The interaction of bluff body and swirl induced recirculations on the flow, mixing and combustion performance of stratified propane flames

E. Dogkas[1], P. Koutmos[1]

1. *Laboratory of Applied Thermodynamics, Department of Mechanical Engineering and Aeronautics, University of Patras, Patras, 26504, Greece*

Introduction

Pollutions regulations and the need for increased fuel consumption efficiency are imposing stricter requirements on high intensity industrial combustion systems. In particular, the need to cut energy consumption costs, the restrictions on NOx and soot emissions and the competitive placement in the market have encouraged the introduction of innovative solutions for better combustion control and the extrapolation of new concepts to practical designs [1–3]. The present work investigates the characteristics of partially premixed or stratified flames, established by staged fuel-air premixing in a multi cavity arrangement, formed along three concentric disks and stabilized in the vortex region of the afterbody [4].

Experimental Methodology

The studied burner geometry has been described in part [4,5] and are shown in Figures 1a, b, c and d. An axisymmetric, double-cavity premixer, formed along three concentric disks promotes propane-air premixing and supplies the combustion zone at the afterbody disk recirculation with a radial equivalence ratio gradient. The burner assemblies are operated with a swirl co-flow to study the interaction of the recirculating stratified flame with the surrounding swirl. [4,5]. Ignition and flame stabilization was achieved at the afterbody recirculation. A rectangle confinement was used to house the central bluff body/swirl burner. The Reynolds number, based on the afterbody diameter and the central air supply velocity (Uc), was maintained at 8000.

corresponding author: Dogkas Eleftherios, lefterisdogas@gmail.com

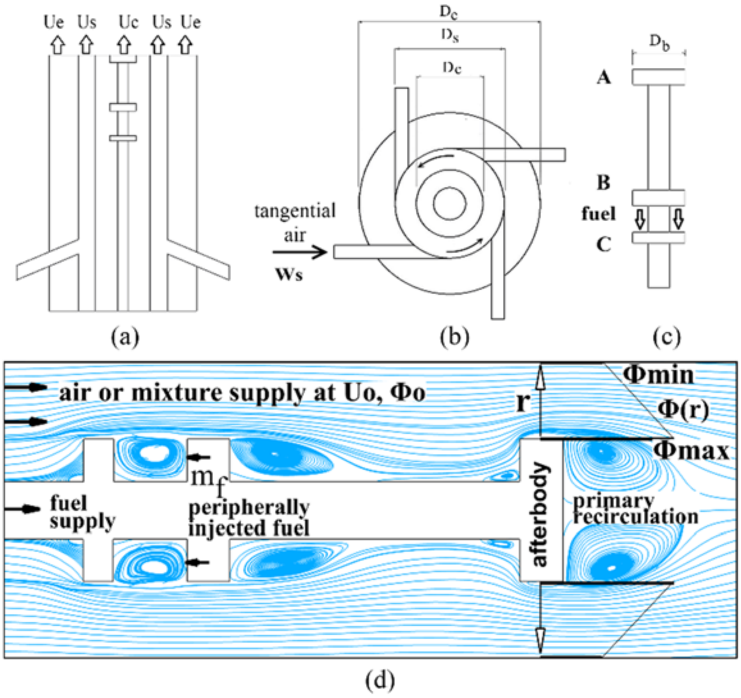

Figure 1: Experimental rig

The swirl number was defined as the ratio of the axial to the circumferential velocity at the surrounding co-flow outlet. Both velocities were evaluated by integration of the corresponding laser Doppler velocimetry (LDV) profiles at the exit plane of the surrounding swirl tube. Temperatures were measured with Pt-Pt/10%Rhuncoated beaded thermocouples of 50 to 75-μm-diameter wire.

Global exhaust emissions could also be measured by extracting flue gases and measuring bulk species (NO_x, CO, CO_2, O_2, and C_xH_y) concentrations with a Kane-May KM9106 Quintox flue gas analyzer further downstream of the burner. The emission indices of UHC, CO, CO_2 and the combustion efficiencies were computed from the following equations [6]:

$$\mathrm{EI}_z = \left(\frac{[z]}{[CO]+[CO_2]+[C_X H_Y]} \right) * \left(\frac{10^3 MW_Z}{M_C + a * M_H} \right)$$

where a is the hydrogen / carbon ratio of the fuel

$$\eta_C = 1 - \left(\frac{\mathrm{EI}_{UHC} + 0.232 * \mathrm{EI}_{CO}}{1000} \right)$$

Results and Discussion

The investigated cases were regulated at swirl level of 0.00 and 0.800. Each swirl case was operated with stratification levels that were 3 and 25% from Lean Blow off (LBO). Flames are discussed on the basis of inlet swirl intensity (S0.00 and S0.80) and the proximity to LBO through the parameter, $\delta = (m_{Fuel} - m_{Fuel,\,LBO})/m_{Fuel,\,LBO}$ (%) (FBO: fuel flow at blow off). This allows for a more practical comparison of their performance within their effective operating envelope [5].

Figure 2 displays mean, centerline temperature distributions for the investigated cases for S0.00 and S0.80. The maximum temperatures for the non-swirl case were about Tmax =

1611 K and Tmax= 1838.5 K for δ 3% and δ 25% respectively. Here the temperature increased with the increase of δ. For S0.80 the maximum mean, centerline temperature distributions had different behavior. Particular, for both δ's the maximum mean temperatures were the same level. Tmax was 1623.6K and 1656.9K for δ 3% and 25%. This different behavior was caused by the influence of the intense swirl number. The strong swirled case had different flame topology in contrast with the non-swirl case. The S0.00 flame is of cylindrical shape with a increased length. In contrast the S0.80 is seen to spread the toroidal flame sheet, shorten its overall length and displace its leading edge further off the disk face.

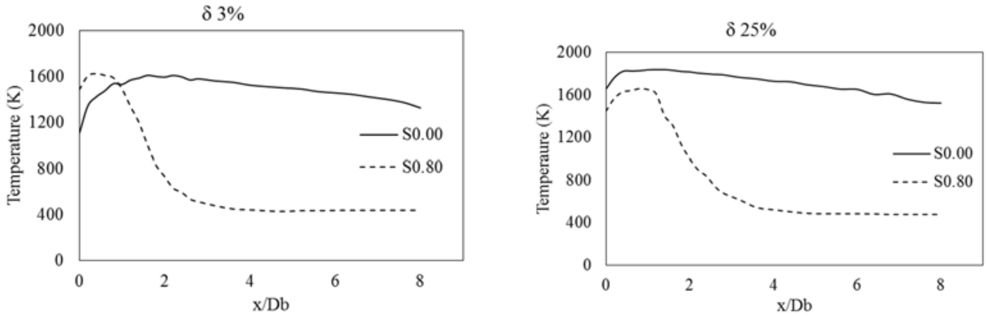

Figure 2: Mean, centerline temperature distributions for the investigated cases

In Figure 3, we observe the mean radial temperature traverses at three stations throughout the flame region. The axial positions are shown in terms of both the distance from the exit plane (x/Db) and the distance (x_f) that corresponds to the different detachment of the flame onset position in each individual nozzle.

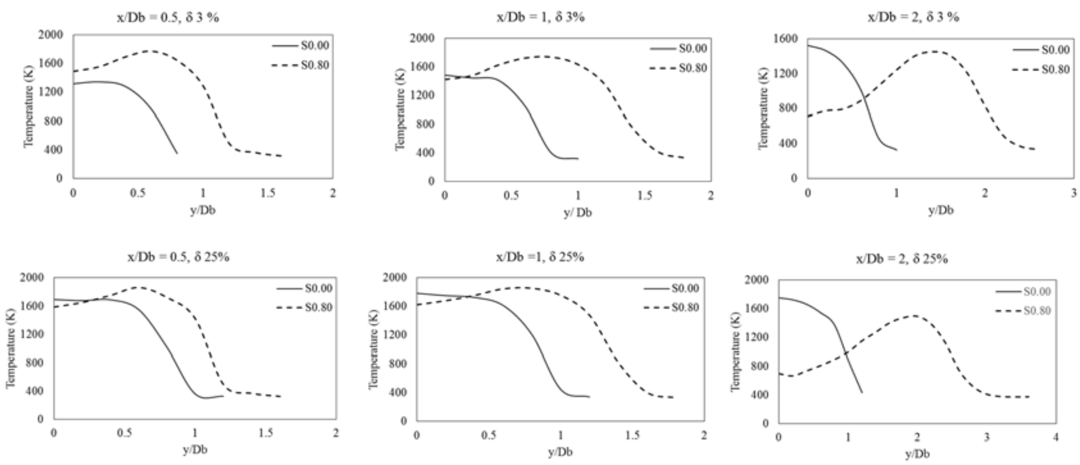

Figure 3: Mean radial temperature traverses at three stations (x/Db 0.5, 1, 2) throughout the flame region

Figures 4a, b, c and d display the variations of the global exhaust emission indices of the unburned hydrocarbons (UHC), CO_2, and CO along with the respective combustion efficiencies (η_c) for each investigated set up.

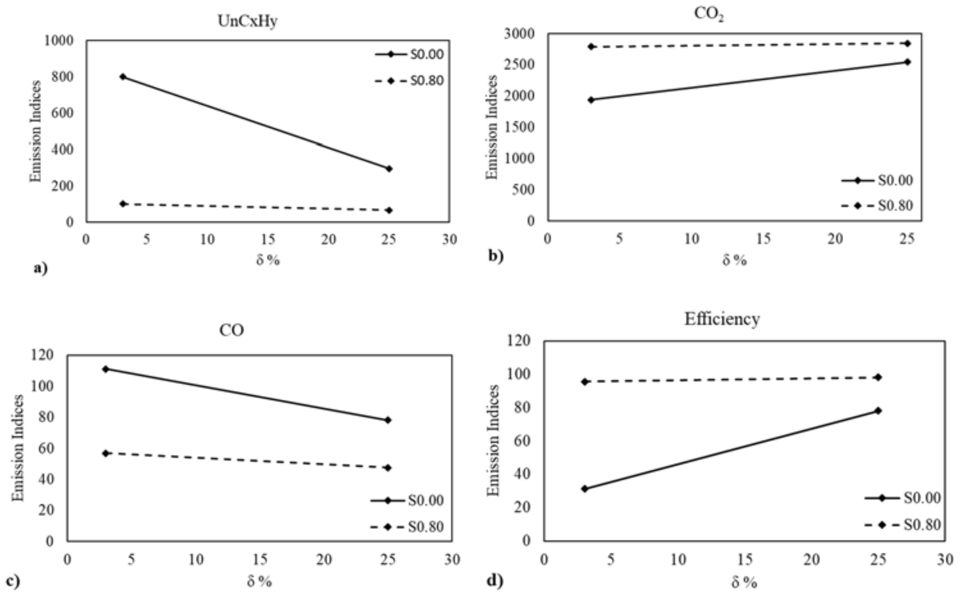

Figure 4a), b), c) and d): Emission indices of the unburned hydrocarbons (UHC), CO_2, and CO along with the respective combustion efficiencies (η_c) for each investigated set up.

The figure 4a, c suggest that the rapidly increase of the swirl number reduce the emissions of unburned hydrocarbon and CO. Moreover the two emissions reduced due to increase of the fuel. This is more obvious in the S0.00 case. On the other hand the emission indices for the CO_2 exhibits different behavior. The increase of either the swirl number or the fuel, led to the increase of the CO_2 emissions. Finally figure 4d suggested that the strong swirl intensity increased the efficiency of the system. The efficiency also increased from δ 3% to δ 25%, especially for the non-swirl flame. Further results will presented in the full paper.

References

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