Numerical Simulation of Gas Turbine Swirl-Stabilized Injector Dynamics

Shanwu Wang*, Shih-Yang Hsieh†, Vigor Yang‡

The Pennsylvania State University
University Park, PA 16802, USA

Abstract

A comprehensive numerical analysis has been conducted to investigate the vortical flow dynamics and acoustic response of a gas-turbine swirl-stabilized injector. The theoretical formulation is based on the complete conservation equations of mass, momentum, and energy in three dimensions. Turbulence closure is achieved by means of the large-eddy-simulation (LES) technique. The compressible version of the Smagorinsky eddy-viscosity model is employed to describe the subgrid-scale turbulent motions and their effect on large-scale structures. The governing equations and the associated boundary conditions are solved by a finite-volume, Adam-Bashforth predictor-corrector scheme along with the implementation of the message passing interface (MPI) parallel computing architecture. Detailed flow structures are studied for two different swirl numbers. Results show that the internal flowfield in the injector is intrinsically unsteady and subject to shear and centrifugal instabilities. The unsteady flow evolution and vortex breakdown are clearly visualized and can be explained on theoretical bases. The unsteadiness may be related to periodic vortex shedding, vortex breakdown and breakup, mode competition, and other phenomena that are sensitive to the swirl number.

Introduction

The unsteady flow dynamic interaction with chemical reactions in combustion chamber has hindered the development of gas turbine engines for years. The resultant flow oscillations may reach certain level to interfere with proper engine operation and eventually may significantly shorten the engine lifetime. Although it is well known that combustion instability is a consequence of the transient response of flame dynamics to local flow oscillations in a confined volume, the triggering mechanism that leads to the onset of combustion instability is still unclear. Therefore, it is essential to specifically identify the most important physical processes for driving instabilities in order to establish a knowledge-based design methodology for successful development of gas turbine combustion systems.

Swirl-stabilized injectors have been commonly used in modern gas turbine engines as an aid to stabilize the high intensity combustion process and to promote efficient clean combustion. One of the most important flow characteristics produced by swirl-stabilized injectors is the central toroidal recirculation zone (CTRZ) [1], which serves as a flame stabilization mechanism. The flows in this region are generally associated with high shear rates and turbulent intensity resulting from vortex breakdown so that large-scale unsteady flow motions occur. The flow oscillations may couple resonantly with fundamental acoustic modes in the combustor, causing combustion instabilities. Since the flow fields generated by swirl-stabilized injectors play an important role in defining the stoichiometry and fluid dynamics of the primary combustion zone, it is thus necessary to investigate the flow response and acoustic coupling in the injectors for diagnosing the cause of the instability. The purpose of this work is to conduct a comprehensive numerical investigation into the detailed flow structures and the effects of the swirl number on the stability of a swirl-stabilized injector.
Theoretical and Numerical Framework

Governing equations

The basic concept of LES is that the large-scale turbulent structure is directly computed and small dissipative structure is modeled. Mathematically, the LES methodology begins with filtering of small-scale effects from large-scale motions in the full conservation equations. A filtered (or resolved) variable, denoted by an overbar, is defined as

$$\bar{\phi}(x) = \frac{1}{\Omega} \int_G \phi(x') G(x,x') dx'$$ \hspace{1cm} (1)

where $\Omega$ is the entire domain and $G$ is the filter function, which determines the size and structure of the small scales. To account for variable density effects, the filtering operation given by Eq. (1) is augmented with Favre decompositions of the form

$$\bar{\rho} = \rho + \rho^*, \quad \bar{\rho} = \bar{\rho}/\bar{\rho}$$ \hspace{1cm} (2)

After applying the filtering operation to the instantaneous governing equations, one obtains the filtered equations of motion to be solved in large-eddy simulations. The Favre-filtered conservation equations of mass, momentum, and energy can be expressed in the following conservative forms:

$$\frac{\partial \bar{\rho}}{\partial t} + \frac{\partial (\bar{\rho} \bar{u}_i)}{\partial x_j} = 0$$ \hspace{1cm} (3)

$$\frac{\partial \bar{\rho} \bar{u}_i}{\partial t} + \frac{\partial (\bar{\rho} \bar{u}_i \bar{u}_j + \bar{\rho} \delta_{ij})}{\partial x_j} = \frac{\partial (\bar{f}_{ij} - \tau_{ij}^{\text{SGS}})}{\partial x_j}$$ \hspace{1cm} (4)

$$\frac{\partial \bar{\rho} \bar{e}_i}{\partial t} + \frac{\partial (\bar{\rho} \bar{e}_i \bar{u}_j + \bar{\rho} \delta_{ij})}{\partial x_j} = \frac{\partial}{\partial x_j} (\bar{u}_i \bar{f}_{ij} + \bar{q}_j - H_j^{\text{SGS}})$$ \hspace{1cm} (5)

where

$$\bar{f}_{ij} = -\frac{2}{3} \frac{\partial \bar{u}_i}{\partial x_i} \delta_{ij} + \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i}$$

$$\bar{s}_{ij} = \bar{s} + 1/2 \bar{u}_j \bar{u}_j, \quad \bar{c} = \bar{c} = \bar{h} - \bar{\rho} / \bar{\rho}$$

$$\bar{q}_j = \lambda (\partial \bar{T} / \partial x_j)$$

In the above equations, $\rho$, $u_i$, $p$, $T$, $e_0$, $h$, $\mu$, and $\lambda$ represent the density, velocity components, pressure, temperature, specific total energy, specific enthalpy, viscosity, and thermal conductivity, respectively, and $\tau$ represents the viscous stress tensor. The unclosed subgrid-scale terms in Eqs. (3)-(5) are the subgrid stresses $\tau_{ij}^{\text{SGS}}$ and subgrid energy fluxes $H_j^{\text{SGS}}$.

Subgrid-scale model

Turbulence closure of the filtered governing equations can be achieved with the implementation of an appropriate subgrid-scale model. The Favre-averaged generalization of the Smagorinsky model proposed by Erlebacher et al.\cite{2} is employed in the present study. The subgrid stress terms and respective counterparts in the energy equation are modeled using the Smagorinsky eddy-viscosity model for compressible flows coupled with a gradient transport model to simulate $\text{sgs}$ energy transport. The subgrid stress $\tau_{ij}^{\text{SGS}}$ is modeled as follows:

$$\tau_{ij}^{\text{SGS}} = -\frac{2 C_R}{3} \bar{\rho} \Delta^2 \left( \bar{s}_{ij} - \frac{\delta_{ij}}{3} \bar{s} \right)$$ \hspace{1cm} (6)

$$\tau_{kk}^{\text{SGS}} = 2 C_I \bar{\rho} \Delta^2 \bar{s}_{ij} \bar{s}_{ij}$$ \hspace{1cm} (7)

where

$$\bar{s}_{ij} = \frac{1}{2} \left( \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right), \quad \left| \bar{s} \right| = (2 \bar{s}_{ij} \bar{s}_{ij})^{1/2}$$

Here, the dimensionless quantities $C_R$ and $C_I$ represent the compressible Smagorinsky constants, with $\Delta$ being the filter width. The subgrid energy flux term $H_j^{\text{SGS}}$ is modeled as

$$H_j^{\text{SGS}} = -\bar{\rho} \frac{C_R}{Pr_t} \Delta^2 \left[ \left( \frac{\partial \bar{h}}{\partial x_j} + \bar{u}_i \frac{\partial \bar{u}_i}{\partial x_j} \right) \Delta^2 \right]$$ \hspace{1cm} (8)

where the constant $Pr_t$ represents the turbulent Prandtl number. The constants $C_R = 0.012$ and $C_I = 0.0066$ are adopted in the current work. The value of 0.5 is employed for $Pr_t$.

Numerical method

The theoretical formulation outlined above is solved numerically by means of a density-based, finite-volume methodology. The spatial discretization employs fourth-order and second-order, central-differencing schemes for convective terms and viscous terms, respectively, in generalized
coordinates. Temporal discretization is obtained using a second-order, four-stage Runge-Kutta integration method. Further efficiency is obtained by implementing an MPI (Message Passing Interface) parallel computing architecture with a multi-block domain composition technique.

**Results and Discussion**

The physical model includes the internal flowfield in a swirl-stabilized injector. The external region downstream of the injector is also considered to provide a complete description of the flow development. The injector [3] consists of a mixing duct and a fuel nozzle located coaxially upstream of the mixing duct, as shown in Fig. 1. The mixing duct includes a center cylindrical duct, two annular ducts, and three passages corresponding to the three ducts, respectively. Those passages are spaced radially outward from each other. Three sets of air swirlers, denoted as $S_1$, $S_2$, and $S_3$, respectively, are located upstream of the air-passage, and the first and second swirlers are counter-rotating relative to the x-axis. The mixing duct measures a length of 28 mm, and the first duct outlet has a diameter of 32 mm. The computational domain is carefully chosen such that the outer boundaries in the $x$ and $r$ directions are sufficiently far enough from the injector exit to minimize the propagation of boundary-induced disturbances into the computational domain. The entire grid system has 1.9 million points, of which 0.9 million points are located within the injector. A total number of 54 computational blocks are used. The baseline flow condition in the current study includes an ambient pressure of 1 atm, an inlet temperature of 293 K, and an inlet mass flow rate of 0.077 kg/s. The corresponding Reynolds number based on the diameter and average velocity at the injector outlet is $2 \times 10^5$. Calculations are conducted for two different sets of swirl vane angles. The low swirl-number case has swirl vane angles of $S_1 = 30^\circ$, $S_2 = -45^\circ$, and $S_3 = 50^\circ$, and the high swirl-number case has $S_1 = 50^\circ$, $S_2 = -60^\circ$, and $S_3 = 70^\circ$.

Figures 3 and 4 show the snapshots of the vorticity magnitude contours on the $x$-$y$ and $y$-$z$ planes for the low and high swirl-number cases, respectively. The flow patterns exhibit two common features as follows. The first is the vortex breakdown due to the radial-entry swirling flows. A CTRZ is found downstream of the center body due to this vortex breakdown. Because of the strong shear layer between the inlet passage and CTRZ, a strong vorticity layer is produced, which subsequently rolls, tilts, stretches, and breaks up into small vorticity bulbs. These vorticity bulbs are then convected downstream and interact with the surrounding flow structures. The evolution of these vortex structures may serve as a source of low-frequency flow oscillations. Figure 5 presents the power spectral density (PSD) of pressure oscillations at the injector exit for both cases, showing a rather broadband behavior. From the experiments conducted by Cohen and Hibshman [4], it was found that a swirl-stabilized injector closely similar to those examined in this study would cause preferential amplification and instability coupling of pressure fluctuations in the range of 400-600 Hz. This finding, however, was not observed herein probably because no external forcing was imposed at the inlet in the present calculations as opposed to the experiments. Nonetheless, the flowfields tend to exhibit low-frequency oscillations when the swirl number increases, as shown in Fig. 5 for the high swirl number case. Further study of the dynamical behavior of the injector and its response to externally imposed acoustic forcing is currently underway.

The second phenomenon is the high-frequency vortex shedding from the tip of the guide vane between the first and second passages, arising from the Kelvin-Helmholtz instability. Because of the opposition of the swirler vane angles, two highly swirled counter-rotating flows with different velocities, joining at the rim tip, produce a strong shear layer that promotes mixing process. The flow at the rim tip act as a ‘vortex source’, i.e., vorticies are generated and shed downstream sequentially with alternating direction, similar to those produced at the trailing edge of an airfoil at a high angle of attack. The ensuing influence on the fuel/air mixing may be significant because the strong vortical flow will interact with the thin fuel film on the surface of the guide vane between the second and third passages.

Figures 3 and 4 also present the mode-competition between two major dynamical phenomena—vortex breakup in the CTRZ and vortex shedding near the guide vane. For low swirl-number flows, the two phenomena coexist well and the vortex shedding phenomenon is more organized, and several well-defined harmonics in the pressure-frequency spectrum are observed in Fig. 5. When the swirl strength increases, the interactions between the shed vortices and CTRZ flow become stronger, and the mode-competition appears. For high swirl flows, the CTRZ size and strength increase, and the vortex streets originating from the tips of the guide vanes are suppressed and become disordered in the downstream. In addition, the low-frequency content of the pressure spectrum increases significantly. The vorticity fields on the $y$-$z$ plane in Figs. 3 and 4 show the asymmetrical flow oscillations in the azimuthal
direction, further revealing the complicated flow structures.

Figure 6 shows the contours of mean axial velocity superimposed on the streamline patterns on the x-y plane. A CTRZ, where the mean axial velocity is negative near the center line, is clearly observed in both cases. The size and the onset location of the CTRZ, however, are quite different. In swirling flows, an adverse pressure gradient in the axial location is required to balance the centrifugal force. When the swirl strength is large enough, the low-pressure core near the x-axis may induce vortex breakdown and result in a CTRZ. In the high swirl-number case, because the azimuthal velocity is higher than that in the low swirl-number case, a larger low-pressure region is required to balance the centrifugal force. This explains that the size of the CTRZ in the high swirl-number case is much larger than that with the small swirl number. It is also observed in Fig. 6 that the highest axial velocity for the low swirl-number case occurs near the center body of the fuel nozzle, while in the same region the axial velocity is negative for the high swirl-number case.

The contours of the mean turbulent kinetic energy for both cases are presented in Fig. 7. Two high turbulent-kinetic-energy (tke) regions are observed in regions downstream of the center body and the first guide vane, where the vortex breakdown and vortex shedding occur. The strong vortical motions promote locally the mixing between the fuel and air. The two high tke regions almost merge together at the exit of the injector for the high swirl-number case, but there still exists a gap for the low swirl-number case.

**Conclusions**

A comprehensive numerical analysis has been conducted to investigate the vortical flow dynamics of a swirl-stabilized injector. The formulation treats the unsteady, three-dimensional conservation equations, with turbulence closure achieved using the large eddy simulation technique. Results show that the swirl-stabilized injector considered in the present study is effective in mixing the fuel and air. Various unsteady flow characteristics, such as vortex shedding and vortex breakdown as well as their interactions, are investigated in depth for cases with different swirl numbers.

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**References**

Fig. 1 Schematic diagram of swirl-stabilized injector

Fig. 2 Grid system

Fig. 3 Snapshots of vorticity contours on x-y and y-z planes, low swirl number
Fig. 4   Snapshots of vorticity contours on $x$-$y$ and $y$-$z$ planes, high swirl number

Fig. 5   Power spectral density: a) low swirl number and b) high swirl number
Fig. 6 Contours of mean axial velocity and streamline pattern: a) low swirl number and b) high swirl number

Fig. 7 Contours of turbulence kinetic energy: a) low swirl number and b) high swirl number