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VALIDATION OF NUMERICAL METHODS AT A CONFINED TURBULENT NATURAL GAS DIFFUSION FLAME CONSIDERING DETAILED RADIATIVE TRANSFER

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ABSTRACT

Radiation heat transfer in flames depends strongly on local quantities such as pressure, temperature and concentration of participating species. In the present study, 3D numerical calculations of radiative heat transfer together with the reacting flow field are compared to detailed measurements of the velocity, temperature and spectral radiation field of a model combustor.

The geometry of the combustion chamber (d_{ch}=0.5m), the flame configuration (type-II swirling diffusion flame) and the highly turbulent flow conditions resemble the characteristics of industrial combustors.

The concentrations of CO_2, H_2O, CO, CH_4, NO, NO_x, O_2 and H_2 as well as local mean temperatures and their fluctuations were recorded at 300 locations at 14 axial planes. The radiation intensity incident on the wall was measured spectrally and time resolved at 11 axial planes within the spectral range of 1.4 to 5.4 μm.

For numerically solving the reacting flow field, spectral methods for calculating the radiative heat transfer were coupled to fluid mechanical methods for calculating the reacting flow.

The agreement between numerical prediction and measurements for the reacting flow field as well as for the radiative heat transfer is reasonably good. The numerical computations show that radiative transfer is of major importance. The temperature in the hot reaction zone was found to be lowered by approximately 400 K by radiative losses.

NOMENCLATURE

Symbol	Unit	Description
d	[μm, m]	diameter
D	[mm]	burner orifice diameter
f	[Hz]	frequency
p	[bar]	pressure
P	[kW]	power
S	[-]	swirl number
T	[K, °C]	temperature
z	[m]	axial distance
λ	[-]	fuel air ratio
λ	[μm]	wavelength

Subscript	Description
ch	combustion chamber
i	inner
o	outer
pearl	thermocouple pearl
Probe	measurement probe
th	thermal
w	wall
0,th	theoretical

1. INTRODUCTION

Because of excellent mixing, short flamelength, good stability and the potential of reducing emissions, swirl flames are widely used in combustion engineering applications like gas turbines, industrial furnaces as well as boilers of power plants.

However, the prediction of turbulent swirling flames by numerical methods is still not always satisfying. In order to validate and optimize the modeling of turbulent swirl flames, the research council TECFLAM has defined a model combustor of industrial scale which is characterized by good accessibility. The flame is of the type-II swirl type (Leuckel, 1971).

Recently, extensive measurements of concentration and temperature profiles of the flame have been carried out at the Engler-Bunte-Institut (EBI) and subsequent to this investigation, radiative

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heat transfer has been studied by the Institut für Thermische Strömungsmaschinen (ITS).

In this paper, the experimental data are compared to detailed numerical predictions taking into account radiative heat transfer. The goal of the present study is twofold. On one hand, the efficiency and capability of different numerical models for predicting reacting turbulent flows is to be evaluated. The other goal is to assess the role of radiative heat transfer. Consequently, in particular turbulence modeling and radiative transfer will be discussed.

2. EXPERIMENTAL SETUP

2.1 THE COMBUSTION CHAMBER AND MEASUREMENT TECHNIQUES

The burner is located at the bottom of the cylindrical combustion chamber. It consists of a variable swirl generator with movable-block-design (Leuckel, 1967) by which the swirl number can be adjusted over a wide range. The burner is moved axially for traversing the flame in axial direction (Fig. 2.1). The combustion air passes the swirl generator and is then fed through a concentric orifice (D=60mm) into the combustion chamber. Natural gas is injected centrally by an axial jet (outer diameter D_o =26mm, inner diameter D_i =20mm). The walls of the combustor are cooled by water.

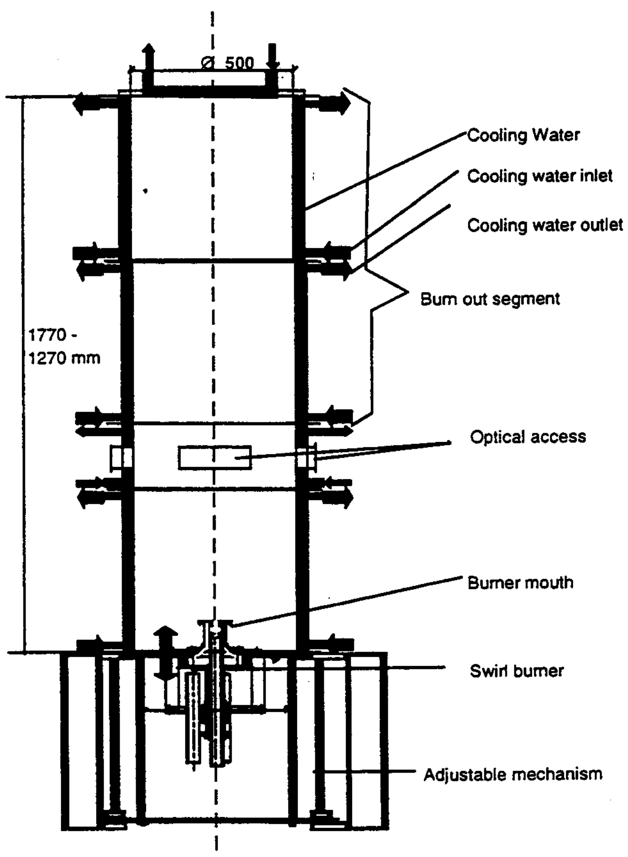

Fig. 2.1 Model combustor

2.2 THE STANDARD FLAME

Standard operating conditions have been defined for this flame:

Thermal power	P_{th} = 150 kW
Fuel air ratio	λ = 0.83
Swirl number	$S_{0,th}$ = 0.9
Wall temperature	$T_w \approx 100°C$
Operational pressure	p = 1 bar

All experimental data to be described subsequently were obtained at these standard conditions. The nominal swirl number $S_{0,th}$ is defined as the ratio of angular momentum flux to axial momentum flux.

Due to the swirl, a ring vortex area with recirculation is generated close to the axis. A highly turbulent, strongly reacting flame of the type-II character (Leuckel, 1971) (Fig. 2.2) is formed at that location.

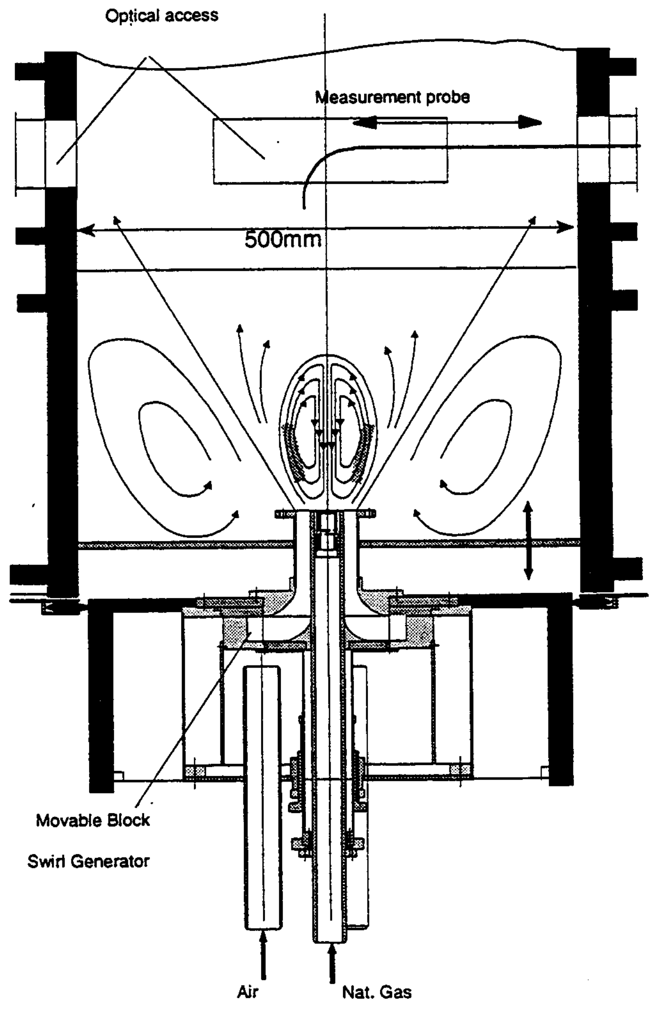

Fig. 2.2 Flow field of a type-II-flame with inner and outer recirculation zone

2.3 VELOCITY MEASUREMENTS

Locations for the measurements described subsequently have been defined at 14 axial planes with each plane comprising approximately 25 measuring points (Fig. 4.4). To resolve the inlet conditions at the burner, the distances between the axial planes were refined in nozzle region.

For measuring the velocities, a standard LDV system has been applied. A detailed description of the measurement technique is given in (Tacke et al., 1996). The mean axial, radial, circumferential components as well as their fluctuating components have been recorded.

2.4 CONCENTRATION MEASUREMENTS

For exhaust gas sampling a multihole suction probe (d_{Probe} = 6mm) thermostatisized above the dew point temperature (T>70°C) was used. The gas sample stream is heated and split into two streams. From one of the streams moisture is removed. The dry gases are analyzed for CO, CO_2, CH_4, H_2 and O_2 by means of standard analyzers (infrared-absorption-, heat conduction-, paramagnetism detectors). The NO- and NO_2-concentrations are measured selectively by chemoluminescence in the moist stream.

2.5 TIME RESOLVED TEMPERATURE MEASUREMENTS

The turbulent temperature fluctuations are measured by means of butt-welded thermocouples Pt/PtRh10 (d_{pearl} =100μm). The frequency response of the elements (f>3-10Hz) is compensated by a preamplifier (signal conditioner) in order to make the frequency fluctuations detectable. As the frequency response of the thermocouples is affected by convective heat transfer, it is determined at each measuring location by a calibrating procedure (Kohler, 1988). The method contains also a correction for radiation (Prade, 1993). The instantaneous temperature was recorded with a sampling frequency of 3.0 kHz. The mean temperature and the rms-value of the temperature fluctuations were calculated from 16,000 samples. For the present study, only mean temperatures were considered.

2.6 RADIATION MEASUREMENT TECHNIQUE

For the experimental investigation of the thermal radiation heat transfer to the wall, the principle of the emission spectroscopy was used. The setup has been presented multiple times (Ganz et al., 1997), (Krebs et al., 1993), (Ganz et al., 1996), (Koch et al., 1994), (Krebs et al., 1994), (Krebs et al., 1996) (Fig. 2.3). It is mainly composed of two components: the infrared-spectroscope and the blackbody. The blackbody serves as reference radiation source to calibrate the spectroscope.

With this arrangement, the normal radiation onto the combustor wall can be detected spectrally and time resolved.

Radiation emitted from either the combustor or the blackbody is reflected by a stepmotor controlled plane mirror into the infrared-spectroscope (Fig. 2.4 and 2.5). In the spectroscope, radiation is focused by a Cassegrain-mirror-optic into the monochromator. The mirror optics are gold coated in order to provide wavelength independent reflectivity.

In the monochromator, radiation is split spectrally with a resolution of 20nm for λ<2.8μm and 30nm for λ> 2.8μm. After passing a chopper and a high order filter, the radiation is finally focused onto an InSb-infrared-detector, which enables time resolved measurements at a sample rate up to 1000Hz with a high signal to noise ratio of 1000. For the present study, only mean intensities were considered.

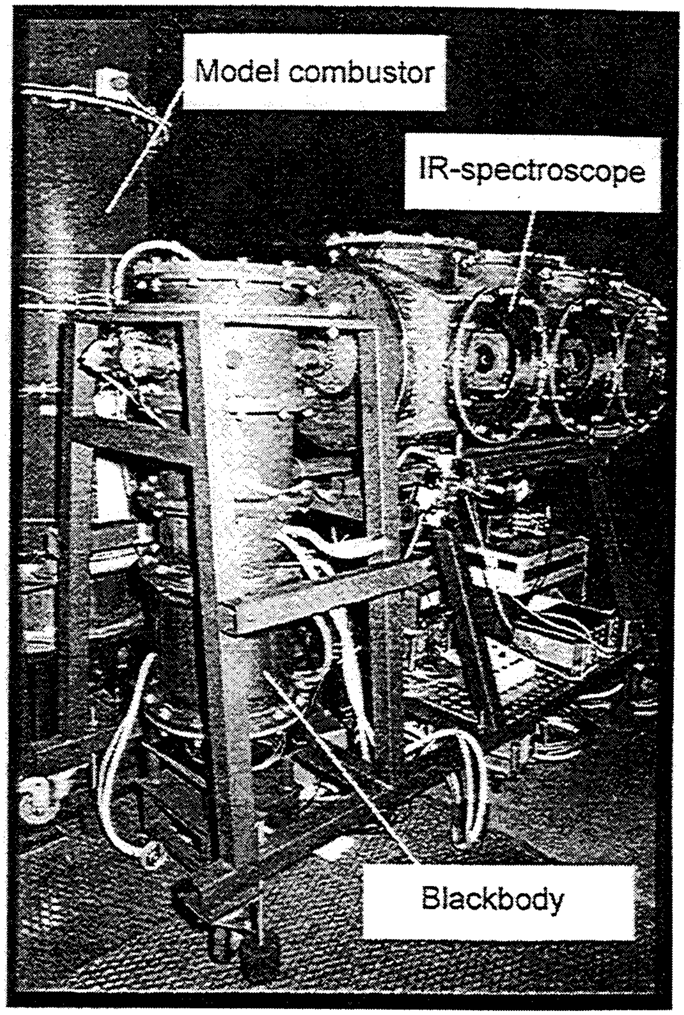

Fig. 2.3 IR-spectroscope and combustion chamber

The infrared spectroscope is purged by pure nitrogen to avoid reabsorption by participating gases. A sapphire window at the combustion chamber and a calcium-fluoride window at the detector serve for separation from the ambient air. For compensation of the absorption losses of the sapphire window at the combustor, an identical window has been placed in front of the blackbody. The sapphire window that separates the spectroscope from the combustion chamber is flushed by air to cool the window as well as to avoid accumulations of absorbing gases in the duct between combustion chamber and window (Fig. 2.4).

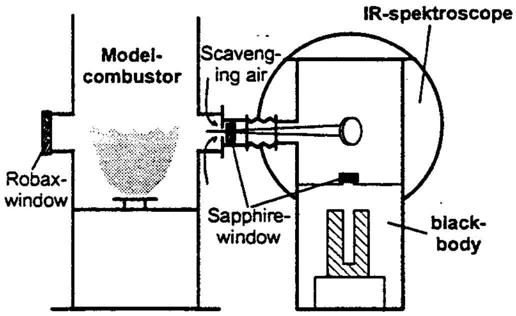

Fig. 2.4 Experimental setup for radiation measurements, front view

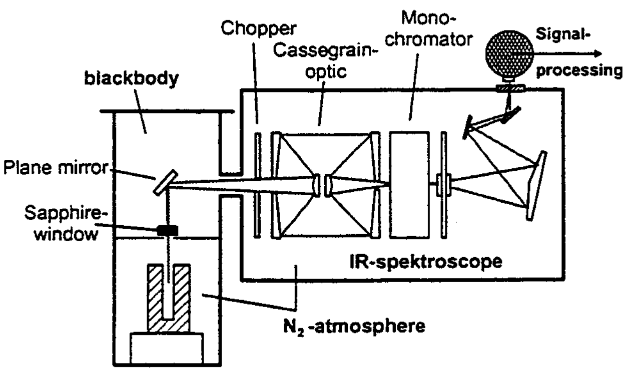

Fig. 2.5 Experimental setup for radiation measurements, side view

3. NUMERICAL CALCULATION OF THE REACTING FLOW FIELD

Supplementary to the experimental investigations, numerical calculations of the reacting flow field have been carried out. The goals of these calculations were on one hand to investigate the efficiency and the speed of numerical models used for predicting highly turbulent reacting swirling flames. Furthermore, the importance and the influence of radiative heat transfer was to be studied.

For the calculations of the reacting flow, a numerical code developed at the ITS (Kurrek and Wittig, 1994) has been used. This code is based on Finite Volume discretization and uses steady mean conservation equations for mass and momentum. To account for turbulence, a Standard-k-ε-model (SKE) (Launder and Spalding, 1972) as well as a Modified-k-ε-model (MKE) according to (Hirsch, 1995) have been utilized. Chemical reaction was considered by an Eddy-Dissipation-Model. For spatial discretization, an UPWIND-scheme as well as a higher order MLU-scheme (Noll et al., 1992) have been applied. The resulting equations are solved iteratively by an ILU-CG solver (Noll et al., 1991).

Taking into account radiative transfer is crucial for this type of flame, since the temperature correction due to radiation is not only important for the flow field itself but has also a strong effect on the NO_x formation. Therefore, the temperature distribution including radiative loss or gain has to be determined exactly for predicting the reacting flow field.

The 3-D radiative heat transfer code also utilizes the Finite Volume discretization. The method is based on the Discrete Ordinates approach (Chandrasekhar, 1960). To account for the spectral radiative properties of the combustion gases, a statistical narrow band model according to (Goody, 1964) has been chosen. The walls of the combustor were assumed to be black and diffusely emitting.

The same grid was used for calculating the radiative heat transfer and the reacting flow field. No modification of grid density was necessary.

The final solution is achieved by iteratively calculating alternatively the reacting flow and the radiative heat transfer and by taking into account the energy source term due to radiation (divergence of the radiative flux).

4. COMPARISON BETWEEN EXPERIMENT AND PREDICTION

Using the models described above, numerical calculations of the model combustor have been performed. Special emphasis was put on the importance of radiative heat transfer. The calculation of radiative heat has been performed spectrally as the usage of a simple gray gas model overpredicts the radiative heat flux by up to 40%. The addition of the radiative transfer calculation leads to an overall increase of the computational time by a factor 5.

4.1 VELOCITY

Fig. 4.1 and 4.2 show the flow stream lines of the calculated flow fields using the Modified-k-ε-model (MKE) and the Standard-k-ε-model (SKE), respectively as well as the measured flow field in the region near to the burner.

The flow field is typical for a type-II swirl flame. A recirculation zone is formed at the outer edge of the combustor and an inner recirculation zone can be found close to the axis directly above the burner orifice. Radiative transfer has a neglectable impact on the flow field.

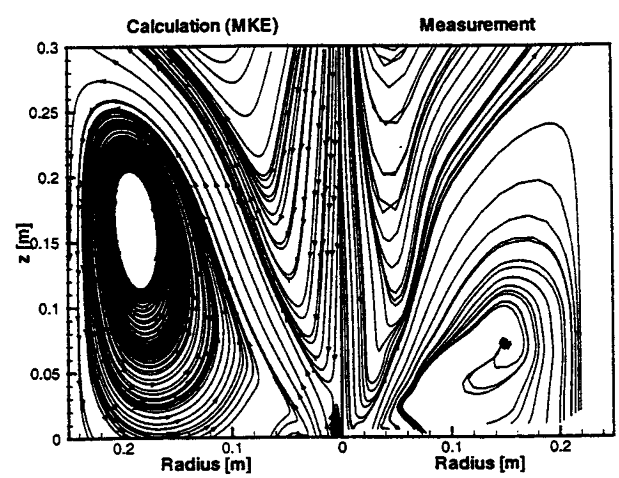

Fig. 4.1: Comparison of stream lines of the measured and calculated flow field using the Modified k-ε-model

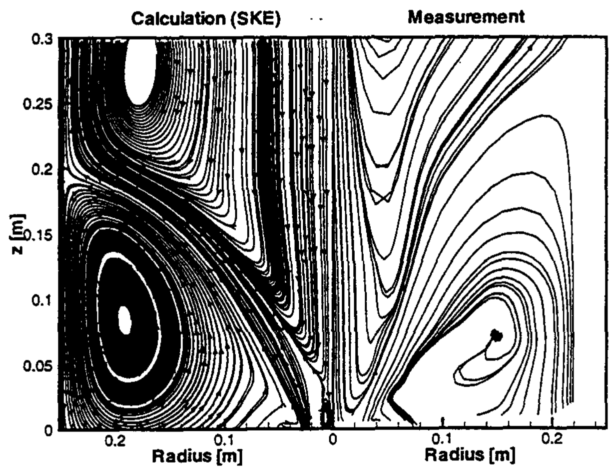

Fig. 4.2: Comparison of stream lines of the measured and calculated flow field using the Standard k-ε-model

The agreement between the calculated flow field using the MKE and the measured flow field is reasonably good. The main difference can be found at the axis directly above the burner. In the calculated flow field, a small recirculation zone is predicted that can not be found in the measurements. However, this may be due to a too coarse resolution of the flow field measurements.

The calculated flow field using the SKE differs strongly from the experiment. The size of the outer recirculation zone is underpredicted in axial direction. Consequently, the flame is spread radially and the inner recirculation zone is broadened. The SKE prediction also shows the small recirculation zone near the axis directly above the burner.

As the MKE predictions are in better agreement with the experiment it will solely be used for further analysis.

A detailed comparison of the velocity field is presented in Fig. 4.3. In particular, two planes will be analyzed. The first plane to be discussed is located at an axial position 6mm (z/D=0.1), the second 300mm (z/D=5.0) downstream the burner orifice. The first plane represents the inlet, the second the end of the visible flame. Because of the rotational symmetry of the flow field, which has been verified at the beginning of the test series, only the area between the burner axis and combustion chamber wall is represented.

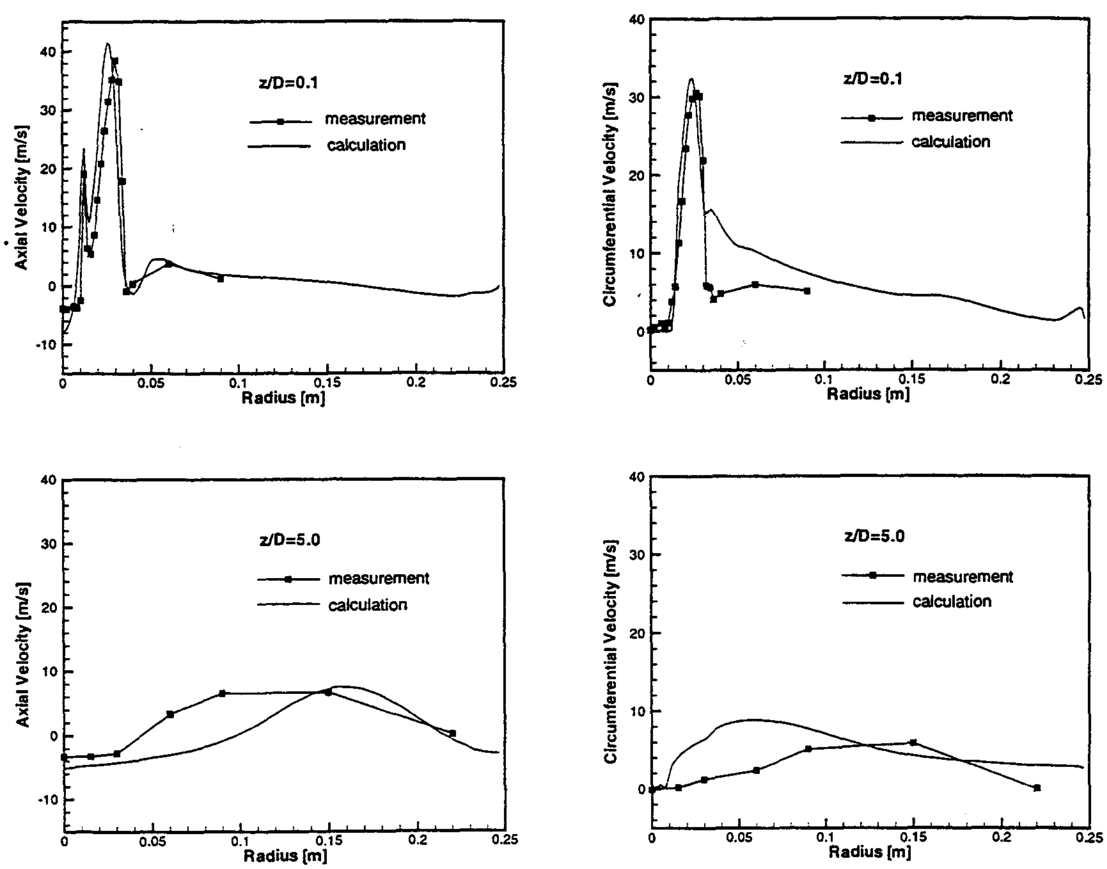

Fig. 4.3: Comparison between measured and calculated (MKE) axial- and circumferential velocities at z/D=0.1 and z/D=5.0

The predicted axial velocity profile at z/D=0.1 is in very good agreement with the experiment indicating that the inlet boundary conditions are correctly chosen. The circumferential velocity is overpredicted by the calculation at the axis and at the outer regions. Downstream of the burner at z/D=5.0, the profiles are broadened and smoothed. Furthermore, their maxima are slightly displaced.

Although the predicted flow field is close to the experiment (Fig. 4.1), even the MKE is not able to resolve the steep gradients in the circumferential velocity near the inlet. This result is not dependent on the grid refinement. This means that the radial exchange of the angular momentum in these regions is inadequately predicted and consequently, the circumferential velocities are overpredicted.

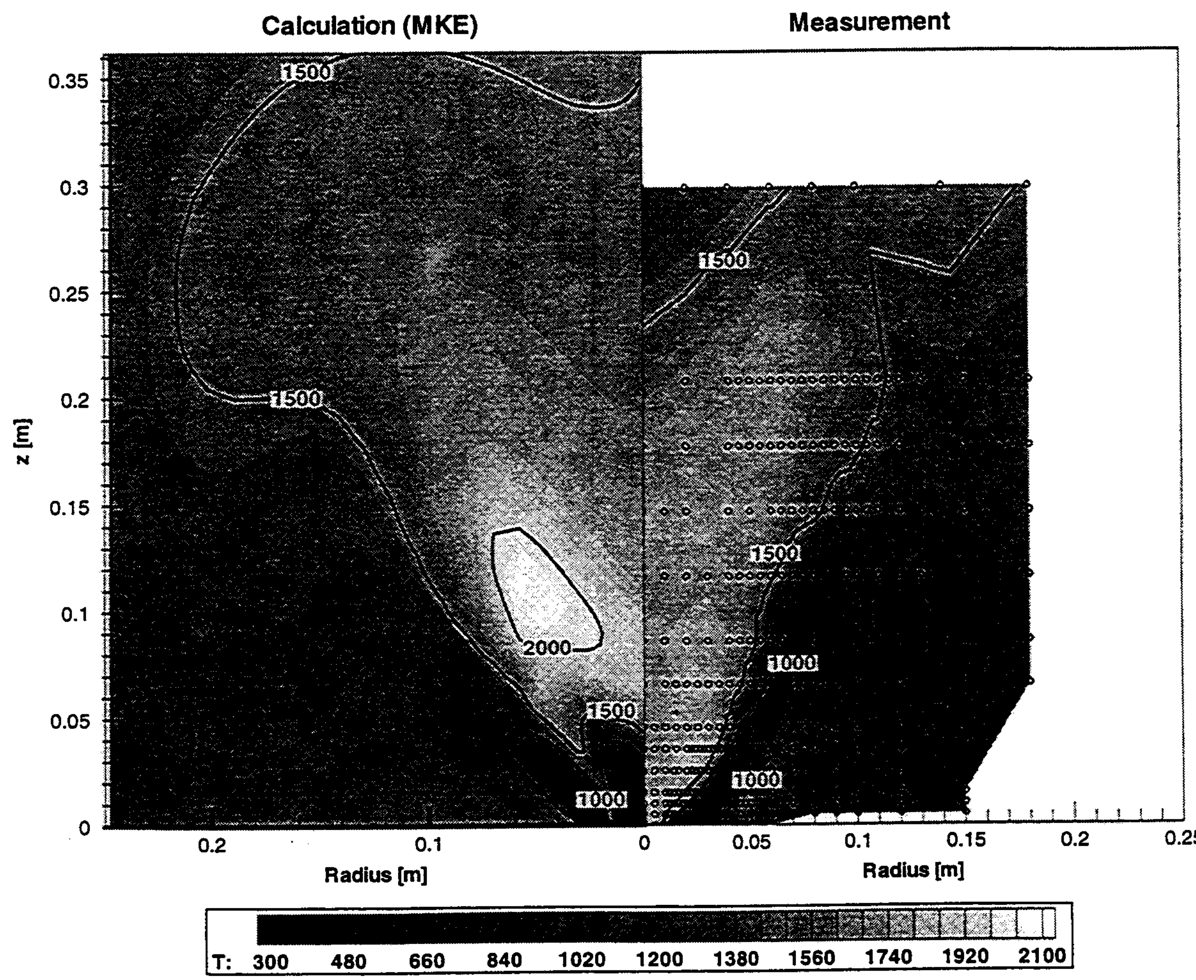

Fig. 4.4: Measured and calculated (MKE) mean temperature field in K in the region near the burner

4.2 TEMPERATURE DISTRIBUTION

In Fig. 4.4 the measured mean gas temperatures are compared to the calculated ones when taking into account radiative transfer. The temperature maximum is located at the edge of the inner recirculation zone. This is the main reaction zone with nearly stoichiometric conditions. Because of the injection of cold combustion air, the temperature decreases in radial direction and finally increases again in the outer recirculation zone.

The calculated temperature field mainly differs from the measured temperatures in the region near the axis directly above the burner orifice. Because of the small recirculation zone that has been found in the calculations, the recirculation of hot reaction gases back to the burner is handicapped. Therefore, the calculated temperatures are much lower than the measured ones in this region. Furthermore, it is remarkable that the calculated temperatures in the reaction zone are slightly higher than the measured temperatures. This is the result of insufficient mixing of the recirculating hot gases with the incoming cold air stream due to the small recirculation zone at the burner outlet.

4.3 EFFECT OF RADIATIVE HEAT TRANSFER

The goal of the present study is mainly to determine the effect of radiative transfer on reacting flows. Therefore, the impact of radiative transfer even for imperfect CFD predictions are presented in this chapter.

To illustrate the importance of radiative heat transfer, the difference between the predicted temperature field without and with radiation is shown in Fig. 4.5.

Due to the large dimensions of the flame, the temperature in the outer recirculation zone and especially in the main reaction zone is decreased by up to 400 K by radiative losses. On the other hand, the temperature of the cold inlet air stream and the natural gas stream is increased by radiative heating. This temperature variation is only due to radiative heat transfer as indicated by a neglectable influence of radiative heat transfer on the flow field. Furthermore, the relative change of the concentrations by radiation is about 2%.

The temperature differences accentuate the importance of radiative heat transfer when calculating the reacting flow field. The importance of radiation is especially pronounced in the chosen test case, with a very large combustor to burner area ratio and with water cooled combustor walls. But also in gas turbine combustors with a high operational pressure and resembling geometrical and burning conditions, the effect of radiative heat transfer is assumed to be in the same order. However, the radiative energy transfer is not only important for the flow field itself, but is crucial for the NO_x formation.

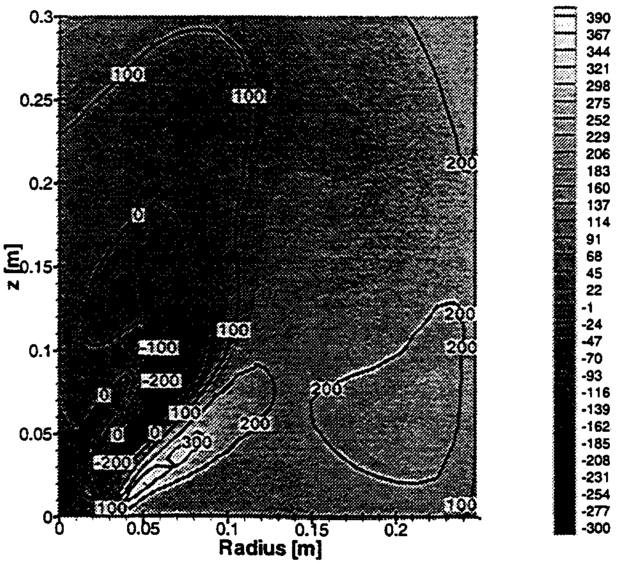

Fig. 4.5: Difference between the predicted temperatures neglecting and considering radiation

4.4 RADIATION INTENSITY

Examples of the radiation measurements and calculations are shown in Fig. 4.6 and 4.7 for the axial planes z/D=0.5 (z=30mm) and z/D=5.0 (z=300mm). In both Figures the spectral intensity is plotted versus wavelength. As the combustor is operated at lean conditions with a fuel air ratio of 0.83, only gaseous radiation is visible. Obviously no soot is produced.

As expected, the highest time averaged intensities are found at z=300mm. This is due to the high temperatures and high concentrations of the emitting species (CO_2, H_2O) at this location. Moreover, the hot gas zone is extended over the whole cross-section of the combustor at this axial location (Fig. 4.4).

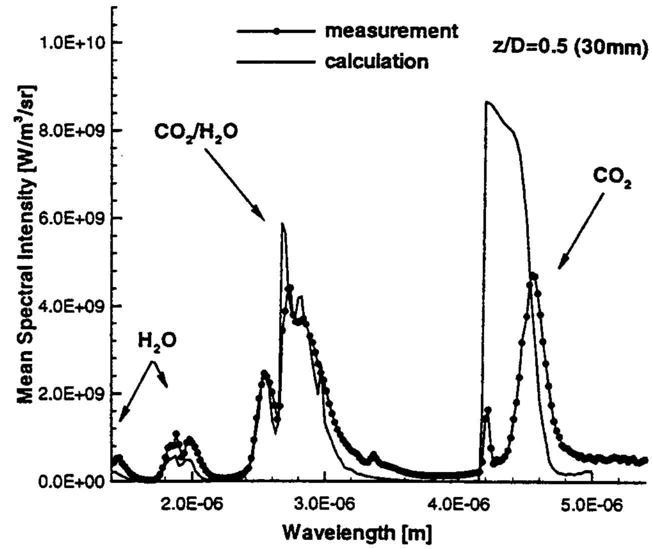

Fig. 4.6: Spectral intensity at z/D=0.5 (30mm)

Near the nozzle at z=30mm, higher intensities than expected from the flame geometry can be found (Fig. 4.6). The reason is the recirculation of hot reaction products in the outer recirculation zone near the wall (Fig. 4.1 - 4.3).

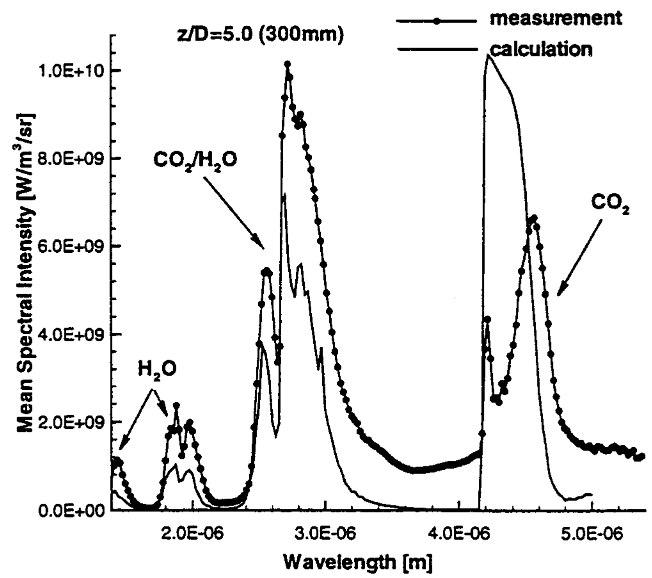

Fig. 4.7: Spectral intensity at z/D=5.0 (300mm)

The recorded radiation spectra contain a small amount of continuous background radiation. This is caused by the Quartz (Robax) window mounted at the opposite burner wall (Fig. 2.4). The surface temperature of the window varies from 370°C at z=30mm up to 600°C at z=300mm.

The calculated radiation spectrum for z/D=0.5 (30mm) agrees reasonably well with the measurements. Deviations in the CO_2-band at a wavelength range around 4.3 µm are due to reabsorption by the cold cooling air in front of the sapphire window (Fig. 2.4). The differences between measured and calculated intensity spectrum at z/D=5.0 is mainly due to the background emission from the Quartz (Robax) window that was not considered in the calculations.

5. SUMMARY

The particular focus of this paper is on analyzing the effect of radiative transfer as well as the efficiency of numerical models for predicting swirl flames. For validation of these predictions, extensive experimental investigations of a model combustor including measurements of species concentration, temperature distribution and spectral radiation have been performed. A highly developed radiation measurement technique has been employed for recording the spectral radiation incident on the combustor wall.

The model combustor resembles an industrial combustion system regarding the flame configuration (type-II swirl flame) and the highly turbulent flow conditions.

The measured velocity and temperature field as well as the recorded radiation spectra have been compared to numerical predictions. The comparison between experiment and prediction reveals that solving the time averaged conservation equations in conjunction with a Modified-k-ε-model and an Eddy-Dissipation-Model can only give reasonable results if radiative heat transfer is accounted for.

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