{U} \) with \( x \) is large as shown in figure 3.18.

The development of the mean axial velocity and the corresponding rms of velocity fluctuations for the lowest degree of swirl of run 3 \((S = 0.47)\) are shown in figures 3.20 and 3.21 respectively. It is very clear that the off-centreline recirculation zone has disappeared completely in the upstream region and a weak
wall-recirculation zone has developed. This is expected since the swirl angle of 30 degrees implies a small angular momentum and consequent low spreading of the combustion air as shown by the maximum mean axial velocity at \( R = 30 \text{ mm} \) at \( X = 25 \text{ mm} \). The reduction of the swirl number, also, results in a small and weak recirculation zone at the downstream stations, i.e. \( X = 100 \) and 125 mm. Due to the near axial injection of the combustion air from the swirler, positive axial velocity values are more peculiar than with the 60 and 45 degrees swirlers. The maximum turbulence velocities of figure 3.21 confirm the high rate of strain, \( \frac{\partial U}{\partial y} \), as discussed previously.

The radial profiles of figures 3.16, 3.18 and 3.20 are interpolated to form the contours of the mean axial velocity corresponding to the three swirling flows which are shown in figure 3.22 and indicate plainly the features of the recirculation zones which are important for the combustion flows of the next subsection. It is very clear, see figure 3.22, that the high swirl number cases (\( S = 1.4 \) and 0.81) are associated with an off-centreline recirculation zone which reduces in size, shape and strength as the swirl number decreases. The decrease of the swirl number results in lower radial spreading of the combustion air, relatively slower decay of the maximum velocity along the shear-layer and a small wall-recirculation zone in the region of \( X < 45 \text{ mm} \). On the other hand, the reduction of the swirl number to 0.47 is accompanied by movement of the off-centreline recirculation zone to a weak and small central one at downstream and a relatively larger wall-recirculation region with consequent movement of the reattachment point up to \( X = 100 \text{ mm} \). The contour maps of the three cases indicate that the region near to the centreline, and in the vicinity of the atomizer, is dominated by higher velocities of around 10 m/s due to the high injection velocity
of the atomizer cooling air. These high velocities decay earlier and extend to larger radii as the swirl number decreases, which can be attributed to the reduced spreading of the combustion air.

3.3.4 Results for combusting flows

It is instructive to consider first the basic flame structures relevant to the range of flames studied and subsequently the experimental data, including mean gas temperature and species concentration of CO₂, CO, O₂ and UHC, which describe them. Visual observation and the detailed results for the different spray flames suggest the four flame structure categories of figure 3.23 to represent the different operating conditions. The experimental data also show that the important parameters which control the flame structure are the size and position of the recirculation zone coupled with the spray mean-droplet diameter.

Figure 3.23A corresponds to the high swirl number of 1.4 and small mean-droplet diameter of 54 μm with consequently large recirculation residence time and short evaporation time so that a high percentage of the fuel droplets evaporate within the recirculation zone. The penetration of the fuel droplets from the recirculation boundary is negligible and the flame is mixing controlled, i.e. the rate of mixing of the fuel vapour with air, rather than the evaporation rate of liquid fuel, determines the heat release rate. Evaporated fuel flows towards the combustion air exit and feeds the shear-layer flame which lies along the outer boundary of the recirculation zone. The shear-layer corresponds to the region of the flame which is hollow and surrounds the recirculation zone. The highly turbulent mixing within the shear-layer and the comparatively high temperature of the recirculated gases intensify
the chemical reaction and combustion intensity. The enhancement of the combustion throughout this region increases the gases temperature and droplet evaporation with consequent convection of fuel vapour downstream and development of a main-flame region.

As the spray mean-droplet diameter increases, which is associated with an increase in the evaporation time, the droplets penetrate the recirculation zone and cross the shear-layer with alteration of the flame structure to that of figure 3.23B. Here, there is a free-stream flame near to the combustor wall has been developed which stems from the evaporation and combustion of the fuel droplets penetrating across the shear-layer.

The reduction of the combustion air swirl number to 0.81 increases the possibility that the smaller droplets ($D_m = 54 \mu m$) penetrate the outer recirculation boundary because it is nearer to the atomizer, see figure 3.22(ii), and the basic flame structure of figure 3.23C will be common to the different droplet sizes at this swirl number. The experimental data indicates that a large number of the droplets which penetrated the recirculation region near the fuel atomizer were swept back to the wall-recirculation zone or to the main-flame region downstream due to the higher gradient of the gas velocities.

The fourth case, which corresponds to a weak swirl ($S = 0.47$) is characterized by figure 3.23D. The flame is stabilized close to the wall-recirculation zone and the main-flame region extends downstream of the shear-layer flame and near to the combustor wall.

This subsection presents the combusting flow measurements including the mean gas temperature, species concentration, droplet velocities and the wall-temperature distribution for the range of the
flames given in table 3.3. The detailed data will be presented in the form of contour maps, since they show more clearly the changes in the flame properties; the corresponding radial profiles at different locations along the combustor are given for reference purposes in Appendix A.II. The influence of the spray mean-droplet diameter on the combustion behaviour is quantified in the following subsection. The effect of the combustion air swirl on the flame pattern and properties is described in subsection 3.3.4.2. The influence of the combustion air inlet-temperature on the spray characteristics and combustion performance is given in subsection 3.3.4.3. Finally, a general discussion of the results is presented.

3.3.4.1 Influence of spray mean-droplet diameter

This subsection presents the results of flames 4 to 12 which show the effect of the spray mean-droplet diameter on the combustion characteristics and flame properties for three values of air swirl. In the first part, (A), the results obtained at the highest degree of swirl ($S = 1.4$) are presented and the next two parts, (B) and (C), describe the results obtained with the 0.81 and 0.47 swirl numbers respectively.

(A) Results of 1.4 swirl number flames

Contours of mean gas temperature and species concentrations of CO$_2$, CO, O$_2$ and UHC are shown in figures 3.24 to 3.28 respectively and correspond to flames 4, 5 and 6. Flames 5 and 6 have mean-droplet diameters of 77 and 115 μm respectively and are similar to each other and can be represented by the flame structure of figure 3.23B. On the other hand, the results of the smallest spray mean-droplet diameter
flame, run 4, indicate a different trend which is similar to that of figure 3.23A. Visual observation of the three flames indicated that a relatively large amount of soot was generated with the bigger initial droplet size flames (runs 5 and 6) and was evidenced by the bright yellow colour of the flames and the high accumulation rate of carbon particles on the casing of the fuel atomizer. The temperature contours of figure 3.24 show that the shear-layer flame region, \( X = 15 - 100 \) mm and \( R = 50 - 70 \) mm, is characterized by high temperatures of \( 1300 \) to \( 1600 \) °C and steep gradients with corresponding occurrence of high-intensity combustion. This is accompanied by \( \text{CO}_2 \) concentrations of \( 8 - 10\% \) and destruction of \( \text{CO} \) concentration as indicated in figures 3.25 and 3.26 respectively. The oxygen concentrations of figure 3.27 show that the shear-layer flame region is associated with negligible \( \text{O}_2 \) values for the smallest droplet diameter flame, run 4, and small values of \( 1 - 5\% \) as the mean-droplet diameter increases. Common to the three flames, the unburned hydrocarbons is around \( 1.5\% \) throughout the shear-layer zone, see figure 3.28, which is expected due to the nearly radial injection of the fuel spray within the upstream of the combustor. Close inspection of the shear-layer flame zone indicates rapid destruction of \( \text{CO} \) concentration associated with larger mean-droplet diameter flames (runs 5 and 6), as shown in figure 3.26(B) and (C), and consequent achievement of the maximum mean temperature (\( \approx 1600 \) °C) and \( \text{CO}_2 \) concentration (12%) in the upstream part of the shear-layer. On the other hand, the smallest mean-droplet diameter (run 4) shows slower rate of \( \text{CO} \) oxidation, relatively lower temperatures (\( T_{\text{max}} = 1500 \) °C) and delay of \( \text{CO}_2 \) maximum occurrence. The previous observations are likely because the oxidation rate of \( \text{CO} \) exceeds the formation and vaporization rates within the shear-layer as a result
of slower evaporation rates associated with the larger mean-droplet diameter. The slower vaporization rates of the big droplets flames are evidenced by hydrocarbons measured at many points in the upstream of the combustion chamber and especially very near to the wall for the flame of run 6, see figure 3.28(C).

The upstream part of the recirculation zone of \( X < 90 \text{ mm} \) and \( R = 20 - 50 \text{ mm} \) is dominated by less steep temperature gradients than those observed in the shear-layer flame region and the highest values of unburned hydrocarbons. In particular, the area in the vicinity of the fuel atomizer, \( X = 30 \text{ mm} \) and \( R = 25 \text{ mm} \), shows high values of hydrocarbon up to 5% with a small amount of evaporated fuel as evidenced by the liquid kerosene sucked through the sampling probe. The recirculated hot gases aid the evaporation of the fuel droplets, accelerate the combustion rate and sweep the fuel vapour and probably small droplets towards the combustion air exit to feed the shear-layer zone which stabilizes the flame. The CO\(_2\) and CO contours of figure 3.25 and 3.26 reveal that a significant amount of the fuel vapour is oxidized to CO and CO\(_2\) throughout the recirculation zone. These formations of CO and CO\(_2\) are consistent with the high gas temperature recorded in the range 1300 - 1600 °C and the decrease of oxygen concentration throughout the recirculation zone as shown in figures 3.24 and 3.27 respectively. The temperature contours for the three flames indicate that the transition from large to small droplet diameter is associated with an increase in the measured values at locations close to the fuel spray exit. A similar increase in CO values can be seen in figure 3.26 while the contours of CO\(_2\) shown in figure 3.25 indicate a slight reduction in the measured values with the decrease in the mean-droplet diameter. The trend of these changes in temperature and species concentration, as
expected, implies an increase in the combustion intensity with the reduction of the spray mean-droplet diameter. This can be attributed to the increase in the evaporation rate associated with small droplets and comparatively higher temperature of the recirculated gases from downstream of the flame, which enhance the reaction rate of the fuel-air mixture. The increased evaporation rate shifts the fuel-air mixture towards the fuel rich limit and consequently a significant amount of the hydrocarbons is partially oxidized to CO.

In the region near to the atomizer cooling air exit, \( R < 20 \text{ mm} \), figure 3.24 shows that low temperature values (\(< 500 \degree \text{C}\)) are associated with the large droplet flames, especially run 6, which is likely due to the influence of the atomizer cooling air and the reduction of the droplet number moved with it as the droplet size increases. The influence of the atomizer cooling air is evidenced by the curvature of the temperature and species concentrations contours and the relatively lower temperature, \( \text{CO}_2 \) and \( \text{CO} \) and higher \( \text{O}_2 \) values throughout this region as shown in figures 3.24 to 3.27 respectively. On the other hand, the effect of the cooling air is negligible with the small droplet diameter flame, run 4, as indicated by the very high gas temperature of \( 1600 \degree \text{C} \), high \( \text{CO}_2 \) and \( \text{CO} \) and negligible \( \text{O}_2 \) concentration values. This is due to the high evaporation rate of the small droplets and consequent good mixing between the fuel vapour and oxidant at the prevailing high temperatures and enhanced chemical reaction.

The fuel vapour and hydrocarbons that do not react within the recirculation zone or cross the shear-layer flame regions, continue to move downstream, where further mixing with the combustion air and chemical reaction occur in the main-flame and free-stream.
regions. The temperature contours of figure 3.24 show that the main-flame region covers a large area of downstream combustion chamber and encompasses the centreline. The CO\textsubscript{2} contours of figure 3.25 are consistent with the measured temperatures throughout this region and indicate high values of 9 - 12\% with a tendency to decrease at larger radii; in contrast, the O\textsubscript{2} concentration values increase outwards. The CO and UHC contours shown in figures 3.26 and 3.28 respectively indicate values of less than 0.5\% in the whole region except for run 4 which shows some CO near to the combustor centreline.

The temperature and concentration contours for the three flames of figures 3.24 through 3.28 indicate that the increase in the mean-droplet diameter is accompanied with a shift of the main-flame region towards small radii of the combustion chamber with relatively lower temperature values. These lower temperatures are associated with higher O\textsubscript{2} concentration, 7 - 9\%, relatively lower CO\textsubscript{2} and negligible CO values which imply reduction in the combustion intensity. This is due to the slow evaporation rate and penetration of droplets to the free-stream associated with the large droplets which reduce the amount of fuel vapour convected to the main-flame region and, therefore, the dilution effects of surrounding air stream increases. High temperatures of around 1500 °C are measured within a small upstream region of the main-flame of runs 5 and 6, X < 150 mm and R = 20 - 40 mm, and is due to the influence of the recirculation zone flame. The concentration of CO\textsubscript{2} is comparatively high, 9 - 11\%, O\textsubscript{2} is relatively low, 5 - 7\%, and CO and UHC disappear quickly as the oxidation, formation and dilution rates exceed the transportation of fuel vapour from the spray region. In the downstream region of the main-flame and at small radii, it is clear from figure 3.26 that the destruction rate of CO is slow and a nearly constant value exists in
case of the smallest mean-droplet diameter flame ($D_m = 54 \, \mu m$): it is accompanied by high values of $CO_2$, 12%, negligible $O_2$ concentrations and temperatures of 1500 $^\circ C$ and more. These observations are probably due to the initial higher CO formation rate, convection of most of the fuel vapour towards the main-flame region and the lack of oxygen.

Along the outer boundaries of the shear-layer and main-flame regions of $R$ around 65 mm, where the droplets penetrate the shear-layer to the free-stream and are swept downstream by high velocity of gases, the chemical reaction proceeds in this region and in particular for the flames 5 and 6 which are associated with significant penetration effects. In general, the good turbulent mixing of the fuel vapour and the surrounding combustion air allows chemical reaction to proceed with a high intensity along the outer boundary of the shear-layer for the three flames which can be interpreted from the steep gradient of the temperature and $CO_2$ concentration as shown in figures 3.24 and 3.25 respectively. Further downstream and along the main-flame boundaries, the flame of run 4 shows lower temperatures of 1100 $^\circ C$ which decrease towards the combustor wall to 800 $^\circ C$ due to the dilution effects and the reduced fuel convected to the free-stream zone. The temperature contours along the boundaries of the main-flame of runs 5 and 6 show high values, even close to the combustor wall; this implies that the comparatively higher number of the fuel droplets in the free-stream continue to move with the combustion gases and enhance the combustion intensity in the free-stream zone. Consistent with that, the $CO_2$ contours of figure 3.25 indicate that the increase of the droplet size is associated with relatively higher gradients near to the combustor wall.
The wall-temperature distributions corresponding to the flames of runs 4, 5 and 6 are shown in figure 3.29. The general trends of the three flames are similar and show that the combustor wall has temperatures of less than 125 °C as a result of the large amount of heat transferred to the cooling water jacket as indicated in table 3.4. Within the initial part of the combustor, the wall-temperature increases rapidly to a maximum value which ranges between 80 and 102 °C and then decreases with the downstream distance. The observed rise in the wall-temperature within the upstream part of the combustor is mainly due to the high combustion intensity on both shear-layer and the main-flame regions as indicated previously. It is clear, however, that the wall-temperature attains higher values with the increase of the mean-droplet diameter of the spray and these are directly related to the increased gas temperatures near to the combustion chamber wall and, in particular, in the free-stream zone. This is in agreement with the previously observed intense combustion close to the combustor wall as the droplet size increases and results in more fuel being swept to the free-stream zone. This again confirms that, for large droplets and due to consequent penetration of the fuel droplets to the free-stream, a significant amount of the fuel participates in the combustion process with a corresponding increase of the chemical reaction in the free-stream flame zone. The high values of the heat release to the wall in the initial part of the combustor as indicated in table 3.4 (53.7, 70.5 and 68.6 KW/m² for runs 4, 5 and 6 respectively) are consistent with the observed higher wall temperatures at this part. Further downstream, the wall-temperature continues to decrease and achieves a nearly constant value which is due to the expected reduction in the combustion intensity.
TABLE (3.4) HEAT ABSORBED BY THE DIFFERENT SECTIONS OF THE COOLING JACKET

<table>
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<th>Run No.</th>
<th>1st section X= 0-220 mm</th>
<th>2nd section X= 220-420 mm</th>
<th>3rd section X= 420-620 mm</th>
<th>4th section X= 620-900 mm</th>
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</tr>
<tr>
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<td>61.09</td>
<td>53.50</td>
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</table>
(B) Results of 0.81 swirl number flames

The temperature contours corresponding to the mean-droplet diameters of 54, 77 and 115 μm at 0.81 combustion air swirl number are shown in figure 3.30. The general features of the three flames are similar and consistent with the physical description of the basic flame structure of figure 3.23c. The shear-layer flame region, X = 25 - 100 mm and R = 45 - 65 mm, is characterized by a steep temperature gradient and temperature values of 1200 - 1600 °C. Figure 3.30 shows that the increase in the mean-droplet diameter is associated with a reduction in the maximum temperature achieved within the shear-layer flame and consequent reduced combustion intensity due to the reduced evaporation rate associated with the big droplets. The curvature of the temperature contours in the region surrounding the shear-layer flame region, R = 70 mm, suggests that the relatively high velocity incoming air stream influences the flame in this region and this effect increases as the droplet diameter increases. It can be observed that although the shear-layer flame at the wall-recirculation boundary is not well defined, relatively high temperatures occur at points coincident with the low velocity region near to the wall, with a tendency to increase as the droplet size increases. The former is probably due to the high turbulent mixing rate in this region while the latter stems from the increased number of fuel droplets within this region as result of stepped up penetration of the droplets across the main-recirculation zone and the shear-layer as the initial mean-droplet diameter of the spray increases.

The main-recirculation zone and the region surrounding the combustor centreline, X < 120 mm and R < 45 mm, are dominated by high
temperature values with the implication of high combustion intensity in these regions. However, in the near centreline region, the shifting of the inner recirculation boundary to larger radii, see figure 3.22(ii), allowed some of the fuel droplets, especially small diameter droplets, to move with the high velocity atomizer cooling air. These droplets coupled with the high temperature and air prevailing at this region enhance the combustion intensity as shown by the high temperature values of 1500 - 1600 °C at most of the region for the three flames, see figure 3.30. The curvature of the temperature contours in the vicinity of the atomizer associated with the large droplet diameter flames, in particular run 9, is due to the delayed evaporation of big droplets and increased influence of the atomizer cooling air.

The fuel which does not burn in the upstream part of the combustor is convected to the main-flame region which, as shown in figure 3.30, covers most of the downstream region and is dominated by high temperatures. It can also be seen that the increase of the droplet diameter is accompanied with temperature reduction of 100 °C throughout this region. This can be attributed to the reduced amount of fuel moved downstream as a result of the increased fuel droplets which have been burnt in the upstream low velocity zone near to the wall. Common to the three flames and along the boundary of the main-flame region, the temperature decays towards the combustor wall as shown in figure 3.30 and results from the rapid reduction of the gas velocity towards small radii of the combustor.
(C) Results of \(0.47\) swirl number flames

Figure 3.31 shows the temperature contours for runs 10, 11 and 12 at swirl number of \(0.47\). The general features of the three flames are again similar and in agreement with the basic flame structure of figure 3.23D. The smallest mean-droplet diameter \((D_m = 54 \, \mu m)\) has steeper temperature gradients within the shear-layer and achieves a shorter shear-layer flame length. The temperature peaks within the shear-layer region at \(R \approx 60 \, mm\), see figure 3.31, extend upstream to axial distances of around 60, 42 and 38 mm for the flames 10, 11 and 12 respectively. The observed slower temperature rise and increased length of the shear-layer flame associated with the large droplet diameter flames can stem from the influences of the increased number of droplets entrained in the wall-recirculation zone due to their large life time while, in case of run 10, this effect is small as the rate of evaporation is relatively higher and the fuel vapours follow the incoming combustion air stream. The wall-recirculation region indicates low temperature values for the three flames in addition to a thick layer of carbon deposits on the combustor wall which implies a low combustion intensity within this part of the flame. This is due to the slow evaporation rates, low turbulent mixing and small radial spreading of the combustion air.

The main-flame region, which prevails near to the wall and downstream of the shear-layer zone \((X > 110 \, mm\) and \(R > 55 \, mm)\), reveals that the chemical reaction rates are influenced by the slow evaporation rate and the reduced combustion intensity throughout the shear-layer as the droplet size increases.

The central part of the flames is dominated by lower temperature values, especially in the vicinity of the atomizer, with
a tendency to increase at downstream distances. The temperature contours of figure 3.31 at $X < 100$ mm imply very low combustion intensity as a result of the very low evaporation rate and high dilution and quenching effects: the former is consistent with the low temperatures prevailing in this region and the latter is caused by the high velocity air stream surrounding the atomizer. The reversed hot gases within the central-recirculation zone, which accelerate the evaporation of the droplets, and the improved turbulent mixing within this zone increase the reaction rates. These influences can be recognized from the fast rise of the temperature values within this region for flame 10, see figure 3.31(A), where the evaporation rates are higher.

3.3.4.2 Influence of combustion air swirl

This subsection compares in detail the results of mean temperature, species concentration of $CO_2$, CO and $O_2$ and wall-temperature distributions corresponding to runs 4, 7 and 10. The study of these results will show the extent to which the combustion air swirl influences the interaction between the fuel spray and combustion air and performance of the kerosene spray flames. Figures 3.32 to 3.35 show the results of the temperature and species concentrations obtained at the swirl numbers of 1.4, 0.81 and 0.47 and with the same spray mean-droplet diameter of 54 $\mu$m. The results obtained with the 1.4 swirl number have been described in detail in the previous subsection and are used here for comparison.

It is seen from figures 3.32(B) to 3.35(B) that the shear-layer flame region, $X \approx 25 - 100$ mm and $R \approx 45 - 65$ mm, is dominated by higher temperatures and steep temperature and species
concentration gradients compared to the results of run 4. The contours of CO₂ concentration show a rapid increase from 4 to 13% which is accompanied by a large reduction of O₂ concentration from 18 to 3%, as shown in figures 3.33(B) and 3.35(B) respectively. The CO concentrations have correspondingly small values, 1 - 4%, within the shear-layer flame zone due to the high formation and oxidation rates. The reduction of the swirl number from 1.4 to 0.81 has resulted in a delay and shifting of the shear-layer zone towards small radii as can be detected from the temperature and concentration contours. This result is in agreement with the reduction in the radial spreading of the combustion air stream associated with the lower degrees of vane swirler which can be deduced from the isothermal contours of the mean axial velocity shown in figure 3.22. The reduction in the swirl number also results in a quenching of the chemical reaction along the boundary of the shear-layer flame as indicated by the curvature of the temperature and concentration lines at this region.

The high temperature and concentration values of CO₂ (around 9%) and CO (around 6%) shown in figures 3.32(B) to 3.34(B) respectively within the near-wall region, X < 55 mm and R > 75 mm, indicate a high combustion intensity throughout this zone which is evidenced by small O₂ concentrations. The high intensity of combustion at this zone is due to the increased number of droplets penetrating the shear-layer coupled with the high temperature and low velocities close to the wall-recirculation zone.

In the main recirculation zone and the central region of the flame, 30 < X < 120 mm and R < 45 mm, the temperature and CO₂ concentration values are higher than those of run 4. On the other hand, O₂ and CO concentrations have values less than 3%. The
contraction of the recirculation zone, as shown previously, allows higher penetration of the droplets and movement of small droplets with the atomizer cooling air and results in lower recirculated fuel flow with the additional consequence that the rate of oxidation is increased. The relatively higher \( \text{O}_2 \) values shown by run 7 (figure 3.35B), compared to run 4 (figure 3.35A), are likely due to the influence of the lower radial spread of the combustion air.

The general outlines of the main-flame region, \( X > 110 \text{ mm} \) and \( R < 60 \text{ mm} \), is detectable from the temperature and species concentration contours of figures 3.32(B) to 3.35(B). The temperature and \( \text{CO}_2 \) concentration have high values throughout this region and small \( \text{O}_2 \) values can also be observed. It is clear, however, that the temperature values within the central part of the main-flame region, \( R < 25 \text{ mm} \), are lower than those of run 4 with associated lower \( \text{CO}_2 \) and negligible \( \text{CO} \) values. It can, also, be observed that the rates of decay of temperature and \( \text{CO}_2 \) are slower in the radial direction and faster in the axial direction while \( \text{O}_2 \) has the same trends but increases. These observations imply that the decrease in the combustion air swirl causes a decrease in the flame length and intensity of the combustion and in particular at small radii in the main-flame zone. This is mainly due to the lower turbulent mixing rates and convected fuel to the main-flame region, as detected from negligible \( \text{CO} \) within this region, compared to significant values of run 4, associated with the smaller air swirl. Along the boundary of the main-flame region of run 7, \( R > 60 \text{ mm} \), the temperature contours indicate differences up to 150 °C more than those observed for run 4 while the \( \text{CO}_2 \) and \( \text{O}_2 \) show a slight increment and reduction respectively. This is consistent with the richer fuel-air mixture in the free-stream region resulting from the increased fuel droplet
penetration.

For the swirl number of 0.47, corresponding to a weakly swirling flow, the temperature and species concentration maps of figures 3.32(C) to 3.35(C) show that the extent to which the flame structure has been altered as a result of the interaction between the fuel spray and the surrounding combustion air. Following the movement of the main-recirculation zone near the wall, $X < 100$ mm and $R > 70$ mm, the stabilizing shear-layer flame region is shifted to coincide with the boundaries of the recirculation zone where steep gradients and high temperature values are indicated. The $CO_2$ and CO concentrations show values of 8% while $O_2$ indicates small values of around 2%. The relative maxima shown in the contour maps at further downstream locations, $X > 100$ mm and $R$ around 90 mm, are indicative of the existence of the main-flame region which is, again, shifted towards the near-wall region. Inspection of the temperature and concentration levels of $CO_2$, CO and $O_2$ throughout the main-flame region ($T = 1400$ °C, $CO_2 = 10\%$, CO $= 4\%$ and $O_2 = 4\%$) indicates that a comparatively large amount of fuel has been convected to it from the spray zone and can be attributed to the entrainment of a large number of fuel droplets by the air stream surrounding the atomizer and the decay of the gas velocity towards larger radii of the combustor. The gas temperatures and $CO_2$ concentrations reveal that the spreading of the main-flame region is much lower than that observed for the high swirl flames of runs 4 and 7. This result appears to be consistent with the anticipated reduction in the turbulent mixing rates for this low swirl flame. The consequent reduction in the flame length is evidenced by the gradual decrease in the measured temperature, CO concentration and increase of $O_2$ within the main-flame region which
starts at X of around 160 mm.

Within the central part of the flame, R < 45 mm, it is clear from the contour maps that the temperature, CO$_2$ and CO concentration values are generally much lower than those observed for runs 4 and 7, whereas O$_2$ concentrations are much higher. At small axial distances and in the vicinity of the atomizer the chemical reaction is adversely influenced by the dilution and quenching effects of the high velocity air stream surrounding the atomizer and by the low droplet evaporation rate caused by the consequent low temperatures in this region (< 400 °C). This is accompanied by negligible CO$_2$ and CO concentrations (< 1%) and excessive values of O$_2$ (> 19%). At distances of X > 100 mm, where the central-recirculation zone exists, the accelerated evaporation and the improved turbulent mixing rates enhance the chemical reaction as proved by the gradual increase of the temperature and CO$_2$ concentration and decrease of O$_2$ shown in figures 3.32(C), 3.33(C) and 3.35(C) respectively.

Figure 3.36 shows the wall-temperature distributions for the three flames of runs 4, 7 and 10. Again, the trends of the three flames are similar and indicate a steep rise of wall-temperature at the upstream of the combustor with higher values as the swirl number decreases. The significant increase in the wall temperature with the reduction of the air swirl is consistent with the internal gas temperature distributions described previously which showed that higher flame temperatures are located near to the wall. The heat fluxes to the cooling water jacket, table 3.4, are in agreement with these observations. The slightly higher wall-temperatures associated with the 0.81 swirl number flame (run 7) for X < 70 mm are due to the
higher flame temperatures close to the wall-recirculation zone. With X > 200 mm, the wall-temperature decays and achieves nearly constant values. The slight increase associated with the high swirl number flames (S = 1.4 and 0.81) which can be seen in figure 3.36 is probably due to the increase in the heat transfer coefficient as influenced by the higher gas velocity at the near-wall region.

3.3.4.3 Influence of combustion air inlet-temperature

It has been shown in the previous two subsections that the evaporation rate of the droplets plays an important role in the structure and performance of the spray combustion. Therefore, the combustion air was preheated to temperatures of 125 and 225 °C to study the influence of air inlet-temperature on the spray characteristics and flame properties. The results described in this subsection correspond to the two flames of initial spray mean-droplet diameters of 54 and 115 μm and comprise the measurements of mean gas temperature, species concentration, droplet velocities and wall-temperature distributions.

(A) Flames with 54 μm droplet diameter

Figures 3.37 through 3.41 represent the mean gas temperature, concentrations of CO₂, CO, O₂ and UHC respectively for flames of run 4, 13 and 14 which present results for combustion air inlet-temperatures of 20, 125 and 225 °C respectively. In general, the rise of the combustion air inlet temperature has resulted in onset of combustion nearer to the combustion air exit and propagation of the flame near to the combustor wall. The temperature contours of
figure 3.37 indicate that the shear-layer flame region extends to larger radii and is dominated by higher temperature values and steeper gradients as the preheat temperature increases which imply enhancement of the chemical reaction. The CO₂ concentration contours of figure 3.38 show a corresponding increase of the CO₂ accompanied by a higher rate of destruction of CO and relatively higher oxidation of HC as shown in figures 3.39 and 3.41 respectively. It can also be seen that the increased combustion intensity throughout the shear-layer flame zone is associated with a slight increase of the chemical reaction within the near-wall region of R > 80 mm. This is indicated by the increase of CO₂ and reduction of O₂ throughout this region as shown in figures 3.38 and 3.40 respectively. The enhancement of chemical reaction within the shear-layer and propagation of the flame near to the wall region, with combustion air temperature rise, can be attributed to the increased evaporation rates of the droplets coupled with the high turbulent mixing which suggest higher formation and oxidation rates. The increased velocities of the incoming stream of the combustion air, as a result of temperature difference, are likely to aid the spreading of the flame near the combustor wall.

In the upstream part of the recirculation zone, X < 60 mm, the rise of the combustion air inlet-temperature has increased the evaporation rates of the droplets in the vicinity of the fuel spray with consequent rapid formation of CO. The CO results of figure 3.39 demonstrate an increase to 10% within this region. It is likely that the accelerated vaporization and CO formation rates within the upstream part of the recirculating region contribute to the above-mentioned intensified chemical reaction in the shear-layer zone and the spreading of the flame throughout the wall region.

The values of droplet mean axial velocity presented in
Figure 3.42 for the flames of runs 13 and 14 indicate that the rise of the preheat temperature has resulted in negative droplet velocities in the vicinity of the fuel spray, \( X = 25 \text{ mm} \), which means that the fuel droplets have been swept back into the recirculating zone. This is consistent with increase of evaporation rate, with higher preheat temperature, which has reduced the droplet diameter to very small values to follow the reversed gas velocity.

Downstream of the recirculating zone, \( 60 < X < 100 \text{ mm} \), the \( \text{CO}_2 \) contours show slight reductions with increase of the combustion air temperature; this can be attributed to the faster evaporation and \( \text{CO} \) formation rates.

In the region of the atomizer cooling air and around the centreline, the temperature and concentration contours display increase of the \( \text{CO} \) values (from 8 to 11\%) with associated reduction of the \( \text{CO}_2 \) concentration (from 10 to 7\%) as the combustion air temperature increases. This implies that the \( \text{CO} \) oxidation is quenched within this region by the atomizer cooling air and, in particular, the flame of run 14, leading to temperatures of around 1100 \( ^\circ\text{C} \) which, probably, do not allow further oxidation to \( \text{CO}_2 \).

The increased combustion intensity throughout the shear-layer and near-wall region with the rise of the combustion air inlet temperature, described previously, has reduced the fuel convected to the main-flame region. This leads to lower temperatures within this region, in particular at small radii, and reduction of \( \text{CO}_2 \) by up to 2\%. The relatively higher temperatures shown by the flames 13 and 14 along the boundary of the main-flame region are likely due to the dilution effects of the preheated combustion air and the flame propagation.
Figures 3.42 and 3.43 show the radial profiles of droplet mean axial and tangential velocities at different locations in the near-spray region for the preheated flames of runs 13 and 14. Close to the fuel spray injection, $X = 25 \text{ mm}$ and $R = 20 \text{ mm}$, the two flames indicate high mean axial velocities of around 18 m/s which are due to the influences of the high velocity of the cooling air surrounding the atomizer and especially the satellite droplets. The droplet tangential velocity has its highest values at the same location and can be linked to the high rotational speed of the rotating cup atomizer. On moving radially outward, the mean velocity (axial and tangential) decreases significantly as a result of the drag forces acting on the droplets and as the measurement locations coincide with the reversed gas velocities. It is very clear that the increase of the combustion air preheat has resulted in lower velocities and the differences become more significant at locations adjacent to the combustion air exit where negative mean axial droplets velocities have been obtained. The main reason for these low velocities stems from the increased air preheat to a level near to the final boiling point of kerosene (see table 3.1) which results in accelerating the rate of droplet evaporation with consequent reduction in their diameters until they follow the recirculating gas velocity. In the downstream part of the fuel spray, $X > 25 \text{ mm}$, the measured profiles indicate that the droplets have comparatively higher axial velocities associated with lower tangential velocities. These can be attributed to the larger droplets, with their slower evaporation rate and longer life time, which continue to move under their own initial momentum for longer distances. This is consistent with the high levels of unburned hydrocarbons and the liquid kerosene observed in the
sampling system.

The wall-temperature distributions correspond to runs 4, 13 and 14, shown in figure 3.44, have similar trends and the wall-temperature increases with the combustion air inlet-temperature. The convective heat flux to the combustor wall shown in table 3.4 are in agreement with these observations. The main reason for that is the increased heat transfer coefficient due to higher gas velocities resulting from the increased temperature difference of the combustion air and higher gases temperature close to the wall.

(B) Flames with 115 μm diameter droplets

The results corresponding to 115 μm initial mean-droplet diameter flames and different air inlet-temperatures (runs 6, 15 and 16) are shown in figures 3.45 to 3.49 and indicate the influences are more significant than those associated with 54 μm droplet diameter flames. For example, the temperature contours of figure 3.45 show that the local gas temperatures have been increased and the region of high temperature has expanded as the combustion air inlet-temperature increases. However, the species concentrations indicate improvement of the combustion intensity and chemical reaction throughout the whole combustor. The increase of the inlet air temperature raises the general level of the gas temperature within the shear-layer zone, as shown by the near wide spread high temperatures. Close inspection of the shear-layer indicates that the high gas temperatures are associated with the high CO₂ and lower O₂ values while the CO indicates higher formation and oxidation rates and no significant difference is observed in UHC concentration. This can be attributed to the increase of the air temperature and velocity which result in
reduction of the heating up period of the droplets and consequently accelerating of the evaporation rate.

Throughout the recirculating zone and the region surrounding the centreline at the combustor upstream, the increased air temperature has improved the chemical reaction as indicated by the increased temperature difference and the overall rise of CO$_2$ and reduction of CO and UHC concentrations. The slight reduction of CO$_2$ within the central part of the recirculation zone, see figure 3.46(C), is likely due to the higher formation rate of CO and lack of O$_2$ which shows values of less than 2% at this part.

The temperature and concentration contours reveal that the rise of the preheat temperature has improved turbulent mixing rates between the fuel and oxidant throughout the main-flame region with consequent enhancement of the combustion. The main reason is the acceleration of the evaporation rate within the recirculation and shear-layer regions which has resulted in feeding the main-flame zone by higher rates of fuel vapour and CO than liquid droplets.

The increased gases temperature throughout the shear-layer and recirculating zone has allowed shorter evaporation times for the droplets penetrating them to feed the free-stream flame zone. This can be detected from the shifting of the high temperature area near to the wall to smaller axial distances as the air temperature increases. This is accompanied by earlier increase in the oxidation to CO$_2$ and decrease of O$_2$ concentrations at this area as shown in figures 3.46 and 3.48 respectively. The rise of the inlet air temperature also has resulted in higher gas temperatures and CO$_2$ concentration and lower O$_2$ values within the free-stream zone with an implication of enhancement of the chemical reaction.
The wall-temperature distributions of figure 3.50 for the flames of runs 6, 15 and 16 indicate a significant increase with the air inlet temperature as influenced by the enhanced combustion intensity within the free-stream flame zone. The general trends are very similar to those related to the smaller droplet size flame (see figure 3.44). The heat released to the combustor cooling rings indicated in table 3.4 are in agreement with the observed rise of the wall-temperature as the air inlet-temperature increases.

**3.3.4.4 Discussion**

The flame structures presented in this subsection, 3.3.4, reveal strong similarities with the characteristic time model of, for example, Peters and Mellor (1982), Leonard et al (1981), Ballal and Lefebvre (1981), Mellor (1980), Plee et al (1979), Tuttle et al (1977) and Komiyama et al (1977). The basic picture of a liquid fuel spray feeding a diffusion type flame surrounding a region of reverse flow has been observed in different configurations including, apart from the present work, the confined swirl stabilized flame of El-Banhawy and Whitelaw (1981), Pounti, Hutchinson and Whitelaw (1980), Owen et al (1979), confined disc stabilized flame of Tuttle et al (1976), the Allison J-33 combustor of Plee et al (1978), Tuttle et al (1973) and Mellor (1973) and swirl stabilized flame with air preheat of Katsuki et al (1976). In agreement with the present work, the experimental results obtained within these different flames indicate the existence of the shear-layer, main-flame and free-stream reaction zones. Immediately after injection, CO formation takes place and the fuel is transported in the form of small droplets and unburned hydrocarbon vapours, which are more susceptible to aerodynamic
influence, towards the air swirler and feed the shear-layer flame. The main-flame region is observed downstream of the shear-layer and surrounding the combustor centreline where the unburned hydrocarbons and the carbon monoxide continue to oxidize. With comparatively large droplets (sprays of mean-droplet diameter of 77 and 115 μm) and the consequently greater momentum and larger evaporation times, or with lower swirl number (i.e. $S = 0.81$) and all droplet sizes where the recirculating zone is smaller and shifted further downstream, some of the fuel droplets penetrate the shear-layer to the free stream. In the former case, the droplets evaporate and tend to stabilize a free-stream flame (see for example figure 3.24) while in the latter the droplets increase the combustion intensity near to the wall-recirculation region at upstream (see figure 3.30). Also, in agreement with the present work, El-Banhawy and Whitelaw (1981), Tuttle et al (1973, 1976) and Katsuki et al (1976) found the vicinity of the fuel spray to be dominated by high CO concentrations, see for example figure 3.26, which indicates that CO formation rate is high upon injection of the fuel. This is mainly due to the initial fast reaction of the hydrocarbon fuel to form CO compared to its oxidation to $\text{CO}_2$ as indicated by Bowman (1975), Caretto (1976), Tuttle et al (1977) and Jones (1980).

The present results concerning the effect of the mean-droplet diameter of the spray are in agreement with the unconfined flame results described in section 3.2 which show improvement of the chemical reaction with 10% variation in the Sauter mean-droplet diameter. The results, also, generally agree with those of El-Banhawy and Whitelaw (1981) which showed that the variation of the mean-droplet diameter from 33 to 47 and 96 μm had a significant effect on the combustion intensity and flame properties. It must be noted,
however, that El-Banhawy and Whitelaw (1981) used the same atomizer with 50 mm projection inside the combustor and less atomizer cooling air, whereas a projection of 22 mm and approximately twice the cooling air was used here. The measurements of Nicholls et al (1980), using an air blast multi-needle atomizer, revealed a major increase in the rate of rise of flame temperature and burning of hydrocarbons as the mean-droplet size decreased from 57 to 10 μm, moreover, the results obtained in a heavy fuel oil with variation of droplet mean diameter between 89 and 113 μm of Beer (1962) showed a significant effect on both flame propagation and upon the burnout of carbon in the flame. These results indicate that the fuel spray characteristics can modify the combustion performance of the flame regardless of the location and type of the atomizer and the fuel properties.

The measured effect of the combustion air swirl is, generally, in agreement with the results of El-Banhawy and Whitelaw (1981), Owen et al (1979) and Beer (1965) and indicates that an increase in the inlet air swirl can produce major improvements in evaporation, mixing and chemical reaction rates in confined spray flames and especially in the upstream region and can be attributed to changes in the interaction between the fuel spray and air stream. However, the increase in the flame length with decrease of swirl level reported by El-Banhawy et al (1981) contradicts the present results and those of Owen et al (1979) and Beer (1965) and is probably due to the reduction in fuel convected to the main-flame region as a result of injection of the fuel spray further downstream at the recirculating zone. The movement of the main reaction zone to the near-wall region with the 0.47 swirl number flames is different from the previous observation and is a consequence of the present flat spray geometry and the resulting near-wall locations of main
reversed flow. This result emphasizes the influence of the recirculating region to the stabilization mechanism of spray flames.

The influence of combustion air preheat, subsection 3.3.4.3, on the evaporation characteristics of the fuel spray and flame performance has been shown to be significant. The trends agree with the measurements in non-combusting sprays of Yule et al (1982), which showed reduction in the kerosene liquid phase volume concentration in the hot spray compared to the cold one, and those of Mizutani et al (1977), who indicated that the flame was stabilized closer to the nozzle tip with increase of the air temperature. Katsuki et al (1976) who measured the temperature and species concentration in a model gas turbine combustor with air preheated to 320 and 430 °K and domestic kerosene injected at room temperature through a swirl atomizer, also, concluded that a higher inlet air temperature accelerates the combustion reaction in the upstream part of the combustor and results in higher gas temperature throughout the combustor. The present results also show that the evaporation and chemical reaction rates of the smaller droplet diameter flame ($D_m = 54 \, \mu m$) are less influenced by increase of inlet air temperature than the large droplet size flame. This suggests that the characteristic droplet evaporation times of small droplet diameter sprays (for example < 50 \, \mu m) are small compared to the times characteristic of the fuel-air mixing process and it is anticipated that the combustion of these sprays is very similar to the gas diffusion flame. These observations support the conclusion reported by Komiyama et al (1977), concerning the importance of droplet evaporation overall fuel-air mixing process for small droplet size sprays ($D_m = 10 - 60 \, \mu m$), who indicated that where the characteristic time of evaporation is much less than the jet mixing time, the
details of evaporation process are not important and the jet length
scale and kinetic energy govern the mixing process as for gaseous
flame. It also accords with the results of Onuma et al (1975,1977),
who compared gas and spray diffusion flames \(D_m = 10 - 40 \, \mu m\), and
concluded that most of the droplets within the flame evaporate and
generate a cloud of vapour which burns as a gas diffusion flame.
Therefore, it is expected that the use of combustion air preheat in
gas turbine operation will improve the spray evaporation rate and
flame properties with consequent higher efficiencies.

The results presented in this subsection, 3.3.4, show the
importance of droplet evaporation and turbulent mixing of fuel-air
mixture in controlling the combustion process. It is observed that
the reduction in the spray mean-droplet diameter, or increase of
combustion air inlet temperature and/ or swirl number is associated
with improvement in the intensity of the chemical reaction at most of
the combustor domain. The results have been explained mainly by
reference to the relative importance of droplet evaporation and
turbulent mixing in influencing the chemical reactions. It can be
argued, see for example Mellor (1980), Leonard et al (1981), Plee et
al (1979) and Tuttle et al (1977), that the combustion process of
liquid fuelled flames is governed by the importance of characteristic
times of droplet evaporation and ignition delay and burning for a
homogeneous fuel-air mixture compared to the turbulent mixing time
which must be evaluated in the shear-layer between the fresh air and
the recirculating fuel/burned gas mixture. Following Tuttle et al
(1977) and Plee and Mellor (1979), the turbulent mixing time is
assumed to be the characteristic breakdown time of the large-scale
eddies which is related to a macroscopic characteristic dimension
(L') and convective velocity (V'):

\[ t_{sl} = \frac{L'}{V'} \quad (\text{ms}) \quad (3.9) \]

The ignition delay time for a homogeneous fuel-air mixture of equivalence ratio (\( \beta \)), is defined as the inverse of the hydrocarbon reaction rate:

\[ t_d = 10^{-4} e^{E/RT} / \beta \quad (\text{ms}) \quad (3.10) \]

where \( E \) is the activation energy (\( E = 21,000 \text{ cal/mole} \)) and \( R \) is the universal gas constant. The characteristic time of droplet evaporation process is evaluated from "d^2 law" based on the initial mean droplet diameter of the spray.

\[ t_e = \frac{d_i^2}{C} \quad (\text{ms}) \quad (3.11) \]

where \( d_i \) is the initial mean-droplet diameter and \( C \) is the evaporation rate constant under forced convection, see section 4.5. The three characteristic times can be correlated to give an indication of combustion rate controlling parameters and provide clues which assist understanding of physical processes involved in spray combustion systems. The semi-empirical correlation developed by Leonard and Mellor (1981) can be written as:

\[ t_{sl} >> 2.12 \left( \frac{T}{T_{in}} \right) \cdot (t_d + 0.011 t_e) \quad (3.12) \]

The temperature ratio \( (T/T_{in}) \) accounts for the acceleration of air flow as a result of the increased temperature across the shear-layer.
Note that because of various approximations in the definitions of the characteristic times, they are only expected to be order of magnitude estimates that hopefully support the previous physical explanation of the different influences investigated. The turbulent mixing time is calculated for the case of 1.4 swirl number flow by taking the macroscopic length scale as the width of the recirculation zone (which is also a measure of the size of the recirculation zone), see figure 3.22, and the convective velocity is calculated from the measured isothermal velocity components and is found to be 13.12 m/s. The ignition delay and burning time for the flames 4, 6, 14 and 16 are estimated using the corresponding measured temperature and species concentration data presented previously while the droplet evaporation times are estimated based on the evaporation rate constant described in chapter 4. Table (3.5) shows the corresponding estimated order of magnitude of the different time scales together with the resulting right hand side of equation (3.12).

<table>
<thead>
<tr>
<th>Flame No.</th>
<th>D_m (μm)</th>
<th>T_in (°C)</th>
<th>t_sl (ms)</th>
<th>t_d (ms)</th>
<th>t_e (ms)</th>
<th>R.H.S. Eq.(3.12)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>54</td>
<td>20</td>
<td>3.78</td>
<td>0.034</td>
<td>4.65</td>
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<td>6</td>
<td>115</td>
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<td>0.043</td>
<td>9.8</td>
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</tr>
<tr>
<td>14</td>
<td>54</td>
<td>225</td>
<td>3.78</td>
<td>0.028</td>
<td>3.2</td>
<td>0.59</td>
</tr>
<tr>
<td>16</td>
<td>115</td>
<td>225</td>
<td>3.78</td>
<td>0.028</td>
<td>7.3</td>
<td>0.93</td>
</tr>
</tbody>
</table>
Comparison of the right hand side of equation (3.12) for flames 4 and 6 to the turbulent mixing time (see table 3.5) indicates that the droplet evaporation have an increasing influence on the reaction rate with increase of the droplet mean diameter. This is attributed to the direct increase of the evaporation time with consequent increase of the ignition delay time through the resulting lower equivalence ratio. If the swirl number is reduced (S = 0.81 and 0.47), it is expected, however, that the main conclusion will hold equally where the increase in the mixing time will be balanced by corresponding increases of the right hand side of equation (3.12) as influenced by the changes in flow conditions. For the preheated flames 14 and 16, table (3.5) shows that the corresponding characteristic times of droplet evaporation and ignition delay have resulted in further reduction of the right hand side of equation (3.12). This implies that the increase of the inlet-air temperature improves the chemical reaction, however, inspection of the order of magnitude of the corresponding time scales indicates that the small droplet size flame is less influenced. This confirms, again, the previous observation that for small droplet diameter sprays (i.e. D_m < 50 μm), the combustion process is very similar to the turbulent gas diffusion flame.

3.3.5 Summary

The following is a summary of the more important findings and conclusions of this section:

1. The measurements of the isothermal mean axial velocity within the combustor at different flow swirl numbers have shown that:
(a) decrease of the swirl number from 1.4 to 0.81 reduces the size and strength of the off-centreline recirculation zone and develops another small wall-recirculating region at upstream locations of X < 40 mm;

(b) reduction of the swirl number to 0.47 results in the disappearance of the off-centreline recirculation region and development of a weak wall-recirculation zone upstream (X < 100 mm) and another small central one downstream (X > 100 mm).

2. The combusting flow results have shown the existence of the shear-layer, main-flame and free-stream reaction zones and the relative importance of the recirculation region (size and position) and the spray mean-droplet diameter in controlling the flame structure.

3. The influence of the mean droplet-diameter on the combustion performance under the three levels of air swirl has indicated that:

(a) for the high swirl flame (S = 1.4), the increase of the droplet diameter from 54 to 115 μm, which is associated with an increase of the droplet evaporation time, results in a reduction of fuel-air mixing and chemical reaction rates (particularly within the main-flame region) and development of free-stream flame due to the significant influence of droplet penetration;
(b) the reduction of the swirl number to 0.81 reveals similar trends to that noted in (a) except that the free-stream flame disappears and the reaction rate is enhanced close to the wall-recirculation zone;

(c) the results obtained with 0.47 swirl number flow demonstrate similar trends to the high swirl number cases of reduction in the combustion intensity, especially in the shear-layer and main-flame regions as the droplet diameter increases. However, the influence is more significant at the central part of the flame due to the different characteristics of the flame structure.

4. The influence of the swirl number has emphasized the importance of the contribution of the recirculating region for spray flame stabilization and has shown that:

(a) flows with swirl numbers of 1.4 and 0.81, the flame is stabilized on the off-centreline recirculation zone and the change of the swirl number from 1.4 to 0.81 results in:

(i) reduction of the size of the shear-layer flame with a shift towards small radii,

(ii) reduction of the combustion intensity within the main-flame region,

(iii) increase of the chemical reaction rate within the central and wall regions
upstream;

(b) the reduction of the swirl number to 0.47 is associated with stabilization of the flame on the wall-recirculation region and leads to:

(i) movement of the shear-layer and the main-flame regions to near the wall,

(ii) greater reduction in the combustion intensity within the shear-layer and main-flame zones due to both low evaporation and turbulent mixing. The temperature is reduced by at least 200 °C when compared with the high swirl flames,

(iii) quenching of the chemical reaction rate and severe reduction in the temperature within the central part of the flame as influenced by the high velocity air stream surrounding the atomizer.

5. The effect of increasing the combustion air inlet temperature from 20 to 225 °C is to:

(a) increase the temperature levels throughout the combustor;

(b) cause onset of combustion nearer to the swirl exit;

(c) accelerate the evaporation rates of the droplets, especially with the larger diameter ($D_m = 115 \mu m$) with consequent improvement of the fuel-air mixing
and combustion intensity within the main-flame region.

6. The influences of the droplet diameter and inlet air temperature are examined by calculating the order of magnitude of turbulent mixing, droplet evaporation and ignition delay and burning time scales which have confirmed that:

(a) reduction of the droplet diameter reduces the contribution of the evaporation and ignition delay times in controlling the combustion process with consequent enhancement of the reaction rate;

(b) the influence of the preheat air-temperature on the evaporation rate and flame properties is small for the small droplet diameter flame ($D_m = 54 \, \mu m$). This suggests that the combustion of smaller droplet diameter sprays is very similar to the turbulent gas diffusion flame.
Fig. 3.13a. Diagram for the axi-symmetric combustor used for the confined flame experiment.
Fig. 3.13b. Cross-section at A - A.

Fig. 3.13c. Details of the combustor upstream.
Chromel-Alumel Thermocouple

Fig. 3.13d. Arrangement for wall-temperature measurement (detail at B).

Fig. 3.14. Diagram of probe insertion inside the combustor.
Cooling water jacket:
(4) 1st section
(5) 2nd section
(6) 3rd section
(7) 4th section

(3) Observation ports
(8) Cooling water inlet
(9) Cooling water exit

Plate (3.1) Photograph of the combustion chamber.
Fig. 3.15. Flow diagram and arrangement of the equipments for the confined flame experiment.
Plate (3.2) Photograph of the layout of the confined flame experiment.
Fig. 3.16. Radial profiles of mean axial velocity, run 1.

X: Axial distance measured from the swirler exit.
R: Radial distance measured from the combustor centre-line.
Fig. 3.17. Radial profiles of rms velocity fluctuation, run 1.
Fig. 3.18. Radial profiles of the mean axial velocity, run 2.
Fig. 3.19. Radial profiles of rms velocity fluctuation, run 2.
Fig. 3.20. Radial profiles of the mean axial velocity, run 3.
Fig. 3.21. Radial profiles of rms velocity fluctuation, run 3.
Fig. 3.22. Contour maps of the mean axial velocity at combustion air swirl number of (i) $S = 1.4$, (ii) $S = 0.81$, (iii) $S = 0.47$. 
Fig. 3.23. Basic flame structures for the investigated flames.

(A) $S=1.4$ and negligible fuel droplet penetration.

(B) $S=1.4$ and significant fuel droplet penetration.
Fig. 3.23. continued.

(C) $S = 0.81$

(D) $S = 0.47$
Fig. 3.24. Temperature contour maps (temperature in °C)
(A) run 4, (B) run 5, (C) run 6.
Fig. 3.25. Contour maps of the percentage $\text{CO}_2$ volume concentration.
(A) run 4, (B) run 5, (C) run 6
Fig. 3.38. Contour maps of the percentage CO volume concentration.

(A) run 4, (B) run 5, (C) run 6
Fig. 3.38. Contour maps of the percentage $O_2$ volume concentration.
(A) run 4, (B) run 13, (C) run 14
Fig. 3.28. Contour maps of the percentage UHC volume concentration.
(A) run 4, (B) run 5, (C) run 6
Fig. 3.29. Wall-temperature distribution, runs 4, 5 and 6.
Fig. 3.30. Temperature contour maps (temperature in °C)

(A) run 7, (B) run 8, (C) run 9.
Fig. 3.31. Temperature contour maps (temperature in °C)
(A) run 10, (B) run 11, (C) run 12
Fig. 3.32. Temperature contour maps (temperature in °C)

(A) run 4, (B) run 7, (C) run 10
Fig. 3.33. Contour maps of the percentage CO$_2$ volume concentration.

(A) run 4, (B) run 7, (C) run 10
Fig. 3.34. Contour maps of the percentage CO volume concentration.
(A) run 4, (B) run 7, (C) run 10
Fig. 3.35. Contour maps of the percentage $O_2$ volume concentration.

(A) run 4, (B) run 7, (C) run 10
Fig. 3.36. Wall-temperature distribution, runs 4, 7 and 10.
Fig. 3.37. Temperature contour maps (temperature in °C)
(A) run 4, (B) run 13, (C) run 14
Fig. 3.38. Contour maps of the percentage CO₂ volume concentration.  
(A) run 4, (B) run 13, (C) run 14
Fig. 3.39. Contour maps of the percentage CO volume concentration.
(A) run 4, (B) run 13, (C) run 14
Fig. 3.38. Contour maps of the percentage $O_2$ volume concentration.
(A) run 4, (B) run 13, (C) run 14
Fig. 3.38. Contour maps of the percentage CO\textsubscript{2} volume concentration. (A) run 4, (B) run 13, (C) run 14.
Fig. 3.42a. Droplet mean axial velocity and the corresponding rms of velocity fluctuations, run 13.
Fig. 42b. Droplet mean axial velocity and the corresponding rms of velocity fluctuations, run 14.
Fig. 3.43a. Droplet mean tangential velocity and the corresponding rms of velocity fluctuation, run 13.
Fig. 3.43b. Droplet mean tangential velocity and the corresponding rms of velocity fluctuation, run 14.
Fig. 3.44. Wall-temperature distribution, runs 4, 13 and 14.
Fig. 3.45. Temperature contour maps (temperature in °C)

(A) run 6, (B) run 15, (C) run 16
Fig. 3.46. Contour maps of the percentage $\text{CO}_2$ volume concentration.  
(A) run 6, (B) run 15, (C) run 16
Fig. 3.47. Contour maps of the percentage CO volume concentration.
(A) run 6, (B) run 15, (C) run 16
Fig. 3.48. Contour maps of the percentage $\text{O}_2$ volume concentration.

(A) run 6, (B) run 15, (C) run 16
Fig. 4.49. Contour maps of the percentage UHC volume concentration.
(A) run 6, (B) run 15, (C) run 16
Fig. 3.50. Wall-temperature distribution for runs 6, 15 and 16.
CHAPTER 4

PREDICTION OF FLOW PROPERTIES IN THE
AXISYMMETRIC COMBUSTOR

4.1 Introduction

This chapter describes the prediction method used for the calculation of the flow aerodynamics and the combusting flow properties for the axisymmetric combustor described in chapter 3 and reports results for droplet-sizes of 54 μm and 115 μm mean-diameter flames. The mathematical formulation comprises the application of Eulerian conservation equations to the gas-phase and Lagrangian equations of the droplet motion and thermal balance to the mono-disperse droplet-size representing the spray. The two sets of equations are explicitly coupled through droplet source terms, including the mass, heat and momentum, which are calculated from the solution of the droplet field equations and by utilizing a droplet tracking-technique. The spray combustion model assumes that the evaporating droplets act as distributed point injectors of fuel vapour within the flame and that the combustion process of the fuel vapour can be regarded as gaseous.

Sections 2 through 6 of the present chapter provide a description of the equations solved together with the turbulence, combustion, spray and radiation models used with these calculations. Section 4.7 gives a brief description for the boundary and inlet conditions and the numerical procedure is summarized in section 4.8. The results of the calculations and comparison with the corresponding experimental data are presented and discussed in section 4.9. The chapter ends with brief concluding remarks.
4.2 The differential equations

The equations representing conservation of mass, momentum, enthalpy and species concentration for a chemically reacting, turbulent and droplet-free gas phase can be expressed as:

Continuity

\[ \frac{\partial \rho}{\partial t} + \sum_{i} \left( \frac{\partial (\rho U_i)}{\partial x_i} \right) = 0 \]  

\( (4.1) \)

Momentum

\[ \frac{\partial (\rho U_i)}{\partial t} + \sum_{j} \left( \frac{\partial (\rho U_i U_j)}{\partial x_j} \right) = \frac{1}{\rho} \sum_{j} \left( \frac{\partial P}{\partial x_j} \right) + \frac{\gamma - 1}{\rho} \frac{\partial \rho}{\partial x_j} \left( \sum_{j} \frac{\partial U_j}{\partial x_j} \right)^2 - \frac{\gamma}{\rho} \frac{\partial P}{\partial x_j} \]  

\( (4.2) \)

Enthalpy or species concentration

\[ \frac{\partial (\rho \phi)}{\partial t} + \sum_{i} \left( \frac{\partial (\rho U_i \phi)}{\partial x_i} \right) = \frac{1}{\rho} \sum_{i} \left( \frac{\partial P}{\partial x_i} \right) \phi + \frac{\gamma - 1}{\rho} \frac{\partial \rho}{\partial x_i} \phi \frac{\partial \phi}{\partial x_i} + S_{\phi} \]  

\( (4.3) \)

The gas density and temperature for low Mach number can also be expressed as:

\[ \rho = \rho(\phi) \quad \text{and} \quad T = T(\phi) \]  

\( (4.4) \)

Equations (4.1), (4.2) and (4.3) with the appropriate boundary conditions and auxiliary expressions for the calculation of density
and for the form of enthalpy-temperature dependence, represent a closed set of equations for the instantaneous dependent variables.

In turbulent flows, reacting or otherwise, the variations in the dependent variables cover a wide range of time and length scales as to preclude the direct numerical solution of the above equations. As a consequence the accepted approach is to decompose the dependent variables into a mean and fluctuating components and to average so that the equations describe the transport of mean quantities. For variable density flows, the decomposition can be unweighted or density-weighted as suggested by Favre (1969), extension discussion of the averaging procedure and their implications have been provided, for example, by Jones and Whitelaw (1981,1982), Jones (1980) and Bilger (1976). The present formulation is based on an unweighted decomposition which can be represented by:

\[ U = \bar{U} + u^* \quad , \quad \phi = \bar{\phi} + \phi^* \quad \text{etc} \]

where \( \bar{U} = \lim_{t_0+\tau} \int_{t_0}^{t_0+\tau} U dt \) and \( u^* = 0 \)

With unweighted averaging, equation (4.1) to (4.3) may be written as:

\[ \frac{\partial}{\partial x_i} \left( \bar{\rho} \bar{U}_i + \bar{\rho} u^*_i \right) = 0 \quad (4.5) \]

\[ \frac{\partial}{\partial x_i} \left( \bar{\rho} U_j + \bar{\rho} u^*_j \bar{u}^*_i + \bar{U}_j \bar{u}^*_i + \bar{U}_i \bar{u}^*_j + \bar{\rho} u^*_i u^*_j \right) \]

\[ = \frac{\partial}{\partial x_i} \left( \frac{\partial \bar{U}_j}{\partial x_j} + \frac{\partial \bar{U}_i}{\partial x_i} \right) - \frac{\partial \bar{F}}{\partial x_j} \quad (4.6) \]
Due to the complexity of, and uncertainties associated with, the transport equations for the density correlation terms appearing in equations (4.5) to (4.7), all density correlation terms have been omitted in the final set of equations in the present study. This neglect reduces the unweighted equations to a form identical to the Favre equations and their solution is likely to yield density weighted values as indicated by Jones and Whitelaw (1981, 1982) and Bilger (1975). Therefore, this feature should be considered during the comparison of the predicted results with the measurements.

Equations (4.5), (4.6) and (4.7), after the omission of the density correlation terms and substitution of appropriate expressions for the Reynolds stresses and turbulent fluxes terms obtained from the turbulence model, can be written in the same general form which, for two-dimensional turbulent axisymmetric combusting and steady gas flow with recirculation, is:

$$\frac{3}{3x_1} (\bar{p} \bar{u}_1 \bar{\phi} + \bar{p} \bar{u}_1 \bar{\phi} + \bar{r} \bar{\phi} \bar{u}_1 + \bar{u}_1 \bar{\phi} \bar{\phi} + \bar{\phi} \bar{u}_1 \bar{\phi})$$

$$= \frac{3}{3x_1} \Gamma_{\phi} \frac{\partial \bar{\phi}}{\partial x_1} + S_{\phi}$$

(4.7)

Consequently:

$$\bar{\rho} = \rho(\bar{\phi}) \quad \text{and} \quad \bar{T} = T(\bar{\phi})$$

(4.8)

The source terms $S_{\phi}$ represents the generation or destruction of the
dependent variable, \( \psi \), by processes concerned only with the gas phase and additional source term, \( S_d \), represents the generation or destruction due to the fuel droplets and will be described later in subsection 4.8.2. Table 4.1 gives the exact expressions for the change coefficients, \( \Gamma_{\psi \text{eff}} \), and source terms, \( S_\psi \) and \( S_d \) corresponding to the different dependent variables used in this work.

4.3 Turbulence model

In order to obtain the values of Reynolds stress terms, \( \overline{u_i u_j} \), and the turbulent fluxes, \( \overline{u_i \psi} \) and close the system of equations, the correlations must be approximated in terms of known calculable quantities. Turbulence models based on direct closure of the equations for the components of the stress tensor have previously been proposed in the context of isothermal flows, see for example Bradshaw, Cebeci and Whitelaw (1981), Lumley (1978), Bradshaw (1976) and Launder (1975) and their extension to reacting flows with associated large density variation has been considered by Jones and Whitelaw (1981, 1982) and Jones (1980). The approach to turbulence models commonly used for recirculating flows is the form of two-equation model introduced by Jones and Launder (1972) because higher order models cannot be justified with the uncertainties introduced by the additional assumptions and particularly the combustion and droplet models. The model involves an assumed linear relationship between the Reynolds stress and rate of strain:

\[
\rho \overline{u_i u_j} = \frac{2}{3} \delta_{ij} (\bar{\partial^2} + \nu_{\text{eff}} \frac{\partial u_i}{\partial x_j} - \nu_{\text{eff}} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right))
\] (4.10)
For turbulent fluxes or scalar quantities a gradient diffusion model is used viz;

\[ \rho \cdot \bar{u}_i \phi = - (\mu_t/\sigma_t) \frac{\partial \phi}{\partial x_i} \]  \hspace{1cm} (4.11)

The effective viscosity, \( \mu_{\text{eff}} \), can be defined as follows:

\[ \mu_{\text{eff}} = \mu_\ell + \mu_t \]  \hspace{1cm} (4.12)

where \( \mu_\ell \) is the molecular laminar viscosity and \( \mu_t \) is the turbulent (or eddy) viscosity. The laminar viscosity can be neglected with respect to the turbulent one except in very small region near the walls. The turbulent (or eddy) viscosity is given by:

\[ \mu_t = c_u \cdot \rho k^2/\varepsilon \]  \hspace{1cm} (4.13)

where \( k \left( = \bar{u}_i^2/2 \right) \) and \( \varepsilon \) are the turbulence kinetic energy and dissipation rate respectively and, to close the set of equations, transport equations for \( k \) and \( \varepsilon \) are necessary. These two equations have been derived in detail by many modellers, see for example Jones and Launder (1972), and may be written as:

**k-equation**

\[ \rho \cdot \bar{u} \frac{\partial k}{\partial x} + \frac{\partial \varepsilon}{\partial x} \cdot \frac{\partial (k \varepsilon)}{\partial x} = \frac{3}{2} \left( \frac{\mu_{\text{eff}}}{\sigma_k} \right) \frac{\partial k}{\partial x} + \frac{3}{2} \frac{\partial}{\partial x} \left( r \frac{\mu_{\text{eff}}}{\sigma_k} \frac{\partial k}{\partial x} \right) \]

\[ + \mu_{\text{eff}} \cdot G - \rho \varepsilon \]  \hspace{1cm} (4.14)
\[ e \text{-equation} \]
\[ \rho U \frac{\partial e}{\partial x} + \frac{\partial V}{\partial r} \cdot \frac{3 (r e)}{3 r} = \frac{3}{3 x} \left( \frac{\mu_{\text{eff}}}{\rho} \cdot \frac{\partial e}{\partial x} \right) + \frac{1}{r} \frac{\partial }{\partial r} \left( \frac{\mu_{\text{eff}}}{\rho} \cdot \frac{\partial e}{\partial r} \right) \]
\[ + \mu_{\text{eff}} \cdot \frac{C_1 e}{k} G - \frac{C_2 \rho}{k} \frac{e^2}{k} \quad (4.15) \]
where
\[ G = 2 \left( \frac{\partial U}{\partial r}^2 + \frac{\partial V}{\partial r}^2 + \frac{\partial W}{\partial r}^2 \right) + \frac{\partial W}{\partial r}^2 + \left( \frac{\partial}{\partial r} \left( \frac{W}{r} \right) \right)^2 + \frac{\partial U}{\partial x}^2 + \frac{\partial V}{\partial x}^2 \]

The turbulence model constants \( C_1, C_2, C_\mu, \sigma_k \) and \( \sigma_e \) have been discussed extensively, see for example Jones and Launder (1972), Launder (1974) and Jones (1980). However, it must be recognized that these constants are based on the characteristics of constant density flows. Whether these assumptions are valid for variable density flows and whether explicit modifications are required are questions which remain to be answered. The model has, however, been used extensively for calculation of turbulent reacting flows, see for example El-Banhawy and Whitelaw (1980a, 1980b), Abdalla et al (1981), Gosman et al (1981) and Magnussen et al (1977), and has shown to yield satisfactory results. The values of the empirical constants and turbulent Prandtl numbers used in the present work are indicated in table 4.2.

4.4 Combustion model

The combustion model must provide a method of evaluating the mean formation rate of each species present, and in addition, allow the calculation of the mean fluid temperature and density of
the mixture. The chemical reaction associated with the change from the reactants to the final products involves many intermediate species and free radicals, see for example Jones and Whitelaw (1982), Jones (1980) and Abdalla et al (1982b), which complicate the modelling of the combustion process. In addition the turbulence adds greatly to the complexity through combustion-turbulence interactions which are incompletely understood, see for example Bray (1979), Mellor (1979) and Borghi et al (1979). As a consequence there are different approaches to the determination of the mean reaction rate, \( S_p \), in the conservation equation (4.9) for the chemical species involved in the reaction. In this work, the eddy-dissipation combustion model of Magnusen and co-workers (1977,1978,1979) has been used. It can accommodate diffusion, premixed and arbitrary fuelled flames and is relevant to the present spray flame geometries where the finite rate of droplet evaporation suggests that the fuel supply resembles the conditions of arbitrary fuelled flames. It is assumed that all species and heat transfer have equal diffusion coefficients; i.e. the turbulent Prandtl and Schmidt numbers are equal, and that the reaction takes place as a global, one-step, infinitely fast chemical reaction between the fuel and oxidant where both combine in stoichiometric proportions. A consequence of these assumptions is that chemical equilibrium prevails everywhere in the flame and that the thermodynamic state of the mixture is related to a single, strictly conserved, scalar variable for which a transport equation of the type of equation (4.9) (with no source term) can be solved.

In the eddy dissipation model, the rate of reaction is assumed to be determined by the rate of intermixing of fuel and oxygen on a molecular scale or effectively by the rate of dissipation
of the eddies. In turbulent flows, this molecular mixing is associated with the smallest scales of turbulence and the process of molecular mixing is similar to the dissipation of turbulent kinetic energy. The model also assumes that the dissipation takes place mainly in concentrated highly strained regions that occupy only a fraction of the total volume. These regions are occupied by fine structures, with characteristic dimensions of the order of the Kolmogorov microscale, which are responsible for the dissipation of turbulence into heat. Therefore, the transfer of mass per unit mass and unit time between fine structures and the surrounding fluid can be expressed by, see Maggnussen et al (1979):

\[ \dot{m} = 23.6 \left( \frac{v \varepsilon}{k^2} \right)^{0.25} \left( \frac{\varepsilon}{k} \right) \]  

(4.16)

If the rate of reaction between fuel and oxygen is considered infinitelty fast, the rate of combustion will be limited by the mass transfer between the bulk fluid and the fine structures represented by equation (4.16). In this case, the concentration of fuel and oxygen within the reacting structures will be small. If a reaction takes place in all the fine structures, the rate of combustion will be expressed by:

\[ R_{\text{fu}} = 23.6 \left( \frac{v \varepsilon}{k^2} \right)^{0.25} \left( \frac{\varepsilon}{k} \right) \bar{\dot{m}}_{\text{min}} \cdot \bar{\rho} \]  

(4.17)

where \( \bar{\dot{m}}_{\text{min}} \) is the smaller of \( \bar{\dot{m}}_{\text{fu}} \) or \( \bar{\dot{m}}_{\text{ox}} / i \). However not all the fine structures will be sufficiently heated to react. The fraction of the fine structures which reacts can be assumed to be proportional to the ratio of the local concentration of reacted fuel and the total fuel concentration. Consequently:
\[ F = \frac{\bar{m}_{pr}(1+i)}{\bar{m}_{pr} + \bar{m}_{pr}(1+i)} \] (4.18)

where \( \bar{m}_{pr} \) is the local mass fraction of reaction products given by:

\[ \bar{m}_{pr} = 1 - \bar{m}_{ox} - \bar{m}_{fu} \] (4.19)

By combining equations (4.17) and (4.18), the following general expression is obtained for the rate of combustion at infinite reaction rate between fuel and oxygen:

\[ R_{fu} = 23.6 \left( \frac{\nu_c}{k} \right)^{0.25} \left( \frac{\epsilon}{k} \right) \bar{m}_{min} \cdot F \cdot \bar{F} \] (4.20)

The solution of conservation equation (4.9) for the mass fraction of the fuel, \( \bar{m}_{fu} \), using the combustion rate expressed by equation (4.20), provides the local values of the averaged mass fraction of fuel. The mixture fraction, \( \bar{F} \), defined as the mass fraction of fuel both burnt and unburnt, can be obtained by solving a transport differential equation for the time averaged mixture fraction which has no source term and can be expressed for two-dimensional, axi-symmetric turbulent flow in the presence of fuel droplet as:

\[ \frac{\partial}{\partial x} (p\bar{U}\bar{F}) + \frac{1}{r} \frac{\partial}{\partial r} (r p \bar{V} \bar{F}) = \] \[ \frac{\partial}{\partial x} (\Gamma_{eff} \frac{\partial \bar{F}}{\partial x}) + \frac{1}{r} \frac{\partial}{\partial r} (r \Gamma_{eff} \frac{\partial \bar{F}}{\partial r}) + S_d \] (4.21)

The mass fraction of oxygen, \( \bar{m}_{ox} \), is calculated from this expression:
\[ \phi = \frac{\bar{m}_{\text{fu}}}{i} - \left( \frac{\bar{m}_{\text{ox}}}{i} \right) = \left( \phi_F - \phi_A \right) \bar{T} + \phi_A \]  

(4.22)

where \( \phi \) is a passive scalar of the flow and subscripts \( F \) and \( A \) refer to the values appropriate to the fuel and air streams respectively. The mass fraction of the products \( \bar{m}_{\text{pr}} \), is calculated from equation (4.19).

The local values of the mean enthalpy were obtained by solving the transport equation (4.9) given in table 4.1 with a radiation source term. The mean enthalpy in the present study is defined as:

\[ h = \sum_{i}^{T} \frac{m_i}{\bar{T}_{\text{ref}} P_i} \int_{T_{\text{ref}}}^{T} C_p(T) dT + m_{\text{fu}} H_{\text{fu}} \]  

(4.23)

The reference temperature, \( T_{\text{ref}} \), was taken as 288 °K and the individual constant pressure specific heats, \( C_{p_i} \), were determined from the third order polynomial in temperature given by Tribus (1961). The local mean gas temperature was obtained from the calculated enthalpy from the conservation equation and equation (4.23):

\[ h - m_{\text{fu}} H_{\text{fu}} + \sum_{i}^{T} m_i \int_{T_{\text{ref}}}^{T} C_p(T) dT \]  

\[ T = \frac{1}{\left(1/T\right) \sum_{i}^{T} m_i \int_{T_{\text{ref}}}^{T} C_p(T) dT} \]  

(1/T) \sum_{i}^{T} m_i \int_{T_{\text{ref}}}^{T} C_p(T) dT

(4.24)

The local mean gas density, \( \rho \), appearing in the previous equations is calculated from the equation of state of the perfect gases:
\[ \rho = \frac{P}{(RT/M_{\text{mix}})} \quad (4.25) \]

where \( R \) is the universal gas constant and \( M_{\text{mix}} \) is the mixture molecular weight defined as:

\[ \frac{1}{M_{\text{mix}}} = \sum_{i} \frac{m_i}{M_i} \quad (4.26) \]

where \( m_i \) is the mass fraction of species \( i \) calculated through the combustion model. It should be recognised that Jones and Whitelaw (1982) and Jones (1983) have indicated that the common evaluation of the conventional density by this manner is inconsistent with the solution of the conservation equations and will yield erroneous results. This feature should be considered during the comparison with the experimental data.

4.5 Spray model

The spray model is used to represent, both physically and mathematically, the fuel spray and is based on the assumption that liquid droplets act as distributed sources of fuel vapour. The experimental studies of spray evaporation and comparisons of spray flames and turbulent gas flames support this assumption as reported by Onuma et al (1977b), Kamiyama et al (1977), Yule et al (1982) and Negus (1982). The model considered is of the droplet-tracking type and has been described in detail by El-Banhawy and Whitelaw (1980b). It assumes that the fuel is injected into the combustion domain as a fully atomized spray which consists of spherical droplets. As mentioned before (subsection 3.3.1) the spray had two initial droplet sizes (main and satellite), but it was represented here by six size ranges for better distribution of the fuel within the calculation.
domain. For each size range droplet velocities, size and temperature history within the flame can be obtained by solving the respective balance equations for the droplet representing this size range. The model also assumes that the rate of number of droplets associated with each size range is constant along its trajectory and this implies that the turbulent diffusion of droplets and the effects of droplet collision and break-up are neglected. These assumptions are necessary to enable the calculation of the droplet source terms, $S_d$, for each calculation cell (see subsection 4.8.2) which represent the coupling between the droplet and gas field. These source terms are obtained by tracking droplets throughout their movement inside the calculation domain using the Lagrangian framework suggested by Crowe et al (1977). The following paragraph describes the equations used to describe the behaviour of the fuel droplets.

For each size range, the momentum conservation equations for the axial, radial and circumferential directions assume that the drag, inertial and pressure forces are the only acting forces on the droplets, and may be written as; see Hinze (1971):

$$\frac{3U}{3t} = - \frac{1}{\rho_p} \left( \frac{3P}{3x} \right) + \frac{3}{4} \frac{\mu_D}{\rho_p D_p^2} (U - U_p) \cdot Re_p$$ (4.27)

$$\frac{3V}{3t} = - \frac{1}{\rho_p} \left( \frac{3P}{3r} \right) + \frac{3}{4} \frac{\mu_D}{\rho_p D_p^2} (V - V_p) \cdot Re_p + \frac{W_r^2}{r_p}$$ (4.28)

$$\frac{3W}{3t} = \frac{3}{4} \frac{\mu_D}{\rho_p D_p^2} (W - W_p) \cdot Re_p - \frac{V \cdot W_r}{r_p}$$ (4.29)
where \( \text{Re}_p \) is the droplet Reynolds number based on the droplet diameter, \( D_p \), and the relative velocity, defined as:

\[
\text{Re}_p = \frac{\rho |\nabla - \nabla |_p \cdot D_p}{\mu}
\] (30)

A fair amount of experimental data is available for the drag coefficient, \( C_D \), but because of the diversity of the experimental conditions, no precise correlations are applicable over the whole range of interest. The correlations given by Williams (1973) which have been used by several investigators, for example El-Banhawy and Whitelaw (1980a, 1980b), Abbas (1983) and Boysan et al (1982) and found to be reasonably accurate, are used in the present study, i.e.

\[
C_D = 27.0\text{Re}_p^{-0.84}
\] \( 0 < \text{Re}_p < 80 \)

\[
C_D = 0.271\text{Re}_p^{0.271}
\] \( 80 < \text{Re}_p < 10^4 \)

\[
C_D = 2
\] \( \text{Re}_p > 10^4 \)

The effects of evaporation and droplet distortion on \( C_D \) are included implicitly in the preceding correlations.

The expression for the rate of change of the droplet diameter with respect to the time is based on a single droplet. In the case of forced convection evaporation, this can be written as, see Agoston et al (1957):

\[
\frac{dD_p}{dt} = C/2D_p = (C_b/2D_p)(1 + 0.24\text{Re}_p^{0.5})
\] (4.31)

\( C \) and \( C_b \) are the evaporation rate constants under forced convection.
and stagnant conditions respectively and their values depend on the type of fuel used as well as the properties and temperature of the surrounding medium. A widely used expression for \( C_b \), derived from quasi-steady analysis of droplet combustion, is that given by Wise et al (1958):

\[
C_b = \left( \frac{3 \lambda_f}{\rho_f C_p f} \right) \ln(1+B) \tag{4.32}
\]

where \( B \) is the transfer number given by:

\[
B = \frac{C_p}{f} \left( T_g - T_s \right) / L \tag{4.33}
\]

The final expression for the droplet size change can be expressed as:

\[
\frac{dD}{dt} = \left( \frac{4 \lambda_f}{\rho_f C_p f} . D \right) . \left( 1 + 0.24 \text{Re}^{0.5} \right) . \ln[1 + C_p \left( T - T_s \right) / L] \tag{4.34}
\]

To allow for the heating up time of the droplet after entry into the gas stream, it is assumed that the evaporation begins when the droplet temperature reaches the boiling point and the temperature is uniform across the radius. The equation for the rate of change of droplet temperature can be expressed as:

\[
\frac{dT}{dt} = \left( \frac{6. \lambda_f}{\rho_f C_p P_r D^2} \right) . \left( 2 + 0.6 \text{Re}^{0.5} \cdot P_r^{0.33} \right) . \left( T - T_s \right) \tag{4.35}
\]

where the Nusselt number for forced convection motion of the droplets has been replaced by the familiar expression of Ranz and Marshall (1952):

\[
Nu = \left( 2 + 0.6 \text{Re}^{0.5} \cdot P_r^{0.33} \right) \tag{4.36}
\]
Equations (4.27) to (4.29), (4.34) and (4.35) represent a closed set of equations which, with appropriate boundary conditions and knowledge of gas properties, provide the variation of the droplet velocity, size and temperature in the time domain. The corresponding droplet properties in the space domain, which allow the calculation of the droplet source terms, are obtained by droplet-tracking, see subsection 4.8.2.

4.6 Radiation model

The radiation heat transfer between the combustion gases and the combustor walls was calculated using the "four flux" radiation model proposed by DeMarco and Lockwood (1975). It has been applied by several research workers for different flames, see for example El-Banhawy and Whitelaw (1980a), Khalil and Truelove (1977), Lockwood et al (1976) and Abbas (1983), and has yielded satisfactory results. In general, the equation of radiant energy transfer within an absorbing-emitting non-scattering grey medium in local thermodynamic equilibrium may be written as:

\[
\frac{3I(r,Ω)}{3s} = -K_Ω I(r,Ω) + (K_Ω/π) \cdot σ T^4(r)
\]  

(4.37)

where \( I(r,Ω) \) is the total radiant intensity at the point \( r \) in the direction specified by the unit vector \( Ω \), \( s \) is the distance measured in the direction \( Ω \), \( K_Ω \) is the absorption coefficient of the medium and \( T \) is the temperature. In the "four flux" radiation model, the directional dependence of the radiant intensity is approximated by a truncated Taylor series expression as:
where \( A_1, A_2, A_3, B_1, B_2 \) and \( B_3 \) are functions of \( r \) only and \( x', y' \) and \( z' \) are the direction cosines of the unit vector \( \mathbf{n} \). A set of flux equations is obtained by substituting \( I(r,\mathbf{n}) \) from equation (4.38) into the transport equation (4.37) and integrating over six selected solid angles. For an axisymmetric cylindrical geometry the flux equations are:

**radial direction**

\[
\frac{1}{r} \frac{\partial}{\partial r} \left( \frac{1}{k} \frac{\partial I_r}{\partial r} \right) = 1.33 \kappa_g \left( 2R_r - R_x \right) - 1.33 \kappa_g \sigma_\pi^4
\]

(4.39)

**axial direction**

\[
\frac{\partial}{\partial x} \left( \frac{1}{k} \frac{\partial I_x}{\partial x} \right) = 1.33 \kappa_g \left( 2R_x - R_r \right) - 1.33 \kappa_g \sigma_\pi^4
\]

(4.40)

where \( R_r \) and \( R_x \) are the radial and axial total radiation fluxes respectively defined in terms of coefficients of four term Taylor's series which are presumed to represent the spatial variation of radiation intensity at a specified combustor location;

\[
i.e. \ R_r = (2B_1 + B_2 + B_3)
\]

(4.41a)

\[
R_x = (B_1 + B_2 + 2B_3)
\]

(4.41b)

Equations (4.39) and (4.40) are supplemented by the boundary conditions at the wall, derived by considering conservation of radiation energy when applied to a wall; see DeMarco et al (1975):
For the radial wall

\[ \varepsilon_{w}^{R} + \frac{2}{3K_{g}} \frac{3(rR_{g})}{3r} (2 - \varepsilon_{w}) = \varepsilon_{w}^{\text{rad}} \quad (4.42) \]

and for axial wall

\[ \varepsilon_{w}^{R} + \frac{2}{3K_{g}} \frac{3R_{g}}{3x} (2 - \varepsilon_{w}) = \varepsilon_{w}^{\text{rad}} \quad (4.43) \]

where the positive and negative signs are associated with the negative and positive coordinate directions respectively.

The radiant energy contribution to the source term in the enthalpy equation is expressed as:

\[ S_{n} = \frac{16}{9} K_{g} (R_{x} + R_{r} - 2T_{g}^{4}) \quad (4.44) \]

The radiative characteristics of the gas medium are represented by the one clear and two grey gases as proposed by Truelove (1976) and the local gas mixture absorption coefficient, \( K_{g} \), was calculated from his recommended data.

### 4.7 The inlet and boundary conditions

In addition to the equations reported in the previous sections, the specification of the mathematical problem requires the determination of the boundary and initial conditions.

#### 4.7.1 Boundary conditions

The condition at the axis of symmetry is that of zero gradient that is, zero fluxes of convection and diffusion. This
condition is incorporated by setting the velocity component normal to the axis of symmetry as well as the diffusion flux coefficient to zero. At the outlet plane, provided that the downstream is located where the flow has become parabolic or boundary layer in behaviour, the outlet values of the flow variables with the exception of the axial velocity have no influence on the upstream calculation. To ensure that mass conservation is satisfied for the whole solution domain, the velocity imposed at the grid nodes in the outflow plane is deduced by adding to the adjacent upstream velocity an incremental value calculated from:

\[ U_{\text{inc}} = (\dot{m}_{\text{in}} - \rho U dA) / \rho dA \]  

(4.45)

where \( \dot{m}_{\text{in}} \) and \( \rho U dA \) represent the net mass inflow across the entry plane and that calculated at the outlet plane of the solution domain respectively. As the solution proceeds to convergence \( U_{\text{inc}} \) tends to zero as overall continuity from the solution domain is achieved. A similar procedure was applied to ensure overall conservation of heat and mass of fuel.

The logarithmic wall-functions, proposed by Patankar et al (1970), were used to overcome the problem of viscous effects close to the wall (where the turbulence model is invalid) and link the near-wall grid-line to wall boundaries. Thus:

For the flux momentum:

\[ (\rho U_p / \tau_w) \cdot C_{\mu}^{0.25} \cdot \rho^{0.5} \cdot k_p = (1/2) \ln(E \cdot Y_+) \]  

(4.46)

where \( Y_+ = Y_1 \cdot \rho \cdot k_p \cdot C_{\mu}^{0.25} / \nu \)
where subscripts $P$ and $w$ refer to the conditions near-wall node and the wall respectively. $Z$ and $E$ are the wall law constants and take the values of 0.419 and 9.8 respectively. The value of kinetic energy at point $P$, was calculated from the transport equation of $k$ with the flux energy to the solid wall set to zero. The corresponding value of dissipation rate, $\varepsilon$, was prescribed by:

$$\varepsilon_P = (0.5 \cdot k_P)^{1.5/Z} \cdot Y$$

(4.47)

For the heat flux:

$$C_P \frac{(T_P - T_w)}{\left(\frac{q''}{C_P}\right)} \cdot \frac{k_P}{Y} = \frac{\sigma_h}{Z} \{ E \cdot C_P \cdot k_P \cdot Y_1 \cdot P \} + P_j$$

(4.48)

where $P_j = \left( \frac{\sigma_h}{4 \sin \frac{\pi}{4}} \right) (A/k)^{0.5} \cdot \left( \frac{\sigma_h}{\sigma_L} - \frac{\sigma_h}{\sigma_L} \right)$,

and $T_w$ is the wall-temperature, $q''$ is the wall heat flux per unit area, $A$ is the Van Driest's constant (=26 for smooth walls) and $\sigma_L$ is the laminar Prandtl number. The temperature values at the boundary walls were taken from the experimental measurements, see subsection 3.3.

### 4.7.2 Inlet conditions

Due to the difficulties in the measurement of velocity profiles in the exit plane of the swirler, see subsection 3.3.3 of chapter 3, the inlet velocity profiles were obtained by extrapolation: figure 4.1 shows the resulting curves for the combustion air inlet velocity fitted to the measurements. The inlet
conditions for the spray were taken from the experimental data where available and estimated from suitable correlations where not. As indicated in subsection 3.3.1, the droplets emerging from the rotating cup atomizer had two diameters and their relative percentages depend on the rotational speed and the fuel mass flow rate. In this study the droplet diameters and their mass percentages were estimated from the data of Macfarlane et al (1969). The mean axial and tangential injection velocities of the droplets \( U_p \) and \( W_p \) were obtained from the experimental results. The droplet mean radial velocity component \( V_p \) was not measured and was, therefore, calculated from the following correlation of Fraster et al (1963):

\[
V_p = 0.304 \left[ 0.667 \rho Q^2 \cdot \frac{N^2 \cdot \sin \theta}{VD} \right]^{0.33} \quad \text{(m/s)}
\]

where; \( \dot{Q} \): Volumetric fuel flow rate (cu.ft/min)

\( N \): Rotating speed (rpm)

\( D \): Internal cup diameter (in)

\( \mu \): Dynamic viscosity (cp)

\( \theta \): Half-angle of divergence of cup lip

\( \rho \): Density of fuel (lb/cu.ft)

The average number of droplets associated with the mean and satellite droplets diameters was calculated from the expression:

\[
\hat{n}_j = \left( \frac{6m_j \cdot a_j}{D_j^3 \cdot \rho_p} \right)
\]

where the subscript \( j \) stands for the mean or satellite droplets and represents the percentage associated with each size.

Table 4.3 summarizes the inlet conditions for the two fuel sprays and the values of the main dependent variables are given in
4.8  Numerical solution procedure

The gas phase equations were represented in the general form of equation (4.9) and, incorporating the models for turbulence, combustion and radiation, were solved by a finite difference procedure through the TEACH computer program which has been reported by Gosman and Pun (197?). The solution of the first order differential equations which represent the fuel spray model were obtained numerically with the Runge-Kutta-4 method and the droplet source terms which represent the droplet-gas phase coupling were obtained through a droplet-tracking technique. The following two subsections describe briefly the numerical solution procedure for the gas and droplets.

4.8.1  Gas field

(i)  The grid system and the finite difference equation

The area of integration is shown in figure 4.2 and was overlayed by a rectangular, non-uniform grid, part of which is shown in figure 4.3a. The dependent variables $W$, $h$, $m_{f0}$, $m_{ox}$, $f$, $k$, $\epsilon$, $R_x$ and $R_y$ were evaluated at the nodes where the two sets of perpendicular grid lines intersect and the axial and radial velocities, $U$ and $V$, were obtained at nodes staggered with respect to the grid for the remaining variables as shown in figure 4.3b. This staggered grid system has the advantage that the variables $U$, $V$ and pressure, $P$, are stored such that the pressure gradients which derive the velocities $U$ and $V$ are readily evaluated. The approximate
solution to the partial differential equations for each dependent variable was found at every grid node by representing the spatial derivatives in the equations by finite difference approximations. The finite difference equations were obtained by integrating the respective differential equations over the control volume (cell) surrounding each grid node. Integration of equation (4.9) over the control volume, $V_p$, of a typical cell, see figure 4.3a, yields:

\[
\sum_{all b} \left[ \int (p U_j \phi - \Gamma \phi, j \frac{\partial \phi}{\partial x_j}) \, dA_b + \int \mathbf{S} \, dV \right] = 0 \tag{4.49}
\]

where the summation is over the four boundaries of the cell; $A_p$ is the area of the cell boundary in question and $U_j$ and $x_j$ represent the velocity and the coordinate normal to the boundary. Approximations are introduced for the four integrals of the first term, each of which represents the total transport across each of the cell boundaries by convection and diffusion, so as the flux for the west wall, $R_w$, for example, can be expressed as:

\[
R_w = (\rho U_A) \left[ f_w \phi_w + (1-f_w) \phi_p \right] \tag{4.50}
\]

where $f_w$ is the weighting factor defined according to the hybrid scheme, see Gosman et al (1976), as:

\[
f_w = 0.5[1 + (1/Pe_w)] \quad \text{for } -2 < Pe_w < 2
\]

\[
f_w = 1 \quad \text{for } Pe_w > 2
\]

\[
f_w = 0 \quad \text{for } Pe_w < -2
\]

$Pe_w$ is the Peclet number defined as $(\rho_w U_w) \delta x / \Gamma_w$, $\delta x$ is the
characteristic dimension of the cell in the velocity direction.

The integral of the second term is expressed in the linearized form:

$$\int_S \phi \, dV = S_u + S_p \phi_p$$

(4.51)

where values of $S_\phi$ are deduced from the original equation. This practice promotes stability when $S_\phi$ is itself a function of $\phi$ as indicated by Gosman et al (1974).

The resultant finite difference equation with allowance of the droplet-gas coupling, $S_d$, can be expressed as:

$$R_w - R_e + R_n - R_s = S_p \phi_p + S_u + S_d$$

(4.52)

and the finite difference equation for mass conservation is:

$$\sum_i G_i = S_{dm}$$

(4.53)

where $G = (\partial U A)$ and represents the four faces of the cell $w$, $e$, $s$ and $n$, and $S_{dm}$ is the droplet mass source term which will be described later.

From equations (4.52) and (4.53) the final general finite-difference equation has the form:

$$(A_p - S_p + S_{dm}) \phi_p = \sum_i A_i \phi_i + S_u + S_d$$

(4.54)

where $A_i$ is the finite difference coefficient expressed by the equation, in case of, for example, the cell face (e):

$$A_e = D_e - f e G_e$$
where $D_e$ is the diffusion coefficient which for hybrid-difference scheme is given by:

$$D_e = \max[D_e', -(1 - f_e), f_e G_e]$$

and $A_p = \sum_i A_i$

and $\sum_i$ denotes a summation over the four surrounding nodes.

(ii) Solution of the finite difference equation

The velocities and pressures are calculated by the iterative SIMPLE (for Semi Implicit Pressure Linked Equation) algorithm described by Caretto et al. (1972). This involves the solution of the momentum equations using prevailing pressure $P^*$ to yield a first estimate of the velocity field. There is no guarantee that the resultant velocity field, $U_j^*$, satisifies the finite difference equation (4.53) for continuity. Thus, if the correct velocities, $U_j$, and pressure, $P$, are defined as:

$$U_j = U_j^* + u_j' \quad \text{and} \quad P = P^* + p'$$  \hspace{1cm} (4.55)

and substituted into the momentum equation, a pressure correction equation will be obtained (for more detail see Geman et al 1972), $u_j'$ and $p'$ are the corrections.

The resulting equation can be solved and the values of pressures and velocities corrected. The general finite difference equation (4.54) is then solved successively for the remaining dependent variables and auxiliary quantities. The entire procedure is repeated until convergence is obtained. The convergence factor (residual) achieved for each equation was less than 1.2%. The droplet-gas coupling is incorporated in the numerical procedure as
follows. An isothermal droplet-free solution of the gas field is first obtained and the droplet mass, momentum and heat transfer equations integrated, using the Lagrangian approach, to yield the droplet trajectories and diameter and temperature history along each trajectory. The droplet source terms are calculated for each cell in the calculation domain, see the next subsection, and supplied to the finite-difference equation (4.54) to obtain adjusted values for the dependent variables which are used again in the solution of the droplet equations. This process of solving the gas field equations followed by the droplet equations is repeated until all equations are satisfied.

4.8.2 Droplet field

The Runge-Kutta-4 method was used to solve the differential equations, equations (4.27) to (4.29) and (4.35) (see section 4.5) for the droplets, and the coefficient of the droplet equations updated each time step of the calculation according to the position of the droplet in the flow field.

(i) Droplet-tracking technique

The droplet velocity components, \( U_p, V_p \) and \( W_p \), obtained from the numerical solution of the droplet equations were used through the tracking to determine the instantaneous locations of the droplets relative to the gas field as described in detail by El-Banhawy and Whitelaw (1980b) and Crowe et al (1977), from the following relations:

\[
\begin{align*}
\bar{x}_p &= \bar{x}_p + (U_p + U_e)(\Delta t/2) \\
&= \bar{x}_p + (U_p + U_e)(\Delta t/2) \\
\end{align*}
\]  

(4.56)
\[ y_{P_e} = y_{P_b} + (V_{P_e} + V_{P_b}) \cdot (\Delta t/2) \]  

(4.57)

where the subscripts \( b \) and \( e \) denote the beginning and end of the time increment respectively and the droplet properties at the end of the time-step are stored at \( x_{P_e} \) and \( y_{P_e} \). The time step was chosen so that it did not influence the results.

The treatment of the droplet impinging on the combustor walls is important in modelling spray flames. On impingement, droplets may form a liquid film, shatter, or reflect with reduced momentum, and the rules which govern the choice of these alternatives are not well formulated. Thus, a simple assumption has been made here, namely, if the droplet arrives at the combustor wall, the calculation proceeds by putting a negative sign for the radial velocity component and reducing the three velocity components by a factor accounting for the loss of momentum upon impingement.

(ii) Droplet interaction source terms

The tracking procedure allows the size, velocity and position of the droplets to be determined and the source terms are obtained by calculating the loss or gain of the droplet mass and momentum within each cell. Summing up the respective source terms for droplets representing different size ranges gives the total droplet-source terms.

The net rate of droplet mass for the cell is:

\[ S_{dm} = \sum_{L=1}^{N} [(m_{P_{L_{1}}} - m_{P_{L_{0}}}) \cdot \mathbf{n}]_L \]  

(4.58)

where \( m_p \) is the droplet mass \((=\frac{\pi}{6} D_p^3 \rho_p)\), \( N \) is the total number of size
ranges representing the spray, \( n \) is the number of the droplets issuing from the spray nozzle per second and having a diameter corresponding to the size range, \( L \), and subscripts \( i \) and \( o \) refer to the entry and exit of the cell.

The net enthalpy flux for the cell is:

\[
S_{\text{dh}} = \sum_{L=1}^{N} \left[ m_{p_{i}} - m_{p_{o}} \right] \dot{n}_{L} \left( H_{f} - L \right)
\]

(4.59)

The net rate of axial momentum for the cell is:

\[
S_{\text{dU}} = \sum_{L=1}^{N} \left[ \left( \left( mV \right)_{p_{o}} - \left( mV \right)_{p_{i}} \right) \dot{n}_{L} \right]
\]

(4.60)

Similarly, radial momentum:

\[
S_{\text{dV}} = \sum_{L=1}^{N} \left[ \left( \left( mV \right)_{p_{o}} - \left( mV \right)_{p_{i}} \right) \dot{n}_{L} \right]
\]

(4.61)

and tangential momentum:

\[
S_{\text{dW}} = \sum_{L=1}^{N} \left[ \left( \left( mW \right)_{p_{o}} - \left( mW \right)_{p_{i}} \right) \dot{n}_{L} \right]
\]

(4.62)

4.9 Presentation and discussion of results

In this section results calculated with the procedure described above are described and compared with the experimental data reported in chapter 3, section 3.3. The comparison comprises the isothermal velocity measurements obtained with the swirl number flow of 1.4, run 1 of table 3.3, and measurements for two flames with 54 and 115 \( \mu \)m spray mean-droplet diameter; see table 3.3, runs 4 and 6. The comparison between the measured isothermal velocity and its
prediction allows an assessment of the accuracy of the numerical procedure in the absence of complications and uncertainties associated with the combustion and spray models. The following subsection describes the results related to the isothermal flow and in the next subsection the combusting flow results are presented and discussed.

4.9.1 Isothermal flow results

The calculated results were obtained with three different grid arrangements comprising 22x25, 28x32 and 32x38 grid nodes and the distributions shown in figure 4.4. The number of nodes was increased in the initial region of the calculation domain, where the gradients of the dependent variables are more likely to be large and to reduce the cell Peclet number ($pU\delta x/\mu$) and, therefore, the possibility of numerical errors.

The experimental and predicted radial profiles of mean axial and tangential velocities with the three grid distributions at the conditions of run 1 are shown in figure 4.5 and 4.6 respectively. In general, the predicted velocity is in reasonable accord with the experimental data except in the region immediately downstream of the swirler exit as indicated by the radial profile at $X = 25$ mm. The discrepancies between the experimental and predicted results are concentrated in the region of high velocities and indicate differences of around 2 - 4 m/s with the lower values associated with the fine grid arrangement. It is, also, shown that the discrepancies between the predicted results of the different grids are in the order of 1 - 1.5 m/s with a higher decay of the predicted maximum velocity as the grid nodes decreases. The observation about the effect of the
grid dimensions may be partly attributed to the numerical diffusion associated with the upwind difference scheme in the calculation of turbulent recirculating flows, see McGuirk, Taylor and Whitelaw (1981), Taylor (1981) and Raithby (1976), since cell Peclet numbers in this region were in the range from 12 to 20. It is, also, likely that the indicated differences between the experimental and predicted results are influenced by uncertainties in the turbulence model associated with the isotropic-viscosity hypothesis in the calculation of highly swirling flows as shown by Habib and Whitelaw (1980) and Taylor and Whitelaw (1983). At downstream stations, \( X > 25 \text{ mm} \), the predicted results show reasonable agreement with the experimental data, apart from the central part of the flow, \( R < 12 \text{ mm} \), with discrepancies between 5 and 15\%. The width of the recirculation zone is well approximated and the increased discrepancies around the centreline probably stem from the inadequate representation of the atomizer cooling air inlet velocity.

The 22x25 grid was adopted in the combusting flow calculation to be described in the following subsection for reasons of economy and available computer storage. Figure 4.5 shows that the discrepancies are acceptably small which, in view of the additional uncertainties which will be introduced by the spray and combustion models, appear to be the best compromise.

4.9.2 Combusting flow results

(i) The flame with the 54 \( \mu \text{m} \) droplet diameter, first test case

The measurements for this case corresponds to the operating conditions of run 4 in table 3.3, see also table 4.3. The calculated
and measured radial profiles of temperature and species mass fraction of carbon dioxide (CO₂), oxygen (O₂) and unburned hydrocarbons (UHC) at five axial stations along the combustor of X = 25, 50, 100, 150 and 200 mm are shown in figures 4.7 to 4.11 respectively. The corresponding profiles of the measured mass fraction of carbon monoxide (CO) are also shown for reference purposes. The general features of the temperature and species concentrations are reasonably predicted but discrepancies in the extent of the chemical reaction and the corresponding temperature values are clearly present. At locations close to the combustor wall and close to the centreline, the temperature and CO₂ concentrations are under-predicted by up to 100 °C and 10% respectively, with consequent over-prediction of O₂ concentrations, where discrepancies of around 7% can be observed. These discrepancies may be attributed to deficiencies in the coupled turbulence/chemistry models and, to a less extent, to the boundary conditions specifications. Close to the fuel injection and at small radii, X = 25 mm and R < 50 mm, the temperature values are over-predicted by up to 500 °C due mainly to uncertainties in the combustion model and, in particular, to the assumption of single step chemical reaction and the neglect of soot formation and intermediate species such as carbon monoxide. Although the O₂ and UHC concentrations are, in their general behaviour, correctly predicted, the CO₂ concentrations are over-estimated by around 40% and can be explained by referring to the high CO value of around 9% measured in this zone. In effect, and due to the comparatively high calorific value of CO, the calculated heat release rates are much higher than the actual values. It can also be seen that, at radii corresponding to the maximum gradient in the measured temperature and species concentrations, around R = 70 mm, the predicted profiles of CO₂ and
O₂ are shifted towards smaller radii which indicates an underestimation of chemical reaction rates. With presumed dependence of the reaction rate on the turbulent kinetic energy and its dissipation rate, this result is consistent with the under-prediction of the velocity at this region as evident in figure 4.5.

At downstream stations, X > 25 mm, the discrepancies between the calculated and measured temperature values are less than 100 °C, and up to 200 °C at X = 50 mm, with corresponding overestimation of CO₂ mixture fraction by 1 to 2% and under-estimation of O₂ by 2% and again probably due to deficiencies in the coupled turbulence/chemistry models and to uncertainties of the combustion model (see the corresponding measured CO mixture fraction).

Figure 4.12 shows a plot of the trajectories for the droplets representing the spray. The arrows indicate the positions where the droplet diameter reached a specified value and it is clear that the trajectories for the satellite droplets are short (~10 mm) due to their much smaller diameter and relatively high evaporation rate. On the other hand, the trajectories of the main droplets indicate that the spray completes evaporation at X less than 70 mm. It is likely that the rapid evaporation of the droplets contributes to some extent to the discrepancies of temperature and CO₂ in the upstream region.

(ii) The flame with the 115 μm droplet diameter, second test case

The predicted and measured radial profiles of temperature, mixture fraction of CO₂, O₂ and UHC for this flame are shown in figures 4.13 through 4.17 for five axial locations. The position of
the stations and the operating conditions of the flame are the same as those for the first case except that the spray mean-droplet diameter has increased to 115 µm. In general, the predicted results are again in qualitative agreement with measurements. However, in the vicinity of the centreline and in the upstream region, \(X < 25\) mm, the temperatures are over-predicted by up to \(500\) °C with corresponding over-estimation of \(CO_2\) mixture fraction. On the other hand, at downstream sections and close to the centreline and especially for \(X \geq 100\) mm, the calculated temperatures are lower than the measurement by \(100 - 200\) °C and related discrepancies in the mass fractions of \(O_2\) (2 to 4%) and \(CO_2\) (1 to 3%) can be easily detected. It is also shown at downstream stations, \(40 < R < 80\) mm, that the over-prediction of the temperature by up to \(300\) °C is associated with differences in \(CO_2\) and \(O_2\) mass fractions of around 3%. The over-prediction of temperature in the upstream region and at small radii can be attributed mainly to the neglect of \(CO\) formation in the present procedure, as for the first case. In confirmation with this support, the \(CO\) mass fraction in this region demonstrates values of 8% as shown in figure 4.13. The over-prediction of \(UHC\) at \(X < 50\) mm are probably referring to the significant amount of carbon particles associated with this flame, see subsection 3.3.4, which was neglected in the calculations. The differences observed at downstream sections can be related to the expression used to calculate the droplet evaporation. The trajectories for the droplets used to represent the spray are shown in figure 3.18 and again the arrows indicate the positions where the droplet diameter has reached to a certain value. It is evident that, for the 115 µm droplets, the rate of change of droplet diameter is small in the initial part of the trajectory and that the droplet evaporation continues in the downstream region, up
to $X = 160$ mm, with a concentration of the evaporation in the region of $R$ between 75 and 90 mm. The initially slow evaporation at large radii of the combustor is the likely cause for the under-prediction of temperature and $\text{CO}_2$ concentration close to the centreline, at $X > 100$ mm, shown in figures 4.15 to 4.17. As a consequence, smaller amounts of fuel are transported to the central part of the flame and leads to an under-prediction of the chemical reaction rate. However, most of the fuel has evaporated at large radii and gives rise to a higher reaction rate compared to the experiments. The significant under-prediction of the temperature and $\text{CO}_2$ and related over-estimation of $\text{O}_2$ at the near-wall far stream of 150 mm are probably due to smaller values of fuel vapour convected to this region as a result of the high velocity gradient towards small radii as indicated in figure 4.5. It is unlikely that the results for the flame of the first case were influenced by the same deficiencies due to the much smaller diameters involved in that flame, which evaporate at a higher rate, and the uncertainties in the calculation of the evaporation rate would have a relatively small influence on the overall flame properties.

4.10 Concluding remarks

In the preceding sections of this chapter, a calculation procedure has been presented and applied to the prediction of the flow properties for the axisymmetric combustor described in chapter 3 with the flow of 1.4 swirl number. The predictions were made for the isothermal flow velocity and the performance of the two flames having 54 and 115 $\mu$m spray mean-droplet diameters and the results were compared with the corresponding experimental data. The comparisons have indicated that the general features of the flow fields are
correctly predicted. The extent of the agreement between the experimental and predicted results has showed the relative importance of the sources of uncertainties associated with the modelling assumptions and the spray characteristics. For example, inspection of the predicted isothermal velocity profiles has indicated that the observed discrepancies are related to the effective viscosity hypothesis assumption of the turbulence model adopted in the present work. For the combusting flows the results of the big droplet diameter flame have showed larger discrepancies especially in the downstream region with differences in the temperature and CO₂ and O₂ mixture fractions up to 300 °C, 3% and 4% respectively. These have been attributed mainly to the uncertainties associated with the expression used to calculate the droplet evaporation rate and the deficiencies of the coupled turbulence/combustion models.
<table>
<thead>
<tr>
<th>Variable (Φ)</th>
<th>( \Gamma_\Phi )</th>
<th>( S_\Phi )</th>
<th>( S_d )</th>
</tr>
</thead>
<tbody>
<tr>
<td>U</td>
<td>( \mu \text{eff} )</td>
<td>( \frac{\partial}{\partial x} (\mu \text{eff} \frac{\partial U}{\partial x}) + \frac{1}{r} \frac{\partial}{\partial r} (\mu \text{eff} \frac{\partial V}{\partial r}) - \frac{\partial P}{\partial x} )</td>
<td>( - \sum_{L=1}^{N} \left{ ((mU) - (mU)_L) \cdot \hat{n}_L \right} )</td>
</tr>
<tr>
<td>V</td>
<td>( \mu \text{eff} )</td>
<td>( \frac{\partial}{\partial x} (\mu \text{eff} \frac{\partial U}{\partial x}) + \frac{1}{r} \frac{\partial}{\partial r} (\mu \text{eff} \frac{\partial V}{\partial r}) - \frac{2\mu V}{r^2} + \frac{8\omega^2}{r} - \frac{\partial P}{\partial r} )</td>
<td>( - \sum_{L=1}^{N} \left{ ((mV) - (mV)_L) \cdot \hat{n}_L \right} )</td>
</tr>
<tr>
<td>W</td>
<td>( \mu \text{eff} )</td>
<td>( - \left{ \frac{\mu \text{eff}}{r^2} + \frac{\rho V}{r} + \frac{1}{r} \frac{\partial \mu \text{eff}}{\partial r} \right} W )</td>
<td>( - \sum_{L=1}^{N} \left{ ((mW) - (mW)_L) \cdot \hat{n}_L \right} )</td>
</tr>
<tr>
<td>k</td>
<td>( \frac{\mu \text{eff}}{\sigma_k} )</td>
<td>( \mu \text{eff} \cdot \mathbf{G} - \rho \varepsilon )</td>
<td>0</td>
</tr>
<tr>
<td>ε</td>
<td>( \frac{\mu \text{eff}}{\sigma_\varepsilon} )</td>
<td>( \frac{\varepsilon}{k} \left{ \mu \text{eff} \mathbf{G}^* - C_\varepsilon \rho \varepsilon \right} )</td>
<td>0</td>
</tr>
<tr>
<td>f</td>
<td>( \frac{\mu \text{eff}}{\sigma_f} )</td>
<td>0</td>
<td>( + \sum_{L=1}^{N} \left{ (m - m_f)_L \right} \cdot \hat{n}_L )</td>
</tr>
<tr>
<td>h</td>
<td>( \frac{\mu \text{eff}}{\sigma_h} )</td>
<td>( \frac{16}{9} \cdot \mathbf{g} \cdot (R_x + R_y - 2\sigma^2) )</td>
<td>( + \sum_{L=1}^{N} \left{ (m - m_h)<em>L \right} \cdot (H</em>{fu} - L) )</td>
</tr>
<tr>
<td>m_{fu}</td>
<td>( \frac{\mu \text{eff}}{\sigma_{m_{fu}}} )</td>
<td>( 23.6 \left{ \frac{\mathbf{v} \cdot \varepsilon}{k^2} \right} \cdot \frac{\varepsilon}{k} \cdot \bar{m}_{\min} \cdot \bar{P} )</td>
<td>( + \sum_{L=1}^{N} \left{ (m - m_{m_{fu}})_L \right} \cdot \hat{n}_L )</td>
</tr>
</tbody>
</table>

\( G = 2 \left( \frac{\partial U}{\partial x} \right)^2 + \left( \frac{\partial V}{\partial r} \right)^2 + \left( \frac{\partial V}{\partial r} \right)^2 + \left( \frac{\partial W}{\partial x} \right)^2 + \left( \frac{\partial W}{\partial r} \right)^2 + \left( \frac{\partial U}{\partial r} + \frac{\partial V}{\partial x} \right)^2 \), \( \text{NB All constants are shown in table 4.2} \)
<table>
<thead>
<tr>
<th>Constant</th>
<th>Value</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>$c_\mu$</td>
<td>0.09</td>
<td></td>
</tr>
<tr>
<td>$c_1$</td>
<td>1.44</td>
<td>Turbulence model constants</td>
</tr>
<tr>
<td>$c_2$</td>
<td>1.92</td>
<td></td>
</tr>
<tr>
<td>$\sigma_k$</td>
<td>1.0</td>
<td></td>
</tr>
<tr>
<td>$\sigma_\varepsilon$</td>
<td>1.22</td>
<td>Effective turbulent Prandtl numbers</td>
</tr>
<tr>
<td>$\sigma_f$</td>
<td>0.9</td>
<td></td>
</tr>
<tr>
<td>$\sigma_{mfu}$</td>
<td>0.9</td>
<td></td>
</tr>
<tr>
<td>$\sigma_h$</td>
<td>0.9</td>
<td></td>
</tr>
<tr>
<td>$z$</td>
<td>0.419</td>
<td>Logarithmic law of the wall constants</td>
</tr>
<tr>
<td>$E$</td>
<td>9.8</td>
<td></td>
</tr>
</tbody>
</table>
### TABLE (4.3) SPRAYS INLET CONDITIONS

<table>
<thead>
<tr>
<th>Condition</th>
<th>First test case, run 4 (table 3.3)</th>
<th>Second test case, run 6 (table 3.3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main droplet diameter</td>
<td>54 µm</td>
<td>115 µm</td>
</tr>
<tr>
<td>Satellite droplet diameter</td>
<td>30 µm</td>
<td>60 µm</td>
</tr>
<tr>
<td>% Mass main droplets</td>
<td>80 %</td>
<td>75 %</td>
</tr>
<tr>
<td>% Mass satellite droplets</td>
<td>20 %</td>
<td>25 %</td>
</tr>
<tr>
<td>Mean axial velocity of droplets</td>
<td>5.5 - 10.5 (m/s)</td>
<td>3 - 9.5 (m/s)</td>
</tr>
<tr>
<td>Mean radial velocity of droplets</td>
<td>6.5 - 10.2 (m/s)</td>
<td>5 - 8.25 (m/s)</td>
</tr>
<tr>
<td>Mean tangential velocity of droplets</td>
<td>6.5 - 9.1 (m/s)</td>
<td>5.3 - 6.2 (m/s)</td>
</tr>
</tbody>
</table>
TABLE (4.4) INLET CONDITIONS FOR THE MAIN DEPENDENT VARIABLES
FOR THE TWO FLAMES INVESTIGATED

<table>
<thead>
<tr>
<th>Variable</th>
<th>Inlet condition</th>
</tr>
</thead>
<tbody>
<tr>
<td>$U$</td>
<td>Figure (4.3a)</td>
</tr>
<tr>
<td>$V$</td>
<td>0.0</td>
</tr>
<tr>
<td>$W$</td>
<td>Figure (4.3b)</td>
</tr>
<tr>
<td>$k$</td>
<td>$0.01 \times U_{av}^2$</td>
</tr>
<tr>
<td>$\varepsilon$</td>
<td>$C_{\mu}k^{1.5}/\lambda$</td>
</tr>
<tr>
<td>$m_{ox}$</td>
<td>0.232</td>
</tr>
<tr>
<td>$m_{fu}$</td>
<td>0.0</td>
</tr>
<tr>
<td>$h$</td>
<td>$C_p(T - T_{ref})$</td>
</tr>
<tr>
<td>$f$</td>
<td>0.0</td>
</tr>
<tr>
<td>$T_{air}(^\circK)$</td>
<td>308</td>
</tr>
</tbody>
</table>

$U_{av}$ : Average inlet mean axial velocity component.
$\lambda$ : Characteristic length scale.
Fig. 4.1. Inlet radial profiles of velocity components.

Fig. 4.2. Area of integration of the finite difference equations.
Fig. 4.3a. Rectangular, non-uniform grid showing the locations of scalar variables.

Fig. 4.3b. Relative locations of variables U and V in staggered grid.
Fig. 4.4. The three different grid distributions used for the isothermal flow calculation.
Fig. 4.5. Measured and calculated radial profiles of mean axial velocity (isothermal flow)
Fig. 4.5. Continued.
Fig. 4.6. Measured and calculated radial profiles of mean tangential velocity (isothermal flow)
Fig. 4.7. Experimental and predicted radial profiles of temperature and species concentrations, $X = 25$ mm, first test case.

- - - - - Experimental data.  - - - - - Predictions.
Fig. 4.8. Experimental and predicted radial profiles of temperature and species concentrations, $X = 50$ mm, first test case.
Fig. 4.10. Experimental and predicted radial profiles of temperature and species concentrations, X = 150 mm, first test case.
Fig. 4.10. Experimental and predicted radial profiles of temperature and species concentrations, $X = 150$ mm, first test case.
Fig. 4.11. Experimental and predicted radial profiles of temperature and species concentrations, X = 200 mm, first test case.
Fig. 4.12. Trajectories of the droplets, first test case.
Fig. 4.10. Experimental and predicted radial profiles of temperature and species concentrations, $X = 150$ mm, first test case.

Fig. 4.13. Experimental and predicted radial profiles of temperature and species concentrations, $X = 25$ mm, second test case.
Fig. 4.14. Experimental and predicted radial profiles of temperature and species concentrations, $X = 50$ mm, second test case.
Fig. 4.15. Experimental and predicted radial profiles of temperature and species concentrations, $X = 100$ mm, second test case.
Fig. 4.16. Experimental and predicted radial profiles of temperature and species concentrations, $X = 150$ mm, second test case.
Fig. 4.17. Experimental and predicted radial profiles of temperature and species concentrations, X = 200 mm, second test case.
Fig. 4.18. Trajectories of the droplets, second test case.
CHAPTER 5

CLOSURE

A summary of the main conclusions of the present investigations is given in the first section of this chapter. Section 5.2 presents recommendations for further work.

5.1 Achievements and conclusions

Measurements of droplet velocity and number-density, gas temperature, species concentrations and wall temperature were made in a confined and unconfined kerosene spray flames for a wide range of operating conditions. In additions, the isothermal velocity field corresponding to the confined flame measurements have been described. The results present new information which aids the understanding of the complex phenomena controlling the spray combustion and evaluation of the calculation methods.

The velocity and droplet number-density, gas temperature and species concentrations were measured by laser Doppler velocimetry, 80 µm diameter bare-wire thermocouples and a water-cooled sampling probe respectively. The temperatures of the unconfined flames were also obtained by a suction pyrometer and bare-wire thermocouples of 40 and 300 µm diameters. Comparison of the results obtained by the different bare-wire thermocouples and the suction pyrometer assisted the assessment of the precision of the results in kerosene spray flames. The comparison indicated that the fine wire thermocouple measures higher values of near gas temperature with an accuracy of within ±2% while the large thermocouple and the
suction pyrometer record mean temperatures represented by: \( \bar{h}_c T_g / h_c \). The investigation of the influences of the sample quenching, suction velocity and the probe orientation on the gas sample collected by the probe have indicated that the measured species concentrations are density-weighted averaged and have an accuracy within \( \pm 9\% \).

The results of the unconfined kerosene spray flames, where the spray was generated by a twin-fluid atomizer, have shown the importance of the droplet diameter on the spray evaporation and consequently on the combustion intensity and reaction rate. The measurements showed that the reduction of the droplet diameter, due to higher atomization air/fuel ratio, leads to earlier combustion and shorter flame.

The confined flame measurements were performed in a water-cooled cylindrical combustor and the kerosene fuel was injected from a rotating cup atomizer producing symmetrical spray with near monodisperse controlled droplet size. Measurements of the isothermal axial velocity within the combustor, without fuel injection, were achieved for swirl numbers of 1.4, 0.81 and 0.47. The results indicated that the decrease of the swirl number results in reduction in the size and strength of the main recirculation zone.

The influence of the spray mean-droplet diameter on the combustion performance and flame structure has been examined with the three levels of the air swirl mentioned above. For high swirling flames, the results indicated that the increase of the droplet diameter is associated with longer droplet life time and results in significant droplet penetration with consequent alteration of the flame structure. The results also showed that the reduction in the evaporation rate with increase of the droplet diameter caused
reduction in the fuel-air mixing and chemical reaction rates, especially in the main flame region. Although the low swirl flame structure was different, the results indicated the same trend of the reduction in the combustion intensity with increase of the droplet diameter.

The flames with swirl numbers of 1.4 and 0.81 were stabilized on the off-centreline recirculation zone and the reduction of the swirl results in reduced chemical reaction rates within the main-flame region and reduction of the shear layer size. However, chemical reactions continued within the central and wall zones in the upstream region. The results obtained with the 0.47 swirl number showed that the main reaction zones were concentrated in the near-wall region where flame stabilization was achieved. Further reduction in the combustion intensity and carbon burn-out throughout the wall region of the combustor were observed and were caused by the simultaneous reduction in the evaporation rate and turbulent mixing.

With two combustion air preheat temperatures, field measurements of droplet velocity, gas temperature and species concentrations for 54 and 115 µm droplet diameter sprays with 1.4 swirl number have shown increase of the temperature levels throughout the combustor and the onset of combustion nearer to the swirler exit. The results have also indicated that the larger droplet diameter flame is more influenced by the increase of the inlet air-temperature than the smaller one. The combustion of the smaller droplets was very similar to that of turbulent gas diffusion flames.

A calculation procedure has been described and applied to the prediction of local flame properties of the confined spray flame. The method solves the elliptic differential equations representing...
conservation of mass, momentum, enthalpy and species concentrations in finite difference form. In addition, Lagrangian equations for the droplet motion and thermal balance were solved for the droplet size ranges representing the fuel spray. A two-equation turbulence model, a combustion model based on mean mass fraction of the species and a four flux radiation model were incorporated. The spray model assumes that the evaporating droplets act as distributed point sources of fuel vapour. Comparison between the experimental and predicted results for two flames indicated that the general features are reasonably predicted, but discrepancies in the extent of the chemical reaction and temperature were observed, particularly with the larger droplet size flame. These discrepancies have been attributed mainly to the uncertainties associated with the expression used to calculate the spray evaporation rate and the deficiencies of the coupled turbulence/combustion models.

5.2 Suggestions for further work

The present study has led to suggestions for the topics where work is required to aid further understanding of the various processes controlling spray combustion and the development and validation of related models. These are summarized below.

1. The operating conditions for the present study covered a range of combustion air swirl, droplet diameter and air preheat for different kerosene spray flame geometries. An extension of the operating conditions to those more typical of practical combustors is recommended; they should include poly-disperse sprays, larger mean droplet diameters, variable air-fuel load and higher air preheat temperatures.
2. In the discussion of the confined flame results, section 3.3, it was argued that the results reveal strong similarities with the characteristic time model, developed from study of simplified bluff-body stabilized flames, for spark ignition and flame stabilization, gaseous emission and combustion efficiency. The application of the model to swirl stabilized flames is desirable to modify the model and include explicitly the effects of the swirler. Measurements in addition to those of section 3.3 should be obtained for operating conditions near the lean blowoff equivalence ratio.

3. The predicted results implied that the initial conditions for the spray need to be better specified to allow accurate predictions for the spray combustion properties. Thus, it is desirable to attempt to relate the geometry and flow conditions of pressure jet and air blast atomizers to the flow properties at a location close to point downstream of the exit plane. This is very important for the prediction of large droplet diameter flames, i.e. $D_m > 50 \, \mu m$, where the spray characteristics have a significant influence on the combustion behaviour.

4. The droplet evaporation rate used in the present spray model is calculated from an expression based on single, isolated evaporating droplet in the absence of chemical reaction. More realistic expressions, which consider the influence of the interaction between adjacent droplets and chemical reaction on the evaporation characteristics, are required. This can probably be achieved by performing further experimental studies concerning the combustion of simplified droplet
streams, see Sangiovanni et al (1979, 1982), together with theoretical analysis of the type described by Labowsky (1980) and Correa et al (1982).

5. The combustion model used in the present study neglects the intermediate species and the influence of chemical kinetics on the reaction rate. This contributes to the uncertainties associated with the predicted results of chapter 4. Attempts to predict intermediate species, such as CO, were performed by Jones (1980), Jones et al (1979, 1980) and Ganesan et al (1979) but with large discrepancies between calculated and measured results. The model also neglects the fluctuations of the scalar variables and this contributes to the inaccuracy. An alternative approach, based on a probability density function for mixture fraction, see for example Jones and Whitelaw (1982), can be used to overcome this problem. Comparison of results with experimental data, see Jones (1980) and Jones et al (1979, 1980), indicate that the model is likely to be adequate as far as the calculation of the major species, temperature and heat release is concerned. However, the prediction of pollutant species showed large discrepancies relating to the assumed probability density function used, and this suggests the need for further measurements of single and joint probability density functions. Further work is, therefore, recommended to use and develop the p.d.f. approach with measured probability density functions and adequate reaction rates.
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APPENDIX I

GEOMETRIC DETAILS OF THE SWIRLERS

The geometrical details of the three annular guide vane cascade swirlers used in the confined flow experiments of chapter 3, section 3.3, are shown in figure AI.1. Each swirler has ten blades of 2 mm thickness inclined 60, 45 and 30 degrees to the main flow direction. The corresponding swirl numbers were calculated approximately from the diameter ratio and the vane angle were found 1.4, 0.81 and 0.47 respectively.
Fig. AI.1. Geometrical details of the 3 combustion air swirlers used in the axi-symmetric combustor experiments.
APPENDIX II

REFERENCE EXPERIMENTAL DATA OF THE CONFINED FLAME EXPERIMENTS

The results of temperature and species concentrations of \( \text{CO}_2, \text{CO}, \text{O}_2 \) and UHC for the confined flames of chapter 3, subsection 3.3.4, are presented in this appendix in the form of radial profiles for the ten axial locations of measurements. The position of the different sections relative to the swirler exit together with their corresponding notations of plotting are shown in table AII.1. These profiles serve as a reference to the contour plots presented in subsection 3.3.4 and allow the reader to extract the experimental data of interest with much ease. Figures AII.1 through AII.47 show the radial distributions of temperature and species concentrations of the flames 4 to 16 of table 3.3 respectively.

TABLE (AII.1)

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<td>□</td>
<td>■</td>
<td>△</td>
<td>▼</td>
</tr>
</tbody>
</table>

X : Axial distance measured from the swirler exit.
Fig. AII.1 Radial profiles of mean gas temperature, run 4.
Fig. AII.2. Radial profiles of CO$_2$ volume concentration, run 4.
Fig. AII.3. Radial profiles of CO volume concentrations, run 4.
Fig.AIL.4. Radial profiles of $O_2$ volume concentration, run 4.
Fig. AII.5. Radial profiles of UHC volume concentration, run 4.
Fig.AII.6. Radial profiles of mean gas temperature, run 5.
Fig.AII.7. Radial profiles of CO$_2$ volume concentration, run 5.
Fig. AII.8. Radial profiles of CO volume concentration, run 5.
Fig. AII.9. Radial profiles of $O_2$ volume concentration, run 5.
Fig. AII.10. Radial profiles of UHC volume concentration, run 5.
Fig. AII.11. Radial profiles of mean gas temperature, run 6.
Fig. AII.12. Radial profiles of CO₂ volume concentration, run 6.
Fig.A11.13. Radial profiles of CO volume concentration, run 6.
Fig.AII.14. Radial profiles of $O_2$ volume concentration, run 6.
Fig. AII.15. Radial profiles of UHC volume concentration, run 6.
Fig.AII.16. Radial profiles of mean gas temperature, run 7.
Fig. AII.17. Radial profiles of CO$_2$ volume concentration, run 7.
Fig. AII.18. Radial profiles of CO volume concentration, run 7.
Fig. AII.19. Radial profiles of $O_2$ volume concentration, run 7.
Fig.AII.20. Radial profiles of mean gas temperature, run 8.
Fig. AII.21. Radial profiles of mean gas temperature, run 9.
Fig. AII.22. Radial profiles of mean gas temperature, run 10.
Fig.AII.23. Radial profiles of CO$_2$ volume concentration, run 10.
Fig.AII.24. Radial profiles of CO volume concentration, run 10.
Fig.AII.25. Radial profiles of $O_2$ volume concentration, run 10.
Fig. AII.26. Radial profiles of mean gas temperature, run 11.
Fig.AII.27. Radial profiles of mean gas temperature, run 12.
Fig.AII.28. Radial profiles of mean gas temperature, run 13.
Fig.AII.29. Radial profiles of CO$_2$ volume concentration, run 13.
Fig.AII.30. Radial profiles of CO volume concentration, run 13.
Fig. AIII.31. Radial profiles of $O_2$ volume concentration, run 13.
Fig.AII.32. Radial profiles of UHC volume concentration, run 13.
Fig. AIII.33. Radial profiles of mean gas temperature, run 14.
Fig. AII.34. Radial profiles of CO$_2$ volume concentration, run 14.
Fig. AII.35. Radial profiles of CO volume concentration, run 14.
Fig.AII.36. Radial profiles of $O_2$ volume concentration, run 14
Fig.AII.37. Radial profiles of UHC volume concentration, run 14.
Fig.AII.38. Radial profiles of mean gas temperature, run 15.
Fig.AII.39. Radial profiles of CO₂ volume concentration, run 15.
Fig.AII.40. Radial profiles of CO volume concentration, run 15.
Fig.AII.41. Radial profiles of O₂ volume concentration, run 15.
Fig. AII.42. Radial profiles of UHC column concentration, run 15.
Fig.AII.43. Radial profiles of mean gas temperature, run 16
Fig.AII.44. Radial profiles of CO$_2$ volume concentration, run 16.
Fig.AII.45. Radial profiles of CO volume concentration, run 16.
Fig.AII.46. Radial profiles of $O_2$ volume concentration, run 16.
Fig. AII.47. Radial profiles of UHC volume concentration, run 16.