Effect of boundary conditions on downstream vorticity from counter-rotating swirlers

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Abstract Particle image velocimetry (PIV) is utilized to measure the non-reacting flow field in a reflow combustor with multiple and single swirlers. The velocity field, vortex structure and total vorticity levels are experimentally obtained using two different boundary conditions, representing a single confined swirler and multiple swirlers in an annular combustor. The influence of the boundary conditions on the flow field at several locations downstream of the swirler is experimentally investigated, showing that the central vortex in the multi-swirler case is more concentrated than in the single-swirler case. The vorticity of the central vortex and average cross-sectional vorticity are relatively low at the swirler outlet in both cases. Both of these statistics gradually increase to the maximum values near 20 mm downstream of the swirler outlet, and subsequently decrease. It is also found that the central vortex in the multi-swirler case is consistently greater than the single-swirler case. These results demonstrate the critical influence of boundary conditions on flow characteristic of swirling flow, providing insight into the difference of the experiments on test-bed combustor and the full-scale annular combustors.

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1. Introduction

In aircraft engine combustors, a swirling flow is usually introduced to improve fuel mixing and flame stabilization aerodynamically. This swirling flow is established by introducing a tangential velocity to the main axial flow. When the swirl strength is greater than a critical value, the rearward force induced by pressure gradient exceeds the forward aerodynamic force leading to a central recirculation zone. Central recirculation zone plays an important role in flame stabilization as it can return part of burned gas to the outlet of swirler to reduce the flow velocity of air at the outlet of swirler to local flame propagation speed, meanwhile its position and size directly affects the residence time of liquid fuel vapors, which has a great influence on the generation of NOx. Consequently, the swirler directly affects combustor performance. In this paper, the cold flow field downstream of a swirler was experimentally investigated using the particle image velocimetry (PIV) technique to provide reference data and elucidate the impact of boundary conditions on the behavior exhibited by a given swirler design.
For the purposes of characterizing the flow field downstream of a swirler, PIV has several advantages over both traditional temperature- or pressure-based global measurements and other single-point laser diagnostic techniques such as laser Doppler velocimetry (LDV). In addition to its non-intrusive nature, PIV can provide a near-instantaneous view of a full 2D velocity field, without the directional ambiguity of Doppler-based measurement techniques. These advantages make it an excellent choice for providing both quantitative and qualitative insight into the nature of complicated flow fields. Furthermore, the quantitative measurement of complete velocity field provides the possibility for the measurement of vorticity field.

Due to the inherently 3D nature of swirling flow, the application of 2D PIV must be done with some care. In spite of the aforementioned complications, several studies have been completed to investigate the velocity vector field downstream of a single swirler. Reddy et al. utilized PIV to investigate the non-reacting swirling flow field characteristics of a swirler combustor. They obtained the variations in velocity, vorticity, Reynolds shearing stress, and turbulent intensity at various cross-sections downstream of the swirler and in the plane along the inlet flow direction, and analyzed the flow field characteristics of the central and corner recirculation zones. High levels of turbulence generated due to the swirling effect were noted in the study of Reddy et al., and such turbulence in turn promotes rapid mixing. Kim et al. utilized PIV to study downstream flowfield of swirler in a dump gas turbine combustor to provide new insights into the dynamics of turbulent swirl-stabilized flames, which are very important for the understanding of combustion instabilities. They observed that when combustion instability occurred, it was accompanied with the fluctuation of the recirculation zone. The fluctuation frequency of the recirculation zone was the same as the frequency of combustion instability. Gutmark et al. systematically studied the relationship of vortex breakdown and combustion instability and they found that the source of combustion instability was associated with vortex breakdown. The research showed that vortex located at downstream of swirler had a significant relationship with combustion stability.

Bourgouin et al. experimentally and numerically studied the impact of swirler structure on the central recirculation zone and the precessing vortex core (PVC) and they obtained that an increase in the swirler blade length could augment the maximum axial and azimuthal mean velocities downstream of the swirler and lead to an increase in the root mean square (RMS) velocity levels. With an increase in the swirl number, the central recirculation zone expanded in the transverse direction. The frequency of the PVC was also increased for the local rise of the mean azimuthal velocity. Moreover, the amplitude of the PVC was larger but vanished at a lower height in the combustion chamber.

Fureby et al. and Grinstein et al. experimentally and numerically investigated the non-reacting flow field of a single swirler in the free atmosphere and cylindrical flame tube using LDV, PIV, and flamelet-based large eddy simulation (LES). They obtained time-averaged velocity distributions and RMS turbulent velocities at several cross-sections downstream of the swirler. They obtained that in the confined domain case, the central recirculation zone was both longer and wider than in the open domain case. Fu et al. investigated the impact of confinement on the downstream flow field of a counter-rotating swirler installed in eight square box test sections with different widths reporting that increasing the level of confinement increased the complexity of the flow field and had a significant influence on the mean and turbulent structure downstream from the swirler. Ceglia et al. experimentally investigated the organization of the coherent structures arising within the near field of the swirling jet both in free and cylindrical confined configurations for water. They obtained that the confinement caused an increase of the swirl number and induced a larger spreading of the swirling jet promoting the enhancement of turbulence at the swirler exit.

Fanaca et al. experimentally investigated and compared the flow fields of a 12-swirler annular combustor and a single swirler combustor. A free swirling jet flow was noted to form in the annular combustor, while a swirling wall jet flow regime existed in the single burner configuration. They proposed a new correlation, which allowed estimating the swirling jet flow regime for co-swirling burners in an annular combustion chamber. With this information, the single burner tests can be designed to match the annular combustor flow regime.

Boutazakhti et al. utilized PIV and phase Doppler particle anemometry (PDPA) to map the velocity field downstream of a 3 x 3 square matrix of nine small swirlers, in addition to a single swirler configuration. The experimental results showed that in the merger region close to the swirlers, the characteristics of individual jets were still visible and the expansion rate of the central jet was slowed. In the developed region the cluster blended into a single jet-like flow with the axial component of the velocity field displaying self-similar properties.

Although the above investigations have characterized the flow field downstream of the swirlers in some burners in detail, no open literature has been published to quantitatively investigate the influence of varied boundary conditions on the velocity field and vortex structure along the flow direction for a fixed swirler configuration. Because the vortex structure has significant effects on the combustion performance of the burner, it is critical to understand how the imposition of various boundary conditions changes the overall vortex structure of a given swirler. The current work aims to highlight and quantify this impact with a simplified combustor geometry using PIV measurements to obtain the vorticity fields at several locations downstream of the swirler.

2. Experimental apparatus

2.1. Combustor configuration

To study the effect of boundary conditions on the vertical structures evolved from a given swirler, a three swirler combustor was designed, as shown in Fig. 1. Using this configuration, two different sets of boundary conditions can be studied. The first, utilizing all three swirlers, is similar in nature to the conditions found in an annular combustor where individual swirlers are allowed to interact. In this configuration, the section downstream of the central swirler is the region of interest, with the upper and lower boundary conditions being solid walls, as well as periodic free boundary conditions present at the left and right boundaries. The second configuration consists of a single swirler surrounded on each side by a solid wall. This condition is achieved by placing a solid boundary between Swirlers 1 and 2, isolating Swirler 2. These two cases are referred to hereafter as multi-swirler and single-swirler, respectively.
A Cartesian coordinate system was chosen with the origin at the center of the swirler outlet. The positive $y$-axis was in the flow direction, the positive $z$-axis was in the vertical direction, and the positive $x$-axis was determined by the right-hand rule from $y$- and $z$-axes. The flow field measurements were performed at six cross sections, namely $y = 10, 15, 20, 25, 30, 35$ mm. The primary hole, with a diameter of 3.6 mm, was located at $y = 38$ mm.

The swirler in the present investigation had a counter-rotating dual-swirler configuration, composed by the inner and outer swirlers. For incompressible flow, the swirl number of swirler, $S$, is defined as

\[
S = \frac{\int_0^R u w r^2 dr}{\int_0^R u^2 r dr}
\]

where $u$ is the axial velocity component of swirling jet, $w$ the tangential velocity component of swirling jet, and $R$ the radius of swirling region. The inner swirler, consisting of oblique holes to induct a velocity containing axial velocity component, had a swirl number and effective flow area of 0.91 and 37 mm$^2$, respectively and rotated in a right-handed fashion with respect to the positive $y$-axis. The outer swirler, utilizing radial blades to induct a velocity not containing axial velocity component, had a swirl number and effective flow area of 0.93 and 61.6 mm$^2$, respectively and rotated in the opposite direction to the inner swirler.

The temperature at the combustor inlet was an ambient temperature of 20 °C. The static pressure (total pressure) at the combustor inlet was at an ambient pressure of 114500 Pa. The pressure difference between the combustor inlet and outlet was measured by the U-tube manometer. For all the experiments conducted in this investigation, the pressure difference between the combustor inlet and outlet was kept constant at 3600 Pa (an approximation of the actual engine at idle state). The air mass flow rate, measured by the float type flow meter, was 0.04 kg/s, and the inlet Mach number was calculated as 0.06. (The inlet of air flow is rectangular, with a length of 228 mm and a width of 7.5 mm, see Fig. 1).

### 2.2. PIV system

The combustor system was installed on a displacement mechanism with 1D translational motion (see Fig. 2). This setup allows for the efficient and accurate translation of the laser sheet to various axial locations relative to the swirler exit. To ensure accuracy of the measurements, the optical arm and CCD camera were adjusted to introduce the laser sheet and collect reflected light perpendicular to combustor windows. Using linear scales inserted into the combustor, the appropriate scale factor was determined to convert the particle movement measured in pixels by the camera to the actual distance traveled. Since the combustor and the PIV system is not translated, the distance between the laser sheet and camera remains constant and only a single scale factor need be calculated. To avoid laser scattering into the CCD camera, which may reduce the signal-to-noise ratio, the combustor inner wall was coated with a light-absorbing black coat. The paper with coordinates located at the bottom of the combustor was used to display the distance of the laser sheet from the swirler exit. The PIV system is controlled by dynamic studio, provided by Dantec Dynamics.

Starch granules of 5 μm mean diameter and density of $2.5 \times 10^{-4}$ g/mm$^3$ were used as tracer particles, which had good tracking ability and reflective performance for the flow conditions investigated. The seeding density was set so that a
minimum of 10 particles were present inside the representative interrogation areas.

The laser pulse energy was approximately 650 MJ. The time delay between the two frames of the dual pulses was set for each axial location to ensure that the maximum displacement of particles within an interrogation area obeyed the 1/4 displacement principle. Based on this principle, the delay time can be set using

\[ \Delta t = \frac{d_1}{4MV_{\text{max}}} \]  

(2)

where \( d_1 \) is the grid size, \( M \) the scale factor of the image size to the actual spatial dimension and \( V_{\text{max}} \) the maximum velocity in the field of view. The corresponding laser pulse interval was calculated as \( \Delta t \approx 20 \mu s \). In practice, \( \Delta t \) was set within the range of 10–20 \( \mu s \).

2.3. Data processing

For each measuring section, 50 image pairs were time averaged using the post-processing program of the Dantec software to obtain the mean velocity field. The adaptive correlation technique was used over interrogation domains of the images. In the adaptive correlation, a much larger interrogation area is first applied and subsequently the resulting vector is used as a starting point for calculating the vector for a smaller interrogation area. The final interrogation area was set as \( 32 \times 32 \), with overlap as 50\% and interrogation area offset as “central difference”.

According to the vorticity formula, the time-averaged vorticity field can be calculated from the time-averaged velocity field. The \( y \)-axis component of vorticity is defined as

\[ \omega_y = \frac{\partial u}{\partial z} - \frac{\partial v}{\partial x} = \frac{\partial u}{\partial z} - \frac{\partial v}{\partial x} \]  

(3)

Using this formula, the vorticity field from the acquired velocity field is computed.

The measuring uncertainty of the PIV system included the bias uncertainty and precision uncertainty. In PIV, the velocity is determined as

\[ u' = \frac{\Delta L_O}{\Delta t} \]  

(4)

where \( u' \) is the velocity in a certain direction, \( \Delta L \) the particle displacement obtained in the cross-correlation algorithm, \( \Delta t \) the laser pulse interval, \( L_O \) the width of the camera view in the object plane, and \( L_I \) the width of the digital image. Therefore, the bias uncertainty of the velocity can be estimated by the following equation:

\[ B_M = \sqrt{ \left( \frac{\Delta L_O}{\Delta t} \right)^2 B_u^2 + \left( \frac{\Delta u'}{\Delta t} \right)^2 B_{\omega_y}^2 + \left( \frac{\Delta v'}{\Delta t} \right)^2 B_{\omega_z}^2 + \left( \frac{\Delta w'}{\Delta t} \right)^2 B_{\omega_x}^2 } \]  

(5)

Table 1 lists the typical values of the variables and their uncertainties for Eq. (5) associated with the current PIV setup.

According to the calculation by the data in the table, the bias uncertainty of velocity is 0.31 m/s.

Besides the bias uncertainty, the mean value has its precision uncertainty caused by the flow fluctuation and turbulence. For 95\% confidence, the precision uncertainty is expressed as

\[ \mu_{\text{stand dev}} = \frac{1.96 \mu_{\text{std}}}{\sqrt{N}} \]  

where \( N \) is the sample number of velocity measurements, \( \mu_{\text{std}} \) the standard deviation value in the time-averaged mean velocity, and 1.96 the 95\% confidence interval of standard normal distribution. In the present studies, each cross-section has \( N = 50 \) and the maximum standard deviation is close to 2 m/s. Consequently, the maximum precision uncertainty of the velocity is 0.55 m/s, and the total uncertainty is 0.86 m/s, the sum of the bias uncertainty and the precision uncertainty.

3. Test results

Figs. 3 and 4 show the time-averaged velocity and vorticity fields as a function of axial distance from the swirler exit. For the purposes of direct comparison, within each figure the color scale, representing the magnitude of the velocity or vorticity, is kept constant.

As is observed in Fig. 3, for both single- and multi-swirler cases a clear vortical structure is present near the swirler exit. In the case of the single-swirler, this structure appears to break down relatively more quickly than in the multi-swirler case, with the circular form of the velocity field breaking down by \( y = 20 \text{ mm} \). In comparison, this circular form is maintained all the way to \( y = 35 \text{ mm} \) for the multi-swirler configuration. This may suggest that the periodic boundary condition at the left and right boundaries for the multi-swirler configuration causes the vortex structure to remain coherent at a greater distance from the swirler as compared to the fully-confined single-swirler case. In addition, velocities are apparently higher within the core region of the vortex with the multi-swirler.

Fig. 4 further illustrates the nature of the vortex structure for each case by applying Eq. (3) to convert the velocity field to a vorticity field. For the sake of clarity, the field can be broken up into four general classifications. The vorticity located in the central area and less than \( -800 \text{ s}^{-1} \) can be defined as Vorticity group 1, representing the core of the vortex and generated by inner swirler. Group 2 is located in the periphery of the Group 1 and less than \( -800 \text{ s}^{-1} \), generated by outer swirler. The vorticity between Group 1 and Group 2 is larger than \( -800 \text{ s}^{-1} \), which is an interaction zone between the inner and outer swirlers. Group 3 is adjacent to the periphery of Group 2 and larger than \( 800 \text{ s}^{-1} \), representing the interaction zone between the outer swirler and the exterior, and Group 4 represents the exterior flow near the experimental boundaries. Comparison between the single- and multi-swirler cases indicates that at \( y = 10 \text{ mm} \) the vorticity in the single-swirler case is applied over a significantly larger diameter than that in the multi-swirler case. Whereas for the single-swirler an appreciable region of low vorticity exists between the core (Vorticity group 1) and the
next region of high vorticity (Vorticity group 2) at $y = 10$ mm, the multi-swirler does not exhibit this region, instead displaying a near-continuous region of high vorticity until it reaches the exterior region (Vorticity group 4). Indeed, overall the vorticity in the single-swirler case is distributed more widely across the region of interest for all axial positions. This may be explained by the relative lack of confinement in the multi-swirler case, where moving fluid at the left and right boundaries does not encounter an obstruction and thus does not generate the accompanying shear forces or recirculation zones. In contrast, the single-swirler may generate additional shear forces in the exterior region due to the presence of solid walls on all boundaries. Additional differences can be observed considering the axial development of the core region vorticity. Similarly to the patterns noted with respect to velocity, the core vorticity...
is maintained all the way to $y = 35$ mm for the multi-swirler, while the core is disrupted at $y = 25$ mm for the single-swirler and vortex structure disappears entirely at $y = 30$ mm. This observation suggests that the influence of increased confinement in the single-swirler case is to hasten – relative to the multi-swirler – the breakdown of coherent vortex structures created by the swirler. With respect to a realistic combustor, this would seem to indicate that data derived from a confined single swirler would predict a merger region closer to the swirler exit than would be predicted from data produced from a quasi-annular configuration.

The minimum vorticity in the region of Group 1 was set as the central vortex vorticity and the variation of the central vortex vorticity along flow pathline was investigated. Fig. 5 shows that in both the multi-swirler and single-swirler cases the vortex intensities (the absolute value of vorticity, $|W_y|$) of the
central vortex were relatively low at the swirler outlet, and gradually increased to the maximum values in the range of $y = 15–20$ mm downstream of the swirler outlet. That was because in both the multi-swirler and single-swirler case, vorticity induced by the inner and outer swirler was all in the negative $y$-axis direction, and the outer swirler induced vorticity was transported into the inner swirler induced vorticity region, which increased the vortex intensity of the latter from $y = 10$ mm to 20 mm. The vortex intensities of the central vortex in the multi-swirler case were always larger than

Fig. 4  Variation of vorticity field along flow direction.
that in the single-swirler case before the primary holes. The jets from the primary holes enhanced the vortex intensity of the central vortex, therefore the effect was more significant in the single-swirler case. The experimental results demonstrated that the solid wall boundary had a significant confinement effect on the original vorticity of the central vortex. The vorticity magnitude of the central vortex at the swirler outlet in the single-swirler case was approximately less than that in the multi-swirler by 10%. Downstream of $y = 20$ mm, the mean decay rate of vorticity in the single-swirler case (36.76 $1/s/mm$) was approximately 81.42% of that in the multi-swirler case (45.15 $1/s/mm$), illustrating that the transport rate of vorticity in the multi-swirler case was larger than that in the single-swirler case.

Fig. 6 shows that the variation of the cross-sectional averaged vorticity was similar to that of the vorticity of the central...
vorticity, which also demonstrates the confinement effect of the solid wall boundary on the original vorticity. At \( y = 10 \) mm, the cross-sectional averaged vorticity in the single-swirler case \((-38.589 \text{ s}^{-1})\) was approximately 39.60% of that in the multi-swirler case \((-97.441 \text{ s}^{-1})\). Furthermore, because no more vorticity was added into the flow in the range of \( y = 20-30 \) mm, the cross-sectional averaged vorticity can directly demonstrate the viscous dissipation on the vorticity and the transport rate from the \( y \)-component vorticity to other directions. The experimental data show that from \( y = 20 \) mm (corresponding to the maximum averaged vorticity) to \( y = 30 \) mm, the averaged decay rate of the vorticity in the single-swirler case \((0.496 \text{ 1/s/mm})\) was only 24.64% of that in the multi-swirler case \((2.013 \text{ 1/s/mm})\). Consequently, the vorticity dissipation and transport rate to other directions in the multi-swirler case were larger than those in the single-swirler case. Because the standard deviation of the data sample represents the uniformity of the data, the standard deviation of the vorticity in a cross-section represents the uniformity of that in the cross-section. The larger the standard deviation is, the less uniformity the vorticity is. In the cross-section, the mean vorticity intensity gradually increases from \( y = 10 \) mm to 20 mm, while gradually decreases from \( y = 20 \) mm to 30 mm. Therefore, the variation of the cross-sectional standard deviation of vorticity along the flow direction was divided into two parts in \( y = 20 \) mm. When \( y < 20 \) mm, the mean decay rate of the cross-sectional standard deviation of vorticity in the multi-swirler case (approximately 3.586 \text{ 1/s/mm}) is smaller than that in the single-swirler case (approximately 15.247 \text{ 1/s/mm}) (see Fig. 7). When \( y > 20 \) mm, the mean decay rate of the cross-sectional standard deviation of vorticity in the single-swirler case (approximately 4.895 \text{ 1/s/mm}) is only 31.04% of that in the multi-swirler case (approximately 15.711 \text{ 1/s/mm}). Downstream of \( y = 20 \) mm, no more vorticity is added into the flow. It is seen from Fig. 7 that the cross-sectional vorticity became uniform more quickly in the multi-swirler case, illustrating a larger transport rate of vorticity.

4. Conclusion

A PIV system was utilized to measure the velocity and vorticity components of the flow field generated by the multi-swirler and single-swirler, respectively. Based on the results presented in this paper, the influence of boundary conditions was investigated and some useful conclusions can be drawn as follows.

(1) The solid wall boundary affects the stability and intensity of the central vortex. In the case of the single-swirler, this vortex structure appeared to break down relatively more quickly than that in the multi-swirler case. This may suggest that the periodic boundary condition in the multi-swirler case causes the vortex structure to remain coherent at a greater distance from the swirler as compared to the fully-confined single-swirler case. Furthermore, overall the vorticity in the single-swirler case was distributed more widely across the region of interest for all the axial positions than that in the multi-swirler case. This may be explained by the relative lack of confinement in the multi-swirler case, where moving fluid at the left and right boundaries does not encounter an obstruction and thus does not generate the accompanying shear forces or recirculation zones. In contrast, the single-swirler may generate additional shear forces in the exterior region due to the presence of solid walls on all boundaries.

(2) In the multi-swirler and the single-swirler cases, the vortex intensities of the central vortex and the cross-sectional averaged vorticity were relatively low at the swirler outlet for the confinement effect of solid wall boundary on the original vorticity of the central vortex. At \( y = 10 \) mm downstream of the swirler outlet, the vortex intensities of the central vortex and the cross-sectional averaged vorticity were relatively low at the swirler outlet for the confinement effect of solid wall boundary on the original vorticity of the central vortex. Because the standard deviation of the data sample represents the uniformity of the data, the standard deviation of the vorticity in a cross-section represents the uniformity of that in the cross-section. The larger the standard deviation is, the less uniformity the vorticity is. In the cross-section, the mean vorticity intensity gradually increases from \( y = 10 \) mm to 20 mm, while gradually decreases from \( y = 20 \) mm to 30 mm. Therefore, the variation of the cross-sectional standard deviation of vorticity along the flow direction was divided into two parts in \( y = 20 \) mm. When \( y < 20 \) mm, the mean decay rate of the cross-sectional standard deviation of vorticity in the multi-swirler case (approximately 3.586 \text{ 1/s/mm}) is smaller than that in the single-swirler case (approximately 15.247 \text{ 1/s/mm}) (see Fig. 7). When \( y > 20 \) mm, the mean decay rate of the cross-sectional standard deviation of vorticity in the single-swirler case (approximately 4.895 \text{ 1/s/mm}) is only 31.04% of that in the multi-swirler case (approximately 15.711 \text{ 1/s/mm}). Downstream of \( y = 20 \) mm, no more vorticity is added into the flow. It is seen from Fig. 7 that the cross-sectional vorticity became uniform more quickly in the multi-swirler case, illustrating a larger transport rate of vorticity.

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In summary, the boundary conditions have a significant influence on the flowfield characteristics downstream of the swirler. In future experiment, it is proposed that the boundary condition of the model should keep consistent with that of the actual situation. If this condition cannot be attained, the experimental results should be corrected following certain rules.

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References


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