Int. J. Heat Mass Transfer. Vol. 21, pp. 1393–1402
Pergamon Press Ltd. 1978. Printed in Great Britain

PREDICTIONS AND MEASUREMENTS OF A TURBULENT, AXISYMMETRIC DUCTED DIFFUSION FLAME

D. G. ELLIMAN*, D. E. FUSSEY and N. HAY
Department of Mechanical Engineering, University of Nottingham, University Park, Nottingham, NG7 2RD, U.K.

(*Received* 30 *November* 1977 *and in revised form* 21 *March* 1978)

Abstract—A gas-fired axisymmetric combustor in the form of a duct with an axisymmetric baffle has been investigated using intrusive probes to determine the distributions of velocity, temperature and local equivalence ratio for a diffusion flame. The combustor was designed with a simple geometry to facilitate predictions with finite difference numerical models. A program loaned by CHAM Ltd., based on the velocity/pressure correction formulation and the $k-\varepsilon$ turbulence model was used. The flame brush was modelled with a simple square-wave fluctuation of mixture fraction. The experiments were carried out with Re (upstream cold pipe flow) $= 5 \rightarrow 8 \times 10^4$ and $\phi = 0.5 \rightarrow 0.9$. A 7-hole spherical pitot probe, an exposed bead thermocouple probe and a simple gas sampling probe were used.

The numerical model predicted the mean flow and mixing of the fuel and air, but without fluctuations gave poor temperature predictions. The introduction of fluctuations made a radical improvement in the flame prediction. The model performed sufficiently realistically to be of use in engineering design work, although there are still problems connected with the false diffusion errors and both the turbulence and mixing models.

NOMENCLATURE

C_μ, constant in turbulence model,
f, mixture fraction;
g, square of the fluctuation of concentration;
i, stoichiometric mass of oxygen per unit mass of fuel;
k, kinetic energy of turbulence;
K, constant in log law;
l_i, turbulence length scale at inlet ($0.8 \times$ annulus height);
Re, Reynolds number;
x, r, cylindrical polar coordinates;
u_i, mean inlet velocity;
w, θ, probe spherical polar coordinates;
y_p, normal distance from a grid node to wall;
ε, dissipation of energy;
PCE, Pressure Correction Equation;
ZNG, Zero Normal Gradient;
ZAG, Zero Axial Gradient.

1. INTRODUCTION

THIS PAPER reports work which aimed to verify the modelling of combustion chambers. A non-swirling gas-fired axisymmetric combustor in the form of a duct with a plane axisymmetric baffle was chosen for the investigation. This geometry was suitable for modelling with simple finite difference schemes since it avoided awkward boundary shapes. The objectives of the work were to make detailed mappings of velocity, temperature and mixture fraction within the combustor using intrusive probes of established design and to compare the measurements with computed predictions.

In the modelling of the flows it was decided to restrict the investigation to the $k-\varepsilon$ turbulence model [1], and a simple reaction scheme, and to investigate the effects of turbulent fluctuations in mixture fraction. A program based on the pressure/velocity formulation was made available to us by CHAM Ltd.

2. EXPERIMENTAL METHODS

The combustion rig

The conflicting requirements of (i) small physical dimensions to minimise the effect of radiation and to ensure economy in the consumption of fuel at an acceptable combustion intensity and (ii) large physical size to reduce probe blockage ratio effects and allow for detailed probing, resulted in a compromise design in which the duct was 150 mm in internal diameter by 400 mm in length. This gave a blockage ratio (probe area/flow area) of less than 0.5% for all the probes used. The construction adopted is shown schematically in Fig. 1. The baffle and working section were water-cooled but to avoid overcooling of the combustor walls, with resultant condensation and excessive heat loss, a layer of alumina granules was inserted between the water passages and the combustor wall. The baffle was 125 mm in diameter to ensure a large recirculation zone diameter to probe diameter ratio, and was machined with a 45° lip as shown in Fig. 1 to ensure accurate knowledge of the position of the dividing streamline. A maximum heat release of 700 kW was achieved producing an intensity of 113 MW/m^3. The choice of a rear-entry traverse unit dictated that the flow should be exhausted from the rig in a radial

* Present address: Quest Automation Ltd., Ferndown, Dorset, U.K.

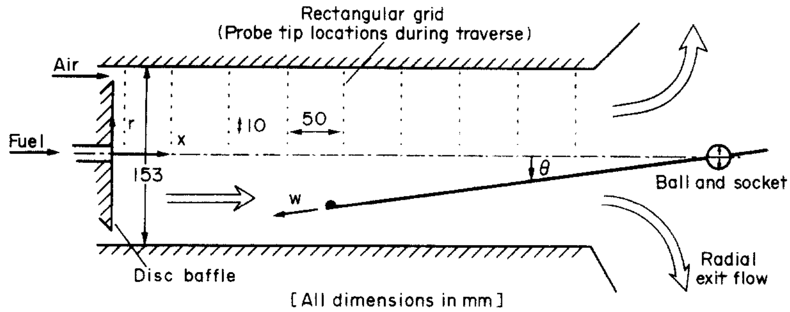

FIG. 1. Schematic of axisymmetric combustor.

direction. In order to minimise any loss of axisymmetry eight separate exhaust ports were arranged around the periphery of a 30° nozzle attached to the rear of the working section. The exhaust flows were quenched rapidly by the injection of water sprays. The rig was supplied with compressed air from Howden–Lysholm compressors, and a pneumatic pressure control system was used to maintain the running conditions constant. Methane was used as the fuel and was fed from a manifold of bottles through a pressure controller. The design of the probe traverse mechanism was based on the rear entry traverser developed at the National Gas Turbine Establishment (N.G.T.E.). This was arranged so as to traverse the rig in spherical polar coordinates originating from a spherical bearing at the rearmost axial point of the combustor. In practice, measurements were made on a rectangular grid in cylindrical polar coordinates originating at the baffle centre. The probe tip coordinates and the twist about the probe axis were each indicated and controlled remotely. An on-line, interactive computer program was used to convert (x, r) values to (w, θ) values, and the probe tip could be positioned to within ± 0.2 mm of a selected position.

Velocity probe

A direction sensitive probe in the form of a spherical seven-hole pitot has been developed at the N.G.T.E. [3] for use in combusting flows. This probe was loaned to us for the duration of the investigation. The head of the probe is made from 80% platinum/20% rhodium alloy and runs hot. A wide range of incident flow angles can be detected (up to $\pm 90°$) using a set of calibration curves. A problem arose however, in interpreting the probe measurements in regions of the flow where there was strong streamline curvature, for example in the wake behind the baffle. The calibration which was carried out in a rectilinear flow did not account for the situations in which the flow approached the head almost along the axis of the stem, since in some curvilinear flows the streamtube incident on the head does not interfere with the stem. Despite this, the probe was useful in determining the limits of the region of recirculation and in giving a qualitative measurement of velocity distribution in these regions. Outside the recirculation zones and at distances away from the wall greater than a few probe diameters, it behaved well and gave velocity measurements within $\pm 10\%$.

Laser anemometry was not available for use in this research programme, although the work of Hutchinson *et al.* [4] has demonstrated its usefulness in a similar investigation. The probe did, however, have the advantages that a rapid scan of the majority of the flow volume was possible with single hole access.

Thermocouple probe

A simple exposed bead thermocouple, mounted on a water-cooled stem, was used to measure local temperatures. The errors resulting from conduction and radiation were estimated and the corrections were checked by calibrating the probe against a double-shielded and aspirated thermocouple probe. After the correction procedure the measurements agreed within 2%.

Sampling probe

Since the probe sampled in clean conditions, a straightforward design could be used. The aim of the measurement was to determine local overall equivalence ratio, so trace species were neglected and reaction in the probe was of no consequence.

Combustion products and any unreacted reactants were sampled continuously through the probe, mixed with a stream of oxygen, and passed through an electrically heated oven which was held at 950°C. A residence time of about 1 s was achieved and the reaction was catalysed with platinum. The gas sample was then analysed with a two-channel chromatograph to determine local equivalence ratio.

3. NUMERICAL METHODS

The 2-D axisymmetric flow was modelled by the finite difference program GASEL [2]. The program used a hybrid differencing scheme and a staggered grid for the storage of pressure and velocity. The solution converged satisfactorily with under-relaxation factors of 0.73 or less, and solutions obtained with a grid size of 20×20 were shown to be essentially grid independent for small changes in grid spacing.

Table 1. Conservation equations and boundary conditions

Equation for conservation of	Boundary conditions used At walls	At inlet	At centre line	At exit
Mass	ZNG in PCE	ZNG in PCE	ZNG in PCE	ZNG in PCE
Axial momentum	Calculated from "log law of the wall" relationship.	Measured profile assumed similar for all conditions.	ZNG	ZAG
Radial momentum (no swirl considered)	Calculated from "log law of the wall" relationship.	Measured profile assumed similar for all conditions.	ZNG	ZAG
Kinetic energy of turbulence	generation = dissipation No diffusion to wall—generation uses log law velocity gradient.	$k = 0.03\, U_i^2$	ZNG	ZAG
Dissipation rate	$\varepsilon = \dfrac{C\mu^{3/2}k^{3/2}}{KY_p}$	$\varepsilon = \dfrac{k^{3/2}}{l_i}$	ZNG	ZAG
Mixture fraction	$f = -1/i$	$f = 1$ at fuel inlet $f = -1/i$ at air inlet	ZNG	ZAG
Concentration fluctuation	0	0	0	0

Combustion models

Following the work of Gosman [6], Lockwood [7] and the recommendation of Khalil, Hutchinson *et al.* [4, 8], the combustion process was represented by a one-step finite reaction rate model with a square wave form for the time variation of f, the mixture fraction. This model allows the coexistence at different times of fuel and oxygen at points within the flame, and gives rise to a diffuse reaction zone or flame "brush".

The partial differential equations solved in the program are summarised in [8], and Table 1 describes the boundary conditions applied in this work. In contrast with [8], the enthalpy equation was not solved since, at low Mach Number with near adiabatic boundaries, h becomes, to a good approximation, a unique function of mixture fraction and inlet temperature.

4. EXPERIMENTAL RESULTS

Isothermal, non-reacting flow

As a preliminary exercise, isothermal flow was investigated in the combustor.

As a perfect axisymmetry of the flow is assumed in the computer program, the degree to which the flow in the combustor approached this ideal was investigated. The flow was found to be less than perfectly axisymmetric at all flow conditions, and the offset of the recirculation zone from the axis was not a simple radial displacement, but varied in magnitude and direction with axial position. However, the variations in axial velocity were within $\pm 5\%$ of the mean.

The inlet profile shown in Fig. 2 was measured at a plane 5 mm from the baffle, since it could not be measured at the inlet plane with the probe of finite size. A small radial velocity component was induced by the angled sharp edge of the baffle. The relatively high scatter in the measured radial velocity profile is due to the very small values involved and to errors in measuring the angle of flow.

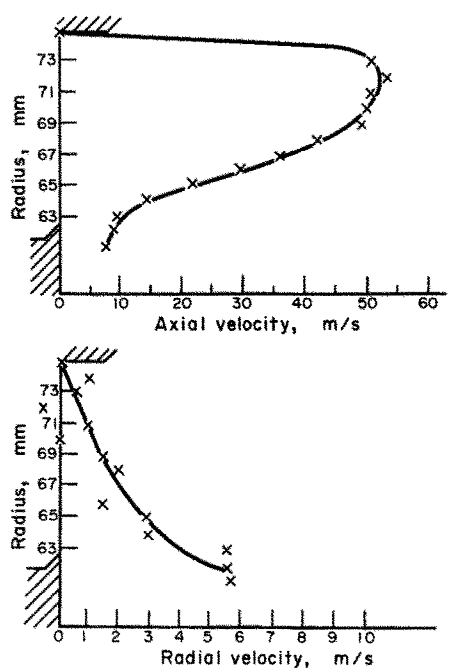

FIG. 2. Measured velocity profiles at inlet (taken at a plane 5 mm from baffle).

Combustion flows

Three conditions were chosen at two values of Reynolds number and two values of equivalence ratio,

Table 2. The mass flow rates of fuel and air at conditions 1, 2 and 3

Condition No.	$Re \times 10^{-4}$ (upstream "cold" pipe flow)	ϕ	$\dot{m}_{air}$ kg/s	$\dot{m}_{fuel}$ kg/s
1	5	0.9	0.10	0.005
2	8	0.9	0.17	0.0086
3	5	1.6	0.10	0.0096

in order that the effect of both parameters might be studied (see Table 2).

The degree to which an axisymmetric flame was produced within the combustion chamber was investigated using the exposed bead thermocouple probe. Figure 3 shows typical polar temperature distributions for condition 1 at two arbitrary axial and radial locations in the combustion chamber.

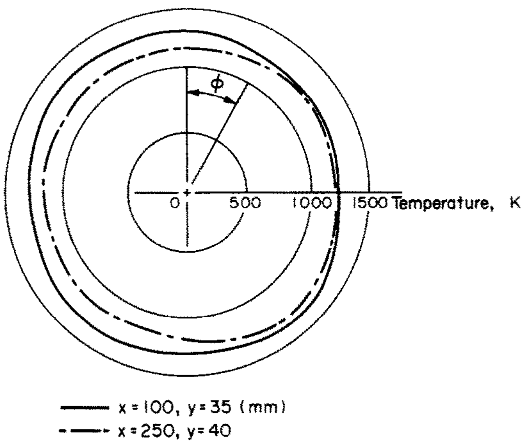

FIG. 3. Radial temperature distribution (condition 1)—showing departure from axisymmetry.

Detailed investigation of the temperature, fuel-air ratio and velocity fields within the combustion chamber

Detailed measurements of the combusting flow were taken at condition 1, using the zero degree plane. The measured contours of temperature at equilibrium conditions are shown in Fig. 4 (top half). The dominant features are a cool central fuel jet, a cool film of air along the wall and an annular flame stabilised by a recirculating flow of very hot gases downstream of the baffle. The reaction zone is diffuse and produces the characteristic flame "brush". The temperatures are very much lower than the stoichiometric value for the fuel used, which is in the region of 2100 K for an inlet temperature of 300 K. The time-averaged fuel air ratio measurements are shown in Fig. 5 (top half). In the rich region along the combustor axis the fuel–air ratio could not be quantified as it was found to be impossible to burn all the excess fuel in the furnace for equivalence ratios in excess of about 1.5. Figure 5 again shows the diffuse reaction zone indicated by the temperature measurements. In fact, with the exception of a small region near the air inlet, a finite equivalence ratio was found throughout the combustor volume. The equivalence ratio contours thus show a remarkably shallow gradient between the methane jet and the wall, with some local distortion in the recirculating region immediately behind the baffle.

The velocity distribution in the combustion chamber is presented as contours of axial and radial velocity in Figs. 6(a) and (b) (top half). In order to interpret the 7-hole probe output, a local density was required. This was calculated from the measured values of temperature and fuel–air ratio using a mean molecular weight and an equation of state. In the "out of range" regions the fuel–air ratio was taken as that fuel-rich ratio which would produce the measured temperature on complete combustion, although the actual fuel–air ratio could lie anywhere between 1.5 and infinity, the error in the density, and therefore the velocity, arising from this assumption was unlikely to be greater than $\pm 5\%$ however, except in regions of very low temperature where density is a stronger function of temperature and equivalence ratio.

Two concentric contra-rotating toroidal vortices are in evidence behind the baffle. The recirculation zone extends to an l/d of approximately unity. Other important features of the combusting flow are a high velocity jet along the axis, and an annular air jet along the chamber wall. These jets initially begin to decay in velocity. Downstream of the recirculation zone, however, expansion of the gases following combustion causes an acceleration across the reaction zone. The coincidence of the larger region of recirculation flow extending from the baffle lip, with the high temperature zone of Fig. 5, is a notable feature of the velocity field.

Only the zero radial velocity contours are shown in Fig. 6(b), since the radial components are small compared with the axial components for most of the flow. Figures 6(a) and (b) demonstrate the usefulness of

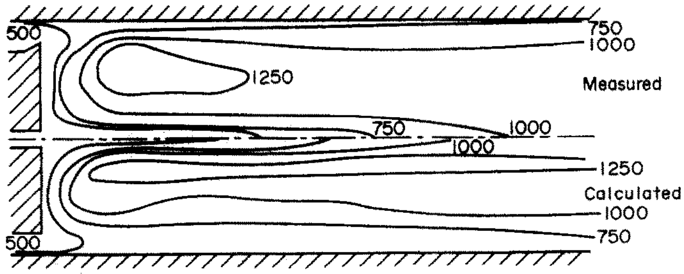

FIG. 4. Comparison between measured and calculated temperature distributions (condition 1).

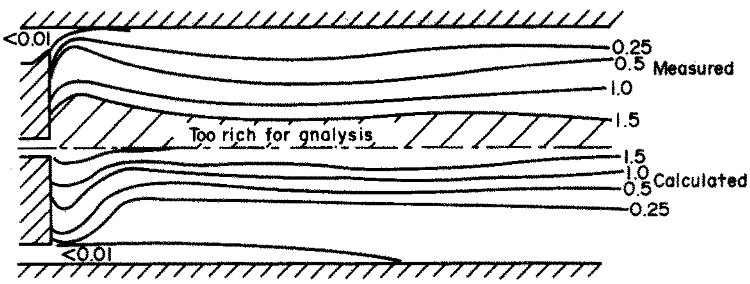

FIG. 5. Comparison between measured and calculated equivalence ratio distributions (condition 1).

the velocity probe in establishing the overall flow pattern, depsite its inherent weaknesses discussed in Section 2.

5. COMPARISON BETWEEN MEASURED AND COMPUTED RESULTS

The experimental rig was modelled using a 20×20 non-uniform grid, the grid spacing increasing monotonically from the walls and axis of symmetry. The program was considered to have converged once the sum of the out of balance mass sources was less than 1×10^{-5} of the inlet mass flow. Any change in the flow pattern with further iteration was found to be negligible ($<0.5\%$). The variation of levels of turbulence at inlet, over the range of values which are physically plausible, had very little influence on the results. The recirculation zone length was also found to be insensitive to the profile of axial velocity assumed at inlet. The small radial velocity component at inlet was, however, found to have a significant effect. In order to provide a more exact boundary condition at inlet, a profile similar to that of Fig. 2 was used, and the velocity magnitude was adjusted to make the integrated mass flow equal to that metered. This gave a significant improvement in agreement between the measured and predicted flow. Underprediction of the length of the recirculation zone was apparent. Although Pope [9] suggests that the $k-\varepsilon$ turbulence model will lead to such underprediction, it is believed that the errors caused by false (numerical) diffusion in the recirculation zone outweigh any deficiency in the turbulence model (see Section 6).

Combustion flows

Profiles of temperature, fuel–air ratio and velocity were measured at each of the chosen conditions. Measurements were taken along the axis and at planes 100 mm and 250 mm from the baffle. The fuel–air ratio measurements were used to calculate the local density and hence the velocity, and were also used to calculate ideal temperatures denoted by the crosses in these diagrams which correspond to 100% combustion efficiency. At each of the three conditions the measured temperature profiles (Figs 7–9) rise smoothly from the wall temperature to a maximum near the mean radius, and fall back to the temperature of the fuel–rich central jet on the combustor axis. This jet remains at near atmospheric temperature for about 100 mm and then begins to

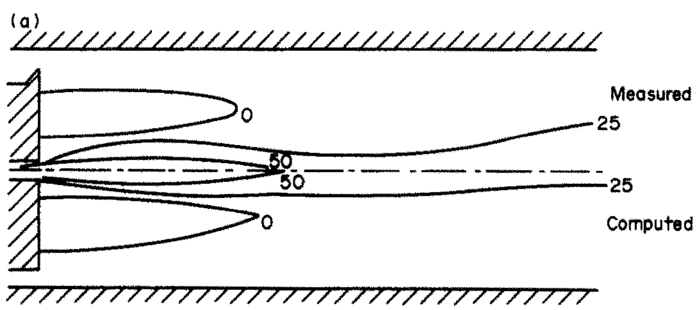

FIG. 6. (a) Comparison between measured and calculated contours of axial velocity (condition 1).

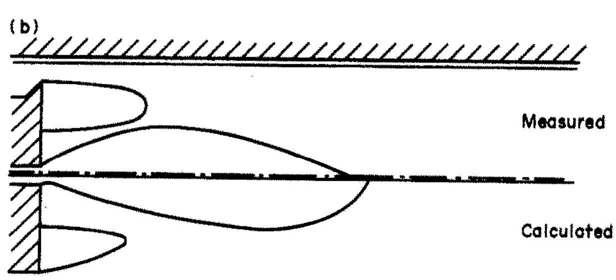

FIG. 6. (b) Comparison between measured and calculated contours of radial velocity (condition 1).

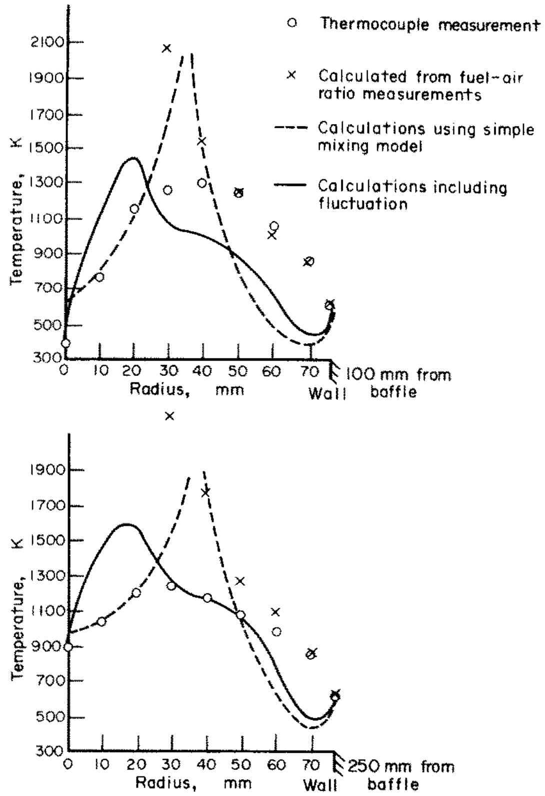

FIG. 7. Comparison between measured and calculated temperature profiles (condition 1).

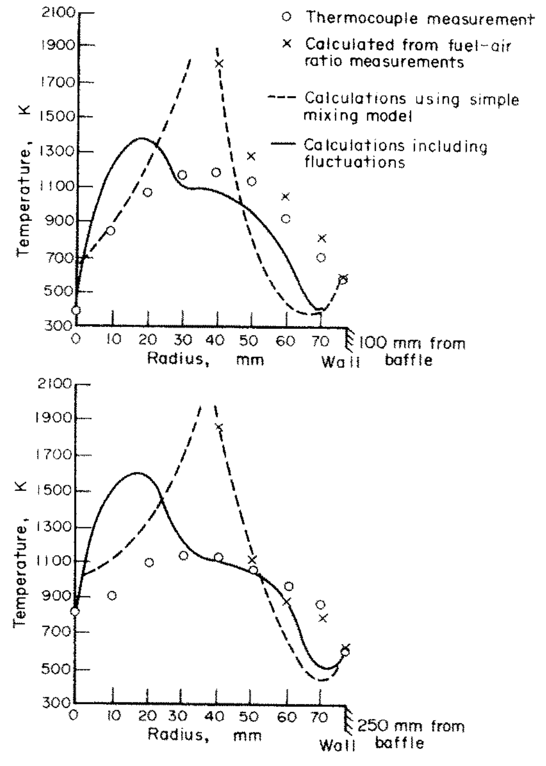

FIG. 8. Comparison between measured and calculated temperature profiles (condition 2).

rise in temperature in a nearly linear manner with distance along the axis. The radial profiles reflect this increase in temperature near the axis, which is the predominant difference between the profiles measured at 100 mm and 250 mm from the baffle. The change from condition 1 to condition 2 represents an increase in Reynolds number from 5×10^4 to 8×10^4 at a constant equivalence ratio of 0.9. Condition 2 shows a fall in temperature of about 80 K (Figs 8 and 9) over most of the combustor volume, the temperature profiles however remaining similar in shape to those measured at condition 1.

The change from condition 1 to condition 3 represents an increase in equivalence ratio from 0.9 to 1.6 at constant Reynolds number. Figures 7 and 9 show that the overall temperatures are much lower at the fuel–rich condition. This is reflected in lower velocities in the air jet and flame although very much higher velocities are found in the methane jet as an obvious consequence of its greater mass flow rate. The temperature profile at 250 mm is much flatter than that for condition 1 and indicates that the angle of flame spread from the baffle (defined by the mean stoichiometric contour) is appreciably steeper than the case for conditions 1 and 2.

6. DISCUSSION

The use of a fluctuations model for the numerical prediction

Typically about 400 iterations were required to achieve convergence for the simple mixing model.

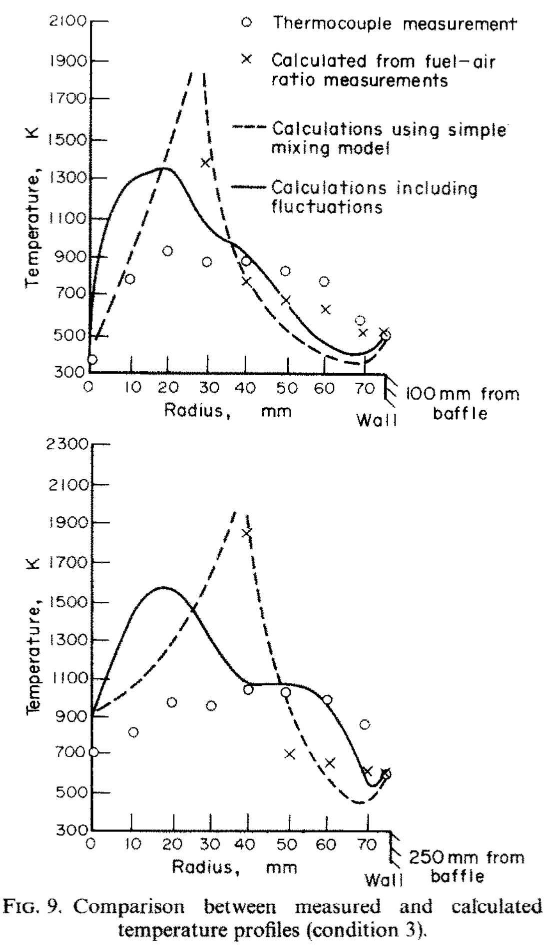

FIG. 9. Comparison between measured and calculated temperature profiles (condition 3).

This represented only a small increase in computational effort over that required for the cold flow situations. The inclusion of the transport equation for g as a basis for the fluctuation model, had an adverse effect on the solution stability. At first convergence was so slow and uncertain that only a very poor solution existed after 1000 iterations. This situation did not improve with changes to the under-relaxation parameters, but was found to be sensitive to the boundary condition used. The numerical method proceeds to convergence by the convection and diffusion of errors in the solution out to the boundaries, where they are corrected. The stability of the solution and the rate of convergence are thus strongly dependent on the "tightness" of the boundary conditions used.

Initially a zero normal gradient was specified at each wall boundary for the g equation, and zero at inlet and outlet planes. This is a "loose" condition and its replacement by a zero value of fluctuation at each boundary yielded a much more strongly convergent solution. This was the practice of Spalding [10]. However about 100 iterations were now required to achieve a converged solution. The use of the g equation thus appears to decrease the stability of the solution, probably as the result of making the coupling between a calculated value of the mixture fraction and the temperature (and therefore the density) very much less direct. The use of a zero value for g at the inlet boundary is reasonable as, clearly, concentration fluctuations cannot occur in a one component environment. The approximation may also be used at the outlet if this is located sufficiently far downstream to have negligible influence on the region of interest. At the wall, fluctuations must be damped to zero. The actual form of variations through the boundary layer is uncertain, although in a turbulent flow fluctuations occur down to the laminar sublayer and a zero value at the wall would be a poor approximation. A zero value at the axis of symmetry may be appropriate in an equilibrium flow, since the radial concentration gradient is zero at this boundary, but in a turbulent recirculating flow, fluctuations are convected and diffused to the axis making a zero value here improbable.

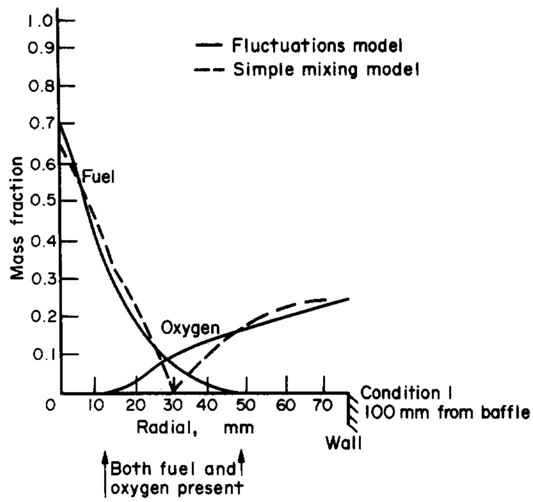

FIG. 10. Predicted profiles of fuel and oxygen concentrations.

However, in the interest of obtaining a solution the zero value boundary conditions were employed for the results presented in this section. Figure 10 shows the predicted radial variation of the mass fractions of fuel and oxygen 100 mm from the baffle at condition 1. The dashed line shows the result obtained by the simple mixing model for which fuel and air do not coexist at any point and the stoichiometric temperature is predicted at the flame front (defined by the stoichiometric contour which is located at a radius of 30 mm in the diagram). However, highly turbulent diffusion flames display a characteristic flame "brush" at well below the stoichiometric temperature. The limits of the flame "brush" as predicted for each of the three conditions are plotted in Fig. 11. The stoichiometric contour predicted by the simple mixing model is also shown. The results for conditions 1 and 2 are very similar indicating a relatively weak Reynolds number effect. Condition 3 however exhibits a flame "brush" which diverges more rapidly from the axis than those predicted at conditions 1 and 2.

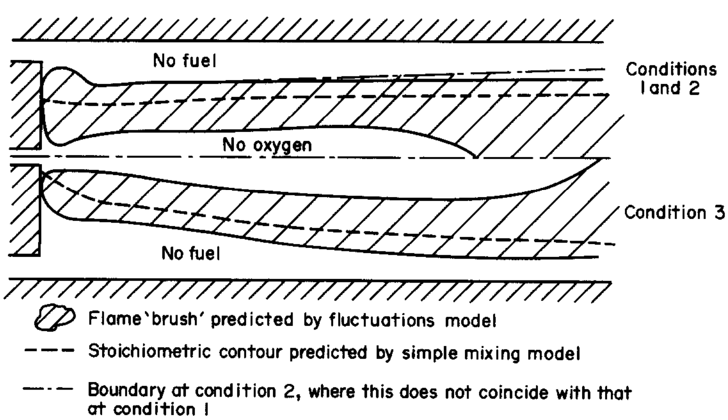

FIG. 11. Flame brush predictions for each condition.

The most serious disagreements lie in the presence of a hot "streak" at a radius of about 20 mm, (Fig. 4) and in the cool jet predicted along the combustor wall. Several explanations for the discrepancies in the region of the cool wall jet can be suggested. It is possible that the flame has been locally distorted by the presence of the probe. Another possibility is that a cyclic instability was present in the flame. The mean measurements taken were, of course, unable to differentiate between steady temperature, a cyclic instability about the measured mean, and a cool jet containing intermittent eddies of hot gas.

A third possibility is that the fluctuation model was unable to model the fluctuations in the areas of disagreement. These areas lie close to the boundaries which were poorly specified in the solution of the g equation. The temperature profiles shown in Fig. 7 to 9 generally indicate a good agreement near the mean radius of the combustion chamber, the temperature being over-predicted near the axis and under-predicted near the wall, although on the axis there is good agreement. These peaks in the calculated profile seem to be inappropriate and it is clear that a smoother curve would give greatly improved predictions. The level of fluctuation in the region of these peaks has been reduced by the influence of the zero value of g at the near boundary, and this is a significant shortcoming of the model.

Assessment of the numerical model

The numerical model has given predictions of sufficient accuracy to be of immediate use to combustion designers. The most pressing need for improvement in the present form of the model lies in the boundary specifications of the g equation, particularly at the axis of symmetry. Wall functions for g in the region of the solid boundary could be developed using local equilibrium in the boundary layer and assuming zero diffusion to the wall. These changes are likely to lead to a significant improvement in the accuracy of the predictions.

A full 3-D prediction program (such as that used in [11]) would be advantageous in dealing with a known departure from axisymmetry in the inlet profiles.

A general criticism of the turbulence model is that it attempts to characterise the local structure of the turbulence by a single length scale. The broad band structure of the eddies is a well documented feature of turbulent flows and it is very optimistic to hope that a typical length scale will suffice. Notwithstanding these criticisms, recent investigations (e.g. [12]) have shown that the numerical diffusion produced by upwind differencing can have severe effects on predictions in regions where the flow is orientated at large angles to the grid and where there are strong gradients in flow properties transverse to the stream direction. It is estimated that the errors incurred by false diffusion could only be eradicated with certainty by using a grid of 2000×2000 points which is clearly beyond existing computing facilities. The false diffusion in the results presented here is a serious problem in the recirculation region, eclipses any defect in the turbulence model, and is probably the cause of the discrepancy in recirculation zone lengths noted earlier.

Despite these reservations concerning the flow prediction, the major failing of the numerical scheme is in the fluctuations model.

Summary of discrepancies between theory and experiment

The discrepancies between theory and experiment have their sources in both experimental errors and in inadequacies of the numerical model. Notional percentage errors for some of the effects are noted below, although there may be interactions between sources of errors and errors produced primarily in the recirculation zone or near the walls might well extend beyond these regions. Clearly, an overall discrepency between theory and experiment can be quoted by reference to earlier figures, for example in Fig. 7 it can be seen that the overall errors lie between 25 and 40%, although, outside the recirculation zone (at 250 mm from baffle) there is very much better agreement apart from the region near the wall and the hot streak described earlier.

However, these discrepancies are comprised of the following notional errors:

Experimental errors.

(a) Errors in thermocouple calibration. $\pm 2.5\%$

(b) Errors in fuel–air ratio analysis. $\pm 10\%$

(c) Errors in 7-hole probe analysis (outside recirculation zone and not close to boundaries)—$\pm 10\%$ in velocity magnitude and $\pm 10\%$ in velocity direction (leading to high percentage error in small component of radial velocity).

Numerical errors.

(d) Errors in turbulence model—inferred from cold flows—$\pm 5\%$.

(e) Errors in fluctuations model—Strong interlinkage—$\pm 15\%$ for both effects.

(f) Errors in chemical reaction model—Strong interlinkage—$\pm 15\%$ for both effects.

(g) Errors in finite difference technique (mainly false diffusion)—typically 20% reduction in length of recirculation zone due to tightening of streamline curvature.

It should be noted that the estimate for error (d) is based on the region outside the recirculation zone, and the estimate for errors (e) and (f) is based on regions away from the walls and axes, although it has been seen that the anomalous results can occur in other regions, for example the hot-streak in Figs 7–9.

Errors caused by mismatching between the model and experiment.

(h) Lack of axisymmetry, invalidating 2-D approach.

(i) Possible periodically unsteady phenomena or large scale, low frequency turbulence, which cannot be modelled with existing turbulence model.

(j) Incompatibilities in boundary conditions, affecting regions near walls, e.g. non-adiabatic walls, fluctuations near walls and axis, inlet velocity profiles.

7. CONCLUSIONS

(i) The simple mixing reaction model realistically predicts the time mean mixing of the fuel and air, but gives unrealistic flame predictions.

(ii) The addition of a fluctuation model greatly improves the temperature predictions. This indicates that the processes dominating combustion in the measured turbulent diffusion flame are indeed mean mixing of the fuel and air and temporal fluctuations.

(iii) The shortcomings of the combustion model are very much greater than those of the turbulence model, and should therefore recieve greater attention.

(iv) Poorly specified boundary values for the g equation were responsible for much of the discrepancy between the predictions and measurements.

(v) Despite some reservations about the applicability of the $k-\varepsilon$ turbulence model in the recirculation region, the predominant weakness of the numerical technique in this region is the false diffusion effect. Further work is necessary to quantify this error in complex recirculation zones, and stable numerical procedures based on the central differencing approach need to be developed.

(vi) Complex models do not appear to be any more satisfactory at present than the $k-\varepsilon-g$ model, probably because of the dominance of false diffusion errors. The user of numerical predictions of this type should guard against this pitfall before concerning himself with detailed improvements in other areas.

(vii) The model is useful for engineering design, provided that the user has an appreciation of the shortcomings described above.

Acknowledgements—This work was supported by the Science Research Council and formed part of a collaborative research programme between Rolls-Royce Ltd., Derby, and the Department of Mechanical Engineering of the University of Nottingham. We are grateful to CHAM Ltd. for the loan of the computer code and NGTE for the loan of the 7-hole pitot probe.

REFERENCES

1. B. E. Launder and D. B. Spalding, *Mathematical Models of Turbulence*. Academic Press, London (1972).
2. K. J. Matthews, GASEL; A two dimensional flame prediction model, CEGB Research Report R/M/R232.
3. J. J. Macfarlane, An omni-directional velocity vector probe suitable for use in gas turbine combustors. Design, development and preliminary tests in a model combustor, ARC CP No. 1254, HMSO (1973).
4. P. Hutchinson, E. E. Khalil, J. H. Whitelaw and G. Wigley, The calculation of furnace-flow properties and their experimental verification *J. Heat Transfer* **99**(2) 276 (1976).
5. A. D. Gosman and F. C. Lockwood, Prediction of the influence of turbulent fluctuations on flow and heat transfer in furnaces, Imperial College, London, Rep. HTS/73/52 (1973).
6. F. C. Lockwood and A. S. Naguib, The prediction of the fluctuations in the properties of free, round jet, turbulent diffusion flames, *Combust. Flame* **24**, 109 (1975).
7. S. E. Elgobashi, Characteristics of gaseous turbulent diffusion flames in cylindrical chambers, Ph.D. Thesis, University of London (1974).
8. E. E. Khalil, D. B. Spalding and J. H. Whitelaw, The calculation of local flow properties in two dimensional furnaces, *Int. J. Heat Mass Transfer* **18**, 775 (1975).
9. S. B. Pope, The calculation of the flow behind bluff bodies with and without combustion, Ph.D. Thesis, University of London (1976).
10. D. B. Spalding, Turbulence models and their experimental verification, Imperial College, London, Reps. HTS/73/18 and HTS/73/19 (1973).
11. W. P. Jones, C. H. Priddin, R. de Chair and M. J. Roberts, A Comparison between predicted and measured species concentrations and velocities in a research combustor, Paper No. 41, *AGARD Symposium on High Temperature Problems in Gas Turbine Engines*, Ankara (September 1977).
12. G. de Vahl Davis and G. D. Mallinson, An evaluation of upwind and central differences approximations by a study of recirculating flow, *Comput. Fluids* **4**, 29 (1976).

PREVISION ET MESURE DE FLAMMES DE DIFFUSION TURBULENTES EN CONDUITE AXISYMETRIQUE

Résumé—Un brûleur à gaz en forme de conduit axisymétrique avec un baffle axisymétrique a été exploré avec des sondes pour déterminer les distributions de vitesse, de température pour une flamme de diffusion. Le brûleur a été conçu de géométrie simple pour faciliter les calculs à l'aide de modèles numériques aux différences finies. On utilise un programme prêté par CHAM Ltd., basé sur la formulation de correction vitesse/pression et sur le modèle de turbulence k-ε. La flamme est modélisée par une fluctuation en créneau de la fraction de mélange. Les expériences sont rélisées avec Re (écoulement froid en amont dans le tube) $\subset 5 \rightarrow 8 \subset 10^4$ et $\theta = 0{,}5 \times 0{,}9$. Une sonde sphérique à 7 trous est utilisée avec une sonde à thermocouple et une sonde pour analyse de gaz. Le modèle numérique prédit l'écoulement moyen et le mélange d'air et de combustible, mais sans fluctuations donne de mauvaises estimations des températures. L'introduction des fluctuations améliore radicalement les prévisions dans la flamme. Le modèle est suffisamment réaliste pour être utilisé dans les travaux d'ingénieur bien qu'il demeure des problèmes liés aux erreurs de fausse diffusion et aux modèles de turbulence et de mélange.

BERECHNUNGEN UND MESSUNGEN EINER TURBULENTEN, ACHSENSYMMETRISCH GEFÜHRTEN DIFFUSIONSFLAMME

Zusammenfassung—Es wurde ein gasbefeuerter, achsensymmetrischer Brenner in Form einer Röhre mit einer achsensymmetrischen Blende untersucht. Mit Sonden wurde die Geschwindigkeits- und Temperaturverteilung und die örtliche Äquivalenzrate für eine Diffusionsflamme bestimmt. Der Brenner wurde geometrisch einfach konstruiert, um die Anwendung eines numerischen Modells mit finiten Differenzen zu ermöglichen. Es wurde ein Programm von Cham Ltd. verwendet, welches auf der Geschwindigkeits-/Druckkorrektur-Formel und dem k-ε Turbulenzmodell basiert. Im Modell des Flammenstrahls wurde für die Fluktuation des Mischungsverhältnisses eine einfache Rechteckfunktion verwendet. Die Experimente wurden mit Re-Zahlen (Aufwärtsströmung im kalten Rohr) von 5 bis $8 \cdot 10^4$ und Werten für Φ von 0,5 bis 0,9 durchgeführt. Es wurden eine kugelförmige Pitot-Sonde mit 7 Bohrungen, ein unisoliertes Thermoelement und eine einfache Gasabsaugsonde verwendet. Das numerische Modell beschrieb die mittlere Geschwindigkeit und die Vermischung von Brennstoff und Luft, konnte aber ohne Fluktuationen die Temperatur nur schlecht wiedergeben. Die Einführung von Fluktuationen erbrachte eine ganz wesentliche Verbesserung in der Flammenbeschreibung. Das Modell ist genügend wirklichkeitsnah und damit brauchbar für technische Entwurfsaufgaben, obwohl es noch Probleme bei der Diffusion und dem Turbulenz- und Mischungsmodell gibt.

РАСЧЁТ И ИЗМЕРЕНИЯ ТУРБУЛЕНТНОГО ОСЕСИММЕТРИЧНОГО ДИФФУЗИОННОГО ПЛАМЕНИ ГОРЕЛКИ

Аннотация — Исследовалась работа газовой горелки, изготовленной в виде трубки с осесимметричным дефлектором. С помощью заделанных внутрь трубки датчиков измерялись распределения скорости, температуры и локального эквивалентного соотношения компонент диффузионного пламени. В экспериментах использовалась горелка простой геометрии для облегчения расчётов методом конечных разностей по программе, предоставленной фирмой CHAM Ltd. и основанной на введении поправки к величине скорости, деленной на давление, и на модели турбулентности $k - \varepsilon$. Факел моделировался с помощью использования простой квадратично-волновой флуктуации компонент смеси. Эксперименты проводились при значениях числа Re (для холодного течения в трубке), равных $5 \rightarrow 8 \times 10^4$ и ϕ $0,5 \rightarrow 0,9$. Для измерений использовались сферический датчик с семью отверстиями, бусинковая термопара и простой датчик отбора газа. Численные расчёты среднего течения и смешения воздуха и топлива без учёта флуктуаций давали плохое совпадение по температурным полям. Учёт флуктуаций позволил получить значительно более точный расчёт пламени. Модель является удовлетворительной для инженерных расчётов. Однако необходимо разрешить еще ряд проблем, связанных как с погрешностями, вносимыми диффузией, так и с моделями турбулентности и длины смешения.