EFFECT OF WALL HEAT TRANSFER ON THE FLAME RESPONSE TO ACOUSTIC PERTURBATION IN A TURBULENT SWIRLING COMBUSTOR

Yu Xia, Davide Laera and Aimee S. Morgans
Department of Mechanical Engineering, Imperial College London, London SW7 2AZ, U.K.
Email: d.laera@imperial.ac.uk

This article presents a numerical study of the forced flame response to acoustic perturbation in a longitudinal turbulent swirling combustor taken from an annular rig built at Cambridge University. Incompressible large eddy simulations (LES) are performed via the open source CFD-toolbox OpenFOAM, applying the partially-stirred reactor (PaSR) combustion model and a one-step global ethylene ($\text{C}_2\text{H}_4$)-air reaction scheme. A lean premixed ethylene-air flame stabilised by a bluff-body is studied in the combustor, which has a long longitudinal extension and attaches to the combustor side walls. The unforced flame is firstly simulated using uniform inflow, for which the adiabatic and non-adiabatic wall conditions are respectively applied, showing that the existence of wall heat transfer will increase the flame’s longitudinal length due to heat exchange on the side walls and locally quench the flame root in the outer shear layer of the bluff-body due to the heat losses on the flame holder wall. The unforced flame is then submitted to an upstream longitudinal harmonic velocity perturbation at combustor inlet, which has a forcing frequency varying from 500 Hz to 1,000 Hz at a forcing amplitude of 10% of the mean inflow velocity. The response of flame heat release rate is then computed over time for each frequency, eventually leading to the construction of flame transfer function (FTF) that relates the linear flame response to the upstream perturbation. The simulated FTFs are compared between adiabatic and non-adiabatic wall conditions, showing that the wall heat transfer largely decreases the FTF-gain, possibly due to the oscillation of flame surface area being reduced by the heat losses on the side walls. In contrast, the heat losses will increase the absolute FTF-phase, which is because the time delay between upstream perturbation and flame response becomes larger due to the extended flame length. This study highlights the importance of correctly accounting for the wall heat transfer in order to accurately predict the flame response to acoustic perturbations in a turbulent swirling combustor.

Keywords: Flame transfer function, Non-adiabatic wall condition, Incompressible LES, Ethylene-air flame, Turbulent swirling combustor

1. Introduction

Thermoacoustic instability is an undesirable phenomenon often occurring in lean premixed gas turbine combustors. It is caused by a two-way interaction between the acoustic pressure perturbation ($p'$) and the flame heat release rate response ($\dot{q}'$), which can lead to irreversible damage to the combustion system (e.g., fatigue failure, excessive heat, noise pollution, etc.). The available prediction approaches (e.g., [1,2,3]) for thermoacoustic instability all need an accurate model describing the flame response to upstream acoustic perturbation, such as the linear “flame transfer function (FTF)” [4]:

$$\mathcal{F}(\omega) = \frac{\dot{q}/\bar{\dot{q}}}{\bar{u}/\bar{u}_1} = G(\omega) \exp(i\varphi(\omega)), \quad (1)$$
where $\bar{\cdot}$ denotes fluctuation in frequency domain and $\overline{\cdot}$ is the time-averaged value. $\dot{q}$ is the volumetric heat release rate and $u_1$ is the velocity perturbation upstream to the flame. $G$ and $\varphi$ are gain and phase of the FTF, both of which are the function of the perturbation frequency ($\omega = 2\pi f$). Multiple numerical FTFs are available for different combustors (e.g., [5, 6, 7]), many of which are simulated by incompressible large eddy simulations (LES) (e.g., [5, 8]), which exploit the fact that the flame primarily responds to hydrodynamic disturbances excited by acoustics. However, most previous FTF simulations (e.g., [6, 9, 8]) only apply adiabatic wall conditions for all the solid boundaries of the computational domain, which may be acceptable when the flame is short and far away from the wall. However, when the flame is longer and attaching to the walls, which is common for turbulent non-swirling (e.g., [10]) and swirling flames (e.g., [11]), the wall heat transfer and accompanied heat losses may play an important role in the flame shape and the forced flame response to upstream perturbations. Only very few works have studied the impact of wall heat transfer on the flame response. For example, Kedia et al. [12] studied the linear response and the FTF of a laminar premixed CH$_4$-air flame under an upstream velocity perturbation, and found that the wall heat transfer would increase/decrease the gain of FTF at low/high perturbation frequency. In contrast, the heat transfer through the wall will increase the absolute FTF-phase over the entire frequency domain. Mejia et al. [13] then investigated the effect of wall-temperature on the response of a non-swirling open laminar flame. They found that the anchoring point of the flame root was modified by changing the temperature on the flame-holding wall, which leads to the change of flame response. The increase of the wall temperature will decrease/increase the gain of FTF at low/high frequencies, but not highly affect the distribution of FTF-phase. These studies, however, did not consider the impact of wall heat transfer on the response of a turbulent swirling flame in a real gas turbine combustor. This work aims to fill this gap by studying a lean premixed turbulent swirling ethylene-air flame, using the adiabatic and non-adiabatic wall conditions respectively.

2. Experimental Set-up and Numerical Framework

In the present work, the annular combustor rig built at Cambridge University (see Fig. 1(a)) is considered, which has been experimentally tested to study the azimuthal instabilities of premixed flames [11, 14, 15]. The entire rig contains an upstream plenum, multiple burners and a large combustion chamber. Since all the burners are identical, only one burner is adequate for longitudinal thermoacoustic prediction, and is used in this work as the computational domain for LES (see Fig. 1(b)). This single burner contains an upstream cylindrical tube (diameter: $D_t = 18.9$ mm) with a conical bluff body inside (diameter of exit plane $d_b = 13$ mm), and a downstream combustion chamber. A
six-vane swirler with a vane angle of 60° is positioned 8.4 mm upstream of the bluff body exit (more details can be found in Ref. [14]). The bulk flow velocity at the bluff body exit is measured to be 18.0 m/s [14], giving a Reynolds number of $1.5 \times 10^4$ based on the exit diameter, $d_b$. For the burner’s combustion chamber, it contains a rectangular section of width $H = 42$ mm equal to $2.2d_b$, chosen to have the same flame-to-wall distance as in the annular rig [11]. The height of combustion chamber is $L = 50$ mm, giving a dimension ratio of $L/D_t \approx 2.5$, which has the same order as in another similar combustor studied in Ref. [16]. Fully-premixed ethylene ($\text{C}_2\text{H}_4$)-air mixture enters the domain through its inlet with a constant equivalent ratio (denoted $\phi$) of 0.70. A mesh sensitivity analysis has been performed on this domain, and an optimal mesh with $\approx$9 million hexahedral structured cells is used to achieve the balance between simulation accuracy and computational cost. A constant inlet flow velocity of $u_{in} = 10.22$ m/s is defined at the domain inlet, in order to achieve the same velocity at bluff body exit as in experiments [14]. The zero-gradient pressure and convective outflow velocity conditions are both applied at the domain outlet. For all the solid boundaries, the non-slip wall condition is used, with adiabatic (Case A) and non-adiabatic (Case B) thermal wall conditions used respectively (see Figs. 3–4).

The low Mach number ReactingFOAM LES solver taken from the open-source CFD toolbox OpenFOAM is used by the present simulations. The incompressible approximation ($\left(\partial \rho / \partial p\right)_T = 0$) is used for the flow-field with the density ($\rho$) assumed as a function of temperature ($T$) only and not relevant to the pressure ($p$) — the same assumption has been used in previous LES studies of perturbed and unperturbed turbulent flames [5, 9]. The reactive conservation equations are Favre-filtered based on the assumption of ideal gas mixture with laminar viscosity (modelled by Sutherland’s law), Fourier heat conduction and Fickian diffusion laws, etc. In order to complete the filtered equations, the constant Smagorinsky model [17] is used to model the sub-grid Reynolds stresses, with the Van Driest damping function [5] applied near the walls to reduce the over-estimated turbulent viscosity. The Partially-Stirred Reactor (PaSR) combustion model [18] is used to solve the filtered chemical reaction rates in order to predict the turbulence-chemistry interaction. This PaSR model is developed based on the finite rate chemistry theory, which involves a reduced chemical reaction scheme. For the lean combustion regime (i.e. $\phi < 1$), the impact of reduced chemistry was found to be small on the global flame dynamics [19], therefore a global one-step ethylene ($\text{C}_2\text{H}_4$)-air reaction scheme is used in this work with 5 intermediate species [20]. The system of LES equations are then discretised in OpenFOAM by a finite volume method (FVM), based on the Gaussian theorem together with a semi-implicit time-integration scheme. The PIMPLE method is applied for the coupling between pressure and velocity. The convection and divergence terms are spatially discretised using a 2nd-order central difference scheme, and the unsteady terms are discretised by the 2nd-order Crank-Nicolson temporal scheme. A fixed computational time-step of $1.5 \times 10^{-6}$ s is applied, in order to maintain the simulation accuracy without any divergence.

3. Effect of Wall Heat Transfer on the Unforced Flame

The unforced flame shape simulated by LES using adiabatic wall conditions is firstly shown in Fig. 2. It is noted that the 3-D flame shape (see Fig. 2(a)) is generally axial-symmetric, with the flame tips touching all the four rigid side walls in the combustion chamber. The 2-D flame shape on a $y$-$z$ plane (see Fig. 2(b)) shows the ring-style flame surfaces in the inner and outer shear layers.

The unforced simulations using different wall conditions are then compared in Figs. 3–4. Case A (see Fig. 3(a), 4(a)) applies adiabatic condition for all the solid boundaries, while Case B (Fig. 3(b), 4(b)) uses the non-adiabatic wall condition with a varying wall temperature, which linearly increases from 500 K on the flame-holder wall to 1,500 K on the top-half of the combustor side walls. The change of wall temperature only occurs within the first 25 mm in the longitudinal ($x$) direction on all of the side walls. This wall temperature profile is chosen based on the relevant experimental measurements.

The 2-D contours of mean axial velocity ($\overline{u}$) fields are then compared between the adiabatic
Figure 2: Unforced flame shape in the form of (a) 3-D iso-surface of mean heat release rate in two directions and (b) contour of mean heat release rate ($\dot{q}$) on an $x$-$y$ 2-D cutting-plane, simulated with adiabatic wall conditions.

Case A and non-adiabatic Case B on two cutting-planes, $x$-$z$ and $x$-$y$, with the iso-curves of mean heat release rate superimposed. As shown in Fig. 3, the two flow-fields are generally similar in both directions, where higher velocity magnitudes are observed within the flame heat release zone, and a large longitudinal flow recirculation zone exists between the two flame branches. Both the inner and outer shear layers are clearly observed close to the exit of the bluff body, each stabilising a flame root. For the non-adiabatic case (see Fig. 3(b)), the lengths of the central recirculation zone and the flame are both longer, possibly due to the heat exchange between the flame and the side walls.

For the mean temperature ($T$) and heat release rate ($\dot{q}$) fields (see Fig. 4), the two cases are also generally similar in both $y$ (Fig. 4-I) and $z$ (Fig. 4-II) directions, being consistent with the comparison of mean velocity fields (see Fig. 3). Compared to the adiabatic Case A (see Fig. 4(a,c)), the case with wall heat losses (i.e. Case B) has a considerably longer flame length due to the large heat exchange on the side walls (e.g., see Fig. 4(d)). The magnitude of heat release rate from the bottom-half of the flame is highly decreased for Case B, mainly due to the heat losses on the flame holder wall (see Fig. 4(b)). The flame root in the outer shear layer is even almost quenched (see Fig. 4(b)-II), although the flame root in the inner shear layer is not highly affected by the wall heat transfer. These observations are all consistent with the experimental measurements.

4. Effect of Wall Heat Transfer on the Forced Flame

Based on above-validated unforced flame simulations, the flame transfer function (FTF) defined in Eq. 1 is computed for both cases imposing a longitudinal harmonic velocity perturbation ($u$) upstream to the flame at the domain inlet as:

$$u = \bar{u}_{in} \left[ 1 + A \sin (2\pi ft) \right],$$

where $f$ is the perturbation frequency and $A = |u'/\bar{u}_{in}|$ is the reduced perturbation amplitude at the domain inlet. In this work the value of $A$ is fixed at 0.1. The flame heat release rate responses forced at frequency $f = 500, 800$ and $1,000$ Hz are respectively calculated at each time-step for $\sim 15 - 20$ forcing cycles (after the initial transients vanish), and then spatially-integrated to obtain the gain ($G$) and phase ($\phi$) of the FTF. Table 1 compares the FTF-gains and phases for Case A and Case B. It is noted that for all perturbation frequencies, the existence of heat transfer through the flame-holder and side walls (i.e. Case B) could largely decrease the magnitude of FTF’s gain, and the gain’s differences between Case A and B are around $\sim 30\% - 60\%$. In contrast, the absolute value of FTF’s phase is
Figure 3: 2-D contours of mean axial velocity ($\bar{u}$) on (I) $x$-$z$ and (II) $x$-$y$ cutting-planes for (a) adiabatic Case A and (b) non-adiabatic Case B with the wall temperature linearly increasing from 500 K (on the flame-holder wall) to 1,500 K (on the top-half of the side walls). The change of wall temperature only occurs within the first 25 mm distance in the $x$-direction on all the side walls.

highly increased by the wall heat losses, with the difference between two cases being as high as $\sim 80\% - 300\%$. The reason for the decrease of gain may be due to the reduction of the flame surface area’s oscillation at non-adiabatic conditions, while the increase of the FTF-phase’s magnitude could be caused by the increase of the flame longitudinal length (see Fig. 4(b,d)), which adds the time delay between the upstream perturbation and the downstream flame response.

5. Conclusions and Future Work

This work numerically studies the impact of wall heat transfer on the flame response to acoustic perturbation in a turbulent swirling longitudinal combustor. A premixed ethylene ($C_2H_4$)-air flame stabilised by a bluff-body is simulated by incompressible OpenFOAM-LES in a single combustor, where the adiabatic and non-adiabatic thermal wall conditions are applied respectively. For the unforced simulations, the flame surrounded by adiabatic walls is relatively shorter and has two clear
flame roots within the inner and outer shear layers, respectively, in the downstream wake of the bluff-body. Under the non-adiabatic walls with wall heat transfer, however, the flame length becomes much longer and its root located in the outer shear layer is almost quenched. This axial extension of the flame length is due to the large amount of heat exchange on the combustor side walls, while the local quenching of the outer flame root is mainly caused by the heat losses on the flame holder wall.

For the forced flame simulations, the gain and phase of the flame transfer function (FTF) are both largely affected by the choice of the wall conditions. The FTF-gain is considerably decreased when the wall heat transfer exists, possibly due to the reduction of the flame surface area oscillation. In contrast, the absolute FTF-phase is significantly increased by the non-adiabatic walls, resulted from the extension of the flame length which increases the time-delay between the upstream perturbation and downstream flame response. Such differences in the FTF confirm the importance of correctly accounting for the wall heat transfer using appropriate wall temperatures, in order to accurately predict the flame response to acoustic perturbation in a turbulent swirling combustor. Future work will focus on the effect of wall heat transfer on the nonlinear flame response using the flame describing function (FDF) [21] at multiple upstream perturbation amplitudes, and a more detailed 2-step chemical scheme for the ethylene-air reaction will also be tested.
Table 1: FTF-gain ($G$) and phase ($\varphi$) for Case A and Case B at forcing amplitude of $A = 0.1$.

<table>
<thead>
<tr>
<th>FTF-gain $G$ [-]</th>
<th>$G_A$</th>
<th>$G_B$</th>
<th>$\frac{G_B - G_A}{G_A} \times 100%$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$f = 500$ Hz</td>
<td>2.886</td>
<td>2.016</td>
<td>-30.1%</td>
</tr>
<tr>
<td>$f = 800$ Hz</td>
<td>0.494</td>
<td>0.249</td>
<td>-49.6%</td>
</tr>
<tr>
<td>$f = 1,000$ Hz</td>
<td>1.440</td>
<td>0.600</td>
<td>-58.3%</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>FTF-phase $\varphi$ [rad]</th>
<th>$\varphi_A$</th>
<th>$\varphi_B$</th>
<th>$\frac{\varphi_B - \varphi_A}{\varphi_A} \times 100%$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$f = 500$ Hz</td>
<td>-0.853$\pi$</td>
<td>-1.564$\pi$</td>
<td>83.4%</td>
</tr>
<tr>
<td>$f = 800$ Hz</td>
<td>-0.408$\pi$</td>
<td>-1.232$\pi$</td>
<td>202.0%</td>
</tr>
<tr>
<td>$f = 1,000$ Hz</td>
<td>-0.536$\pi$</td>
<td>-2.194$\pi$</td>
<td>309.3%</td>
</tr>
</tbody>
</table>

Acknowledgement

This work is funded by ERC Starting Grant (grant No: 305410) ACOULOMODE (2013-2018), Siemens Industrial Turbomachinery Ltd., EPSRC Centre for Doctoral Training in Fluid Dynamics across Scales and the Department of Mechanical Engineering at Imperial College London. Access to HPC facilities at Imperial College and via the UK’s ARCHER are also acknowledged. We are also grateful to Prof. James Dawson and Prof. Nicholas Worth at NTNU for providing the combustor geometry and experimental data used in the present study.

REFERENCES


