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Two-component laser anemometry measurements of non-reacting and reacting complex flows in a swirl-stabilized model combustor*

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Abstract. Simultaneous two-component velocity measurements are acquired in a model, complex flow swirl-stabilized combustor using a two-color laser anemometer. A time base computer interface enables the direct measurement of Reynolds stress ($\overline{u'w'}$) as well as mean and rms axial (u, u') and azimuthal (w, w') velocities. The peak value of the normalized Reynolds stress ($\overline{u'w'}/u_{rms}w_{rms}$) approaches 0.25 which is less than values (~ 0.40) obtained by others using indirect, non-simultaneous measurement methods in complex flows, but similar to a direct measurement in a dump combustor without swirl. Isotropy is satisfied except in regions of high unidimensional shear, and both turbulence intensity and normalized Reynolds stress are reduced in the absence of reaction. Relatively small-scale form intermittencies, associated with a fluctuation of the stagnation point and a precessing vortex core, serve to reduce the measured values of the normalized Reynolds stress at the centerline by increasing the apparent turbulence intensity. At an elevated fuel loading, a global-scale form intermittency is invoked and, while likely realistic relative to practical devices, may not be a viable condition for time-averaged calculations.

1 Introduction

Turbulent and swirling flows with recirculation are found in many engineering systems, notable of which are gas turbines, boilers, furnaces, and incinerators. The degradation of fuel quality, the interest in disposal of hazardous waste, and the requirement for enhanced fuel utilization are each demanding a greater understanding of the turbulent transport and mixing in this class of flows. To enhance the level of understanding, detailed measurements of the flow structure are required to develop physical insight and refine numerical codes.

The experimental data base for turbulent swirling flows was recently analyzed in three separate studies (Kenworthy et al. 1983; Srinivasan et al. 1983; Sturgess 1983). Two principal conclusions emerged. First, the availability of detailed and spatially-resolved flowfield data is limited.

Second, of the data sets available, few are provided for both non-reacting and reacting conditions, sufficient in detail, and simplistic in geometry to provide a meaningful challenge to numerical codes. Clearly identified is the need for detailed flow structure data including (1) spatially-resolved velocity fields, (2) turbulent structure including the nature of such correlations as $\overline{u'w'}$ and the validity of the often made assumption of isotropy, and (3) the impact of reaction on flowfield structure. The present paper addresses this need with the following objectives:

- (1) To establish the flow structure, including the spatial distribution of Reynolds stress ($\overline{u'w'}$), of a swirl-stabilized flow in a model laboratory combustor;
- (2) To establish the effect of reaction on the flow structure;
- (3) To examine the isotropy of the flow and, in particular, the turbulent stress at the interface of a swirling and non-swirling stream;
- (4) To examine the small-scale and global-scale dynamics associated with strongly swirl-stabilized, reacting and non-reacting flows.

The data were first presented by Brum and Samuelsen (1982a). The measurements have subsequently been reproduced and repeated at different times and by different personnel to bring the level of accuracy to the presently reported levels.

2 Experiment

2.1 Geometry

The model combustor configuration, presented in Fig. 1, features a swirl-stabilized recirculation zone and possesses the important features of practical combustors (e.g., swirl, high turbulence intensity, recirculation). The combustor was developed in a series of tests (Brum & Samuelsen 1982b) and consists of an 80 mm I.D. cylindrical stainless steel tube that extends 32 cm from the plane of the nozzle. Rectangular optical windows (25 mm $\times$ 306 mm) are mounted perpendicular to the horizontal plane on both

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Fig. 1 a–c. Combustor (Brum & Samuelsen 1982 b): **a** schematic, **b** photography (15 m/s, $\phi = 0.1$, reacting flow), **c** photography (15 m/s, $\phi = 0.2$, reacting flow)

sides of the combustor tube to provide clear, optical access for the laser anemometry measurements.

A set of swirl vanes (57 mm O.D.) is concentrically located within the tube around a 19 mm O.D. centrally positioned fuel delivery tube. (Geometrical details are provided in Fig. 4.) Dilution and swirl air are metered separately. The dilution air is introduced through flow straighteners in the outer annulus. The swirl air passes through swirl vanes which impart an angle of turn to the flow, 60° in the present study with 70% solidity. The swirl number obtained by integrating across the swirl vanes is 0.8; that obtained by integrating the total inlet air mass flux is 0.3. The combustor is operated at atmospheric pressure.

Fuel (propane for reacting cases; CO_2 for non-reacting cases) is introduced through a nozzle at the end of the central fuel delivery tube. A cone-annular gas injector (Figs. 1 a and 4 c), configured to emulate the directional

momentum flux of a hollow-cone liquid spray nozzle, is used for the present work. The exit planes of both the fuel nozzle and swirl vanes are set coincident to provide a clean, well-defined boundary condition to ease the application of numerical models. Time-averaged photographs of the combustor are shown for the reacting condition at the two overall equivalence ratios considered, $\phi = 0.1$ and $\phi = 0.2$, in Figs. 1 b and 1 c respectively.

2.2 Velocity measurements

Axial and azimuthal, mean and rms velocity measurements are made using the two-color laser anemometry (LA) system shown in Fig. 2. The beam from a 200 mW Argon-ion laser (Lexel Model 75) is collimated and passed through a prism to separate the various wavelengths. The two most intense beams, green (514 nm) and blue (488 nm), are each passed through a series of optics in

Fig. 2. Two-color laser anemometer

which they are polarized and split into two beams of equal intensity 50 mm apart. An upstream 40 MHz frequency shift (TSI model 915 Bragg Cell) is applied to one of each pair of beams in order to avoid directional ambiguity that would otherwise result from the highly turbulent recirculating flow. The four beams (blue pair in the vertical plane and green pair in the horizontal plane) are then focused through a 250 mm lens to a common point within the test section. This results in a set of perpendicular interference fringes spaced at $2.6 \mu\text{m}$ for the green beams (vertical fringes) and $2.5 \mu\text{m}$ for the blue beams (horizontal fringes) which are responsive to the axial and azimuthal velocity components respectively.

Receiving optics consist of a 120 mm lens focused onto a 0.25 mm diameter photomultiplier tube aperture (via and appropriate dichromate filter to selectively pass either the blue or green light). These optics are placed at an angle of 20° off direct forward scatter which results in a probe volume of 0.022 mm^3 and cross-sectional area perpendicular to the axis of measurement of 0.10 mm^2 . However, due to the requirement imposed by the processing electronics that both axial (u) and azimuthal (w) velocity components be obtained simultaneously, the effective probe cross-section is much less (approximately 0.03 mm^2). The transmitting and receiving optics are mounted on an optical bench capable of placing the measurement volume at points throughout the stationary combustor test section.

The advent of laser anemometry has facilitated the measurement of velocity in fluid flows and the applications to flames and combustors is now becoming common. However, care must be taken when applying LA to flows wherein more than one stream is introduced (e.g., fuel, swirl air, non-swirl air). All streams must be uniformly seeded to equal concentration levels to avoid statistical biasing toward one of the streams. In the present study, the main and fuel jet flows are seeded independently but to the same levels of concentration with $1 \mu\text{m}$ alumina particles using a liquid suspension atomization seeding technique (Ikioka et al. 1983). The seeding system allows

a

b

Fig. 3 a and b. Laser anemometer seeder (Ikioka et al. 1983): a atomizer pressure, b output rate

gross and fine control of the seed particle generation rate by varying the seed concentration in the liquid suspension and the atomizer supply pressure respectively (Fig. 3 a). The system produces a steady rate of seed particle generation with respect to time (Fig. 3 b).

2.3 Data acquisition

Signal validation is obtained using two counter processors (Macrodyne Model 2098). A special electronic interface, built to interface the output of the two counter process channels (u , w) directly to a DEC LSI 11/23 computer system, identifies whether or not the u and w events occur within a certain aperture time of each other. If so, they are considered simultaneous, stored and then multiplexed into

the 11/23 via a parallel interface. Once the interface verifies that the 11/23 has read the data, it simultaneously resets both processor channels. The key feature of this system is that it permits a direct and instantaneous measurement of correlations such as $\overline{u'w'}$. An aperture time of $50\text{ }\mu\text{s}$ was selected since, at the bulk reference velocity considered (15 m/s), an equivalent spatial resolution of less than 0.75 mm is obtained.

The 11/23 computer is equipped with an internal clock having a resolution of $100\text{ }\mu\text{s}$ which is initiated at the beginning of each run cycle. As the u, w data are received, the time of event t is combined with the raw data (u, w, t), and the data triplets are permanently stored in an archival fashion for future reference and analysis.

2.4 Accuracy

Sources of inaccuracy evaluated are (1) sampling error resulting from a finite number of samples, (2) positional accuracy of the traversing system, (3) digital resolution (i.e., the magnitude of the least significant bit output by the Macrodyne counter processors), and (4) velocity bias.

Five thousand (5,000) samples are taken at all measurement points. This number is required to obtain convergence of the velocity statistics and results (to a 95% confidence level) in a maximum sampling error of $\pm e_r$ on the mean velocity, where:

$$e_r = \frac{2(v_{\text{rms}})}{\sqrt{n}} \quad n = \text{number of samples.} \quad (1)$$

For the flows measured, this error is less than one percent of the bulk velocity. The translation of positional accuracy ($\pm 0.3\text{ mm}$) into velocity error depends upon the local gradients. For example, at the steepest axial gradient (2.5 m/sec/mm) for the baseline case (15 m/s , $\phi = 0.1$, reacting flow), the positional error translates into a velocity error of $\pm 0.75\text{ m/s}$. The resolution of the least significant bit is approximately 0.25 m/s resulting in a maximum bit resolution error of $\pm 0.125\text{ m/s}$. Finally, velocity bias is avoided by using a low seeding rate and thereby operating the counter processors in an unsaturated mode with sample times more than an order of magnitude above the flow correlation time (Edwards & Jensen 1983). The absence of velocity bias is verified by analysis of the time-marked archived data base in uniform time steps of differing intervals.

A detailed error analysis which considered all contributing sources reveals an overall accuracy of the mean and rms velocity statistics of $\pm 5\%$ and $\pm 10\%$ respectively. To demonstrate repeatability and reproducibility of the data, the test matrix was re-run three times, first after a three month interim period, second after a nine month period, and third after twenty-six months. All repeated measurements agreed with the original data to within the stated error limits. Since the $\overline{u'w'}$ correlation coefficient entails the multiplication of four rms type terms of $\pm 10\%$

accuracy each, its accuracy is expected to be ± 30 to $\pm 40\%$.

3 Results

Data are presented for one reference velocity (15 m/s) and two overall equivalence ratios ($\phi = 0.1$ and 0.2), for both reacting and non-reacting flows. The results are presented in graphical form in consideration of space and ease of presentation. Tabulated values of the data are available as well as detailed information on the geometry and operating conditions (Brum 1983).

3.1 Inlet conditions

Inlet velocity profiles are measured as closely to the upstream entry plane as possible. In particular, the inlet profiles for azimuthal velocity (w, w_{rms}) and axial velocity (u, u_{rms}) are measured at 1 mm and 5 mm downstream of the entry plane respectively, and are presented in Fig. 4 for the nonreacting and reacting cases at 15 m/s , $\phi = 0.1$.

3.2 Mean flow field

Radial profile plots of mean and rms axial (u, u_{rms}) and azimuthal (w, w_{rms}) velocities are presented in Fig. 5a for the baseline case (15 m/s , $\phi = 0.1$, reacting flow). Axial values appear on the top half of each radial profile plot and azimuthal values on the bottom. Radial profiles were

Fig. 4a – c. Inlet velocity profiles and geometry (15 m/s , $\phi = 0.1$): a reacting flow, b non-reacting flow, c inlet geometry

Fig. 5a–c. Results for baseline condition (15 m/s, $\phi = 0.1$, reacting flow): a radial profiles, b centerline profiles, c streamlines

measured at seven axial stations for each run condition; for clarity only four are shown in the figures. The axial (centerline) profile is presented in Fig. 5b.

The expansion of the swirling air forms a strong zone of backmixing extending 1.1 combustor duct diameters downstream of the nozzle. The introduction of swirling air internal to the outer concentric dilution air is a condition similar to that found in practical combustors and exemplary of flows having substantial streamline curvature. Examination of the axial velocity profiles at the inlet plane (Fig. 4) with the 2.0 and 7.0 cm stations (Fig. 5) reveals a rapid downstream shift of the peak axial velocity toward the outer wall. This shift cannot be attributed to turbulent mixing alone; for if it were, the outward shift of the azimuthal velocity would be equally rapid. Instead, the axial acceleration of the outer flow is a direct consequence of the radial pressure gradient imposed by the internal swirling flow.

In contrast to the outward movement of the mean axial velocity peak, the mean azimuthal velocity peak moves inward at downstream locations. This is attributed to (1) the mixing and dilution of the swirl velocity with the dilution stream, and (2) the swirling inlet air that flows initially out and around the recirculation zone and then collapses toward centerline downstream. Conservation of angular momentum in the radially inward flow is responsible for the peak at the inner radii.

At the 2.0 cm station, the azimuthal velocity profile has two peaks. Examination of the axial mean velocity at that station reveals that the inner azimuthal velocity peak is within the flow that is being convected upstream from

downstream locations having relatively high azimuthal velocities near centerline.

At the far downstream profile (24 cm), the axial velocities approach a turbulent pipe flow profile in contrast to the azimuthal velocity profile which exhibits little evidence for transition toward a solid body rotation. This difference is associated with the absence of a swirl component in the annular air stream at the inlet and the substantial inertia required, as a result, to induce rotation in the massive outer flow.

Using the continuity equation, stream functions ψ can be obtained by radially integrating the profiles of mean axial velocity:

$$\psi = \frac{\int_0^r r u dr}{\int_0^R r u dr} \quad (2)$$

where

R = outer radius of test section

r = radial location of interest.

Integration at each profile allows streamlines to be drawn through points of constant ψ . These streamlines, shown in Fig. 5c, illustrate the form of the “time-averaged” flow-field.

3.3 Turbulence field

Radial profiles of both axial and azimuthal root mean square (rms) velocity are also presented in Fig. 5a for the base line case. The peaks in the profiles occur in regions of maximum shear. However, isotropy is a reasonable engineering assumption for this case since w_{rms} is generally within 20% of u_{rms} with one notable exception: the region into which the non-swirling fuel jet issues (illustrated by the peak in the u velocity curve at the 2 cm station).

The turbulence intensity, as measured by a stationary LA probe, is comprised of “form intermittency” (i.e., periodic fluctuations of flowfield structure) in addition to the micro-scale fluctuations intrinsic to fluid mechanic turbulence (i.e., “true turbulence”). For the baseline condition, form intermittency contributes in two regions within the present flowfield: the recirculation zone and the centerline in the wake of the recirculation zone. Although visually stable to the eye, high speed movies reveal an oscillation wherein the luminous length of the recirculation zone experiences a $\pm 10\%$ length change at a rate of approximately 100 Hz. This small-scale form intermittency is manifested in a peak in the u_{rms} velocity at the stagnation point of the recirculation zone (Fig. 5b). Further downstream, the second example of form intermittency occurs. Examination of the radial plot of w_{rms} at the 24.0 axial station reveals an increase near and peak at the centerline. This excursion is produced by a small-scale precessing of the vortex core about the stationary LA probe. Similar excursions in turbulence production rates

Fig. 6 **a** and **b**. Fuel loading (15 m/s, reacting flow): **a** radial profiles, **b** centerline profiles

using LA have been measured by Owen (1976) in a laboratory combustor and Cheng (1984) in a premixed turbulent flame; the form intermittency in these two cases was associated with the periodic movement of the recirculation zone and the flame sheet respectively.

3.4 Fuel loading

An increase in fuel loading from $\phi = 0.1$ to $\phi = 0.2$ (Fig. 6) produces a shortened recirculation zone and a positive centerline velocity at the 1 cm station, the latter of which is evidence that a portion of the conical fuel jet is directed to the centerline for this condition.

Visually, the flame is stable at the elevated fuel loading. However, high speed movies of the flame reveal a global-scale oscillatory interaction between the fuel jet and the recirculation zone that does not exist at $\phi = 0.1$. The relatively higher fuel jet momentum intermittently establishes an off-axis recirculation zone between the jet and swirler (effecting a redirectioning of the fuel jet to the axis). Otherwise, the fuel jet diffuses into the on-axis recirculation zone dominated by the swirler. As further evidence of this global-scale fluctuation, high-speed movies establish that soot is formed along the centerline upon collapse of the jet, a portion of which is momentarily held at the stagnation point upon the establishment of the on-axis recirculation while the remaining portion of

the soot is recirculated, processed, and burned out within the recirculation zone. The soot held at the stagnation point is then emitted downstream as the on-axis stagnation region transitions to the off-axis zone.

This global-scale oscillatory behavior is not evident in the time-averaged mean velocities presented in Fig. 6, nor is it evident visually except by the presence of frequent but randomly spaced streaks of soot along the centerline in the wake of the recirculation zone. This global-scale form intermittency does manifest itself, similar to the smaller-scale form intermittency, as an additional turbulence intensity. Note that turbulence intensity levels in the vicinity of the recirculation zone are noticeably higher for case of the elevated fuel loading. Whereas the $\phi = 0.1$ case is an appropriate condition for the application and testing of time-averaged modeling, the dynamics of the $\phi = 0.2$ case, though consistent visually with a time-averaged perspective, is in fact dominated by transient, large scale fluctuations. Such gyrations are not necessarily unrealistic relative to practical devices. Evidence of global-scale fluctuations of this type, for example, has been observed in full-scale combustors (e.g., Dils 1973).

3.5 Heat release

Figure 7 illustrates the effect of reaction on the flow structure. Radial and centerline profiles of mean and rms axial and azimuthal velocities are presented for the non-reacting and reacting flows at 15 m/s, $\phi = 0.1$. It is evident that the recirculation zone in the case of reaction is (1) stronger (higher negative velocities), (2) more compact (shorter), and (3) radially wider. Root-mean-square axial velocity levels are approximately 50% higher for the reacting case. High speed flow visualization of the non-reacting case using sheet-lit neutrally buoyant bubbles reveals no evidence of global scale form intermittency. Hence, the non-reacting case is well suited for time-averaged modeling.

3.6 Reynolds stress

Radial plots of the normalized Reynolds stress ($C = \overline{u'w'}/u_{rms}w_{rms}$) appear in Fig. 8 (top and bottom) for the two fuel loadings and both the reacting and non-reacting cases along with the axial (u , top) and azimuthal (w , bottom) velocity profiles. Both the direction of the coefficient and the direction of local gradients change. For example, when both the u and w gradients with respect to r are positive:

$$\frac{\partial u}{\partial r} > 0, \quad \frac{\partial w}{\partial r} > 0$$

then the correlation (C) is positive, and when both are negative, the correlation (C) is positive. However, when the u gradient is positive and the w gradient is negative, or vice versa, the correlation is negative. In the areas where there is little or no gradient in either u or w , the correlation goes to zero.

Fig. 7 a and b. Non-reacting flow (15 m/s, $\phi = 0.1$): a radial profiles, b centerline profiles

Fig. 8 a and b. Normalized Reynolds stress (15 m/s): a $\phi = 0.1$, b $\phi = 0.2$

For well developed boundary layer flow, the coefficient (C) has been classically measured to be on the order of 0.40 (Liepmann 1979). In swirl flows, measurements of $\overline{u'w'}$ have been made using indirect methods under isothermal conditions. Janjua et al. (1982), for example, used a six orientation hot wire probe in an expanding swirl combustor and derived the Reynolds stress algebraically based on time-averaged quantities obtained at each orientation. The peak $\overline{u'w'}$ correlation coefficient calculated was 0.38. Fujii et al. (1981) used measurements of a single component LA system set at various angles to algebraically determine $\overline{u'w'}$ in a swirling jet. The peak coefficient calculated from their results was about 0.40. Using a direct method of measurement similar to that adopted for the present study, Smith (1983) obtained peak values of the $\overline{u'w'}$ correlation coefficient of 0.23 in a complex but non-swirling flow (dump combustor).

The peak value of the correlation coefficient measured via the direct simultaneous method used here is 0.20 for the $\phi = 0.1$ reacting flow case (Fig. 8 a), a value (1) less than that measured in boundary layers, (2) less than values obtained for indirect methods of measurement in

complex flows, but (3) similar to a value measured directly in a non-swirl, dump combustor. In the absence of reaction, the correlation coefficient is notably lower. The peak value of the correlation coefficient is clearly higher for the $\phi = 0.2$ case (Fig. 8 b), approaching 0.25 in reacting flow. In the absence of reaction, the coefficient is again lower.

Recalling that the correlation coefficient is normalized by the measured rms velocities and that, in measuring the rms velocity, the stationary LA probe is not capable of distinguishing between fluctuations arising from true turbulence and form intermittency, the presence of the latter (form intermittency) effectively dilutes the correlation produced by the first (true turbulence). In regions of smaller-scale form intermittency (e.g., oscillating stagnation point and precessing vortex core at $\phi = 0.1$), values of C are suppressed. At $\phi = 0.2$, the effect of the global-scale form intermittency is evident by the substantially suppressed values of the correlation coefficient at the centerline, especially within the zone of recirculation extending from the inlet plane to one duct diameter downstream of the nozzle.

4 Summary and conclusions

The flow structure, including the spatial distribution of normalized Reynolds stress, has been acquired in a model, swirl-stabilized combustor. A two-color laser anemometry with a direct, simultaneous measurement of orthogonal velocity components yielded peak correlation coefficients for the normalized Reynolds stress ($\overline{u'w'}/u_{rms}w_{rms}$) approaching 0.25 which is less than that reported (0.40) using indirect measurement methods on somewhat different complex flow geometries, but consistent with a direct measurement (0.23) in a complex flow without swirl. The absence of reaction notably reduced the magnitude of the normalized Reynolds stress, weakened the strength of the recirculation zone, and lengthened the axial extent of the recirculation zone.

"Form intermittency" was manifested in two forms: (1) small scale, and (2) global scale. Flowfield fluctuations for the reacting baseline and non-reacting cases (15 m/s, $\phi = 0.1$) are small-scale. As such, the conditions reported are attractive as test cases for time-averaged numerical codes. Except in a few regions of high unidimensional shear, $u-w$ isotropy is a reasonable engineering assumption for both the reacting and non-reacting flows. In contrast to the $\phi = 0.1$ fuel loading, flowfield fluctuations at an increased fuel loading ($\phi = 0.2$) are global in scale.

The conclusions deduced from this study are as follows:

1. The direct measurement of Reynolds stress is possible in swirl-stabilized flows. Peak values of the correlation coefficient in the present case varied, depending on the conditions, from 0.20 to approximately 0.25.

2. Form intermittency locally enhances rms velocity and locally suppresses the $\overline{u'w'}$ correlation coefficient. Care is required, as a result, in the interpretation of the velocity statistics. Some form of two-dimensional imaging such as flow visualization, planar laser measurements, and/or two-point measurements are appropriate to establish the regions in which form intermittency may be significant. In the present case, form intermittency is associated with an oscillation of the recirculation zone, and a precessing vortex core.

3. Two-dimensional imaging is also required to detect global-scale dynamic behavior which, although realistic, may produce conditions unsuitable for conventional time-averaged calculations. In the present case, an elevated fuel loading invokes a dynamic coupling between the fuel jet and the swirl-stabilized recirculation zone.

4. Reaction can have a strong impact on the flowfield structure. In the present case, reaction results in increased values of normalized Reynolds stress, an enhanced rate of backmixing, and a shorter and wider zone of recirculation.

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