Towards Ultra-Low Emission Distributed Combustion with Fuel Air Dilution

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Abstract

Colorless Distributed Combustion (CDC) has been shown to offer enhanced combustor performance for stationary gas turbine application with near zero emissions, high combustion intensity and efficiency, thermal field uniformity, and enhanced stability. Mixture preparation to form hot and low oxygen concentration environment paves the path to achieve CDC conditions. In this paper, a new approach of air dilution in partially premixed combustion conditions is employed and the results compared to premixed and non-premixed injection of air and fuel. Portion of the fuel is introduced in the air stream and portion of the air is introduced in the fuel stream such that the local equivalence ratios for each stream is well outside the flammability limit to eliminate flashback and instabilities. The experimental data demonstrated ultra-low emissions with this injection scheme. At equivalence ratio of 0.6, NO emission was 63% lower than non-premixed combustion mode. Also NO emission was similar to the premixed combustion with the advantage of eliminating flashback and flame instabilities that often prevail in premixed combustion conditions. Dilution provided 50% CO reduction as compared to non-premixed combustion. Numerical simulations, validated through Particle Image Velocimetry, were performed to outline the mixing process in each of the three cases. The methane mixture fraction prior to ignition, determined numerically, was found to be one half of that for the non-premixed case and close to that of the premixed case. This enhanced mixture preparation, associated with the new air and fuel dilution technique, resulted in reduced emission. Also the jet momentum ratio (between both streams) is enhanced, mainly due to the air addition to the fuel stream, to result in better mixing and a better reaction distribution for ultra-low emissions. Further reduction of NO_x is expected with improved distributed combustion condition.

Keywords: Air dilution, Colorless distributed combustion (CDC), Ultra-low NO_x and CO emission, Gas turbine combustion, Air-fuel mixing.

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1. Introduction

Gas turbines continue to receive increased attention as a prime source for clean electric energy production through combined cycles or through combined heat and power production. This is attributed to their high efficiency and new discovery of natural gas and shale gas resources in the states, fostering domestic energy and energy security. In addition, gas turbine is extensively used in aviation, both civilian and military, with increased interest in lowering emissions and biofuel adoption. This has motivated combustion engineers to develop new and novel combustion methods that comply with increasingly stringent regulations while reducing energy consumption. Reducing pollutants emission (such as, NO_x , CO, unburned hydrocarbons and soot) and increasing performance of gas turbine combustors continues to be of significant importance for environmental friendliness and energy conservation. The thermal field uniformity at the combustor exit (pattern factor) and within the combustor in gas turbines engines needs to be enhanced with the direct benefits of reducing pollutants emission (NO_x), preventing local burnout of combustors, reducing air cooling requirements for turbine blades, and increasing their lifetime. The thermal field uniformity reduces hot spot zones in the combustor, reduces cooling air requirements of turbine blades so that one can increase engine efficiency from firing at higher temperatures.

Colorless distributed combustion (CDC), which shares similar fundamental principles of high temperature air combustion (HiTAC [1]), has been investigated previously and proven itself as a novel option for near zero emissions of NO_x and CO [2-5]. CDC also offers significantly improved pattern factor, stable combustion, alleviation of combustion instabilities, low pressure drop and low noise emission for gas turbine combustion applications. This combustion method is

named “colorless” as it results in negligible visible signatures from the flames as compared to conventional flames in gas turbine combustors.

Colorless distributed combustion have been shown to provide significant improvement in pattern factor, low sound emission levels and ultra-low emissions of NO_x and CO under different flow configurations conditions [2-5], using a wide variety of fuels that include gas, liquid and bio-fuels [6-8]. The critical requirement to achieve colorless distributed combustion is controlled and rapid mixing between the combustion air and/or fuel stream with hot reactive species from within the combustor so as to form a high temperature low oxygen concentration mixture. Forming this mixture forms the basis for seeking distributed combustion condition as they foster distributed reactions, wherein the reaction zone occupies a larger volume with a lower reactivity due to the lower oxygen concentration. This low reaction rate is made possible through the high temperature of reactants (due to the inclusion of hot reactive species), leading to lower temperature rise [1]. Such distributed reaction combustion results in a large reaction zone volume with low reaction rate to result in low overall temperature rise in the combustor. This is in contrast to traditional (normal, or high oxygen concentration air) combustion that results in sharp temperature rise in the concentrated thin reaction flame front. Such thin reaction zones results in hot spot zones that foster thermal NO_x emissions produced from the Zeldovich thermal mechanism [9], which is the major NO_x contributor in gas turbine engines.

The importance of mixture preparation with hot and reactive fuel-air mixture stream cannot be overstated. Such mixture requires controlled entrainment of hot reactive combustion gases into the reactants. The recirculation of hot reactive species into the fresh stream can be achieved through proper design of the combustor flowfield. Entrainment of hot reactive species have been studied under different configurations, including swirling and non-swirling flow configurations

[4, 10]. Swirling configuration, wherein air was injected tangentially and product gases exited axially, showed higher recirculation ratios compared to non-swirling configurations under isothermal conditions [10]. Such increased recirculation resulted in lower pollutants emission and enhanced OH* chemiluminescence distribution within the combustor [4].

Enhanced hot reactive species recirculation and entrainment through tangential air injection provides only part of the mixing. Fuel introduction still needs to be addressed. Different fuel injection scenarios have been examined including premixed, non-premixed and coaxial-partially premixed. Different separation distances between air and fuel injection locations have been examined [2, 4], with focus on pollutants emissions and reaction distribution. In another study, premixed and non-premixed injection was compared to coaxial injection with focus on ignition time. Early injection and ignition was to be avoided as the air, fuel, and hot reactive gases were not well mixed yet. On the other hand, late injection leads to spontaneous ignition of the fuel before being mixed with the hot and diluted oxidizer due to the high temperature, a condition that should be avoided [11].

To enhance mixing, air and fuel dilution with flue gases have been studied with emphasis on the effect of air dilution versus fuel dilution on nitric oxide emissions [12]. It was concluded that fuel dilution has the greater effectiveness as compared to air dilution in practical applications due to the enhanced turbulent mixing and heat transfer. Dilution of fuel with inert gases has been investigated for combustor operation with ultra-low emission. Nitrogen addition has been shown to decrease NO emissions by 50% to achieve emissions of 2 ppm of NO and 12 ppm of CO [13]. The same trend was observed with air and carbon dioxide dilution [6]. The decrease in pollutants emission is attributed to the fuel jet having higher jet momentum (due to the added diluents).

Also, the added mass acts as a thermal sink that lowers the overall combustor temperature which significantly lowers thermal NO_x through Zeldovich mechanism [9].

Fuel mixing have been extensively studied under various configurations ranging from premixed to non-premixed fuel injection, including partially premixed injection. Traditionally premixed combustion offered lower emissions as compared to non-premixed conditions. The impact of unmixedness or non-uniformities in premixed combustion have been extensively studied. In one of the studies, NO_x emissions nearly doubled with change from premixed to partially premixed configuration [14]. Also, NO_x emissions depended heavily on the mixing length and air/fuel jet Reynolds numbers. Mongia *et al.* [15] reported a dramatic increase of NO_x production with incomplete mixing even at lean conditions. NO_x increased 4 folds when the combustor operated with 0.0085 unmixedness (for premixed stream, unmixedness is 0, while for a non-premixed stream, this value is 1) [15]. Non-premixed combustion has been investigated in detail as well with the goal of lowering emissions and enhance mixing. One common practice used to enhance mixing, is to create recirculation and stabilize combustion using swirl flow that entrains and recirculates a portion of the hot combustion products back to the root of the flame. For such combustors, swirl characteristics play a major role in mixing and combustion [16-19]. The impact of mixing on NO_x emissions in non-premixed configuration was provided with emissions reduced up to one sixth through enhanced swirler and mixing design [19].

In this paper, partially premixing of air and fuel is sought as a means to seek ultra-low emissions without any combustion instabilities. Previous work with fuel dilution have resulted in lower emissions and enhanced mixing between air and diluted fuel in non-premixed configuration [6]. To reap the benefits of this dilution effect, portion of the air is added to the fuel stream (thus diluting it) and portion of the fuel is supplied to the air stream. The amount of

air added to the fuel stream (and amount of fuel added to the air stream) will result in a mixed stream that is well outside the fuel flammability limits, resulting in the elimination of any flame flashback or instabilities. Through portion of the air mixed with fuel and portion of the fuel mixed with air will result in an overall enhanced mixing when these two streams interact in the combustor.

Previously, different air/fuel dilution scenarios were examined, where NO emissions were shown to be as low as premixed combustion case [20]. Note that in these dilution cases, the mixture formed (Fuel added to Air, or Air added to Fuel) had an equivalence ratio that is well outside the fuel flammability limits. Consequently no flashback is possible. However, the study was limited to a single equivalence ratio (0.8) on an ultra-high intensity combustor for aviation application [20]. One of the air dilution cases (case 2 [20]) is further investigated here in detail to evaluate its performance at different equivalence ratios. This investigation is performed using a swirling CDC combustor [4] with view to evaluate the beneficial aspects of air dilution as a mixing scheme under conditions that is more relevant to stationary gas turbine applications.

2. Approach

The performance of new mixing scheme was investigated both experimentally and numerically. Detailed of the experimental facility and the numerical simulations performed are given below.

2.1. *Experimental*

Experiments were performed using a swirling test combustor. This combustor have previously been studied for emissions, flow and thermal fields [4, 10]. Figure 1 shows the combustor examined. Air and fuel injection were modified to handle the fuel and air addition to the fresh

stream, respectively. The main air stream was preheated to 600K while the fuel temperature was 300K at injection. Air was injected at elevated temperature to simulate air heating in compressor in a gas turbine engine. A schematic of the facility with injection schemes is shown in Fig. 2.

The air and fuel (methane) flow rates were measured using choked flow orifice systems. Precision stainless steel orifices were used to choke the flow of gases with upstream pressure controlled to supply the required mass flow rate of gases. The upstream pressure was controlled using pressure regulators to maintain a steady pressure. The upstream pressure for both air and fuel was maintained between ± 0.2 psi of the desired pressure. This resulted in a flow rate accuracy of about 1%. For all experiments, the pressure downstream of the orifices was also measured to verify that choking condition existed for proper flow control at elevated pressure operation.

Detailed investigations on the overall emissions from the combustor as well as visible emissions have been performed at various experimental conditions. NO concentration was measured using a NO–NO_x chemiluminescent gas analyzer, CO concentration using the non-dispersive infrared method and O₂ concentration (used to correct the NO and CO emissions at standard 15% oxygen concentration) using galvanic cell method. At any experimental condition, the measurements were repeated at least three times and the uncertainty was estimated to be about ± 0.5 PPM for NO and $\pm 10\%$ for CO emission. For imaging the OH* chemiluminescent intensity distribution, an ICCD camera coupled to a narrow band filter having center wavelength of 313 nm wavelength was used. The resulting OH* intensity distributions were then normalized to the same scale.

2.2. Numerical

Numerical simulations were performed to determine the mixing behavior under different conditions. Since emphasis here is on the mixing prior to ignition, all simulations were performed under isothermal conditions as these conditions were validated against the experimental data. For these simulations, commercial software FLUENT[®] code was used. The flow was modeled using finite volume method. For solving turbulence, Reynolds Stress Model (RSM) was used with standard wall function. For all numerical models the volume consisted of about 3.5 million tetrahedral mesh cells.

The model boundary conditions used were as follows. For air inlet, “mass flow inlet” boundary condition was used with inlet turbulence intensity of 20%, the mass fraction of methane and oxygen was varied according to the case investigated. For product gases to exit, “pressure outlet” boundary condition was used, with zero-gauge back pressure and 10% turbulence intensity for back flow. For all simulations, the conversion criterion was that the largest normalized residual was less than 4E-04. The initial turbulence intensity was chosen to match values experimentally measured for this combustor geometry [10]. Note that the simulated flowfield and mixing behavior will deviate from the actual case; however, the trends are expected to remain similar, thus providing an indication for the causes of pollutants emission behavior for the various cases investigated here.

3. Conditions Reported

The combustor was operated at a heat load range of 3.9-6.25 kW over a range of equivalence ratios from 0.5-0.8, using methane as the fuel. Three cases presented here include fully premixed case (ATP), non-premixed case (ATF1), and the new dilution case. Table 1 gives the flowrates

for each injection location in standard liters per minute (SLPM) and the local equivalence ratio variation for each stream, as well as the global equivalence ratio.

In the dilution case, air and fuel flow rates are split into two streams. The main “top stream” injects most of the air flow, while the other “cross stream” injects most of the fuel. Fuel is added to the “top stream” and air is added to the “cross stream” such that the equivalence ratio for each streams remains well outside of the flammability limits for methane-air combustion [21]. Such mixing method is expected to yield favorable results as the mixing required inside the combustor is less compared to the non-premixed combustion case. The cross flow equivalence ratio was kept at 5. The top flow equivalence ratio varied from 0.2 to 0.12. Both air and fuel flow rates were varied to furnish the combustor with the appropriate air and fuel flows thus allowing the combustor to operate in the desired range of equivalence ratios.

Adding air to the fuel stream will increase the jet momentum of the cross flow. Increased jet momentum will enhance the penetration of the jet in cross flow to result in better mixing. The higher cross stream jet momentum (or the lower jet momentum ratio) is expected to significantly enhance the mixing which also affects the turbulence in this region.

4. Results and Discussion

4.1 Experimental Results

Figure 3 shows the measured NO emissions for the premixed, non-premixed and dilution cases. Air dilution demonstrated favorable performance as NO emissions were substantially lower than that from the non-premixed combustion. For instance, at an equivalence ratio of 0.6, NO emission for the dilution case is about 37% that of the non-premixed case. This percentage changed between 33% and 60% depending on the equivalence ratio. Comparing these emissions

to the premixed case, NO emission were almost as low as the premixed case especially at lower equivalence ratio of 0.6, where NO emission were only 0.5 PPM higher than that of the premixed case. This air dilution case had demonstrated emissions equal to those of premixed case [20]. However, due to the difference in the flow field, the behavior is different. This difference in flowfield arises from the center exit tube which prevents the fuel (cross jet) from further penetration into the combustor.

The CO emissions were also recorded for these cases, and the results are shown in Figure 4. Air dilution experiments demonstrated CO emission that were only 40% that of the non-premixed case (varied between 35% and 58% depending on the equivalence ratio). On the other hand, air dilution case demonstrated slightly higher CO emissions compared to the premixed case. This is in contrast to the previously reported CO data on air dilution under similar condition, where similar cases produced much higher CO emissions compared to non-premixed and premixed combustion cases [20]. This can be related to the difference in the combustor geometry and heat release intensity where the combustor examined herein has lower heat release intensity and a longer residence time to result in much lower CO emissions as compared to the geometry reported [20]. Thus controlled distribution of reactants in the combustor provides a critical role in mitigating CO emission.

To further examine the effect of air dilution, OH* chemiluminescence was used to identify the reaction zone and outline any differences between traditional non-premixed combustion and the air dilution cases. The obtained OH* intensity is shown in Figure 5 for both the cases, where it can be seen that the air dilution changed the reaction zone location into a larger area compared to a crescent like zone near the wall of the combustor.

This difference in the reaction zone behavior is related to the difference in the injections jets momentum between the air dilution case and the non-premixed case. In the dilution case, the cross jet (fuel diluted with air) had a higher jet force compared to the non-premixed case (fuel jet). For non-premixed case, the top injection force was about 0.246 kg.m/s^2 while the cross injection had a flow force of 0.012 kg.m/s^2 to result in a jet force ratio of 20.4 (top to cross) at an equivalence ratio of 0.8. In the air dilution case, the top injection jet force is about 0.2 kg.m/s^2 while the cross injection jet force is significantly increased to 0.07 kg.m/s^2 at the same equivalence ratio. This change in jet forces and momentums result in a much lower jet force ratio (2.64 vs. 20.4 for the non-premixed case) which significantly impacts the jet in cross stream penetration to impact the mixing. The increased momentum of the fuel jet is expected to lead to better mixing and diverts the main air jet to change the reaction zone. Such better mixing leads to enlargement of the reaction zone which subsequently results in decreased NO emission. Note that the increased cross jet momentum affects the turbulence levels to further enhance mixing. Previous investigations showed that any increase in turbulence level enhances mixing [22] and reduce emissions [23].

4.2 Numerical Simulation

To further examine the mixing process, numerical simulations under non-reactive were performed. The first step was to validate the numerical simulation. The model was set to run with air injection only (no fuel) from the main jet, and the obtained velocity field was compared to experimentally measured velocity data [10]. Figure 6 shows the location and velocity profiles at three different locations within the combustor.

Figure 6 shows good agreement between experimentally measured and numerically simulated velocities. Error bars in velocity profiles (Fig. 6) represents 20% error, typical for the

experimental data from PIV experiments (from particle size, particle material, and velocity gradients) [24]. The deviation between numerical simulations and experimental measurements near the center of the combustor is attributed to the high velocity gradients near the combustor exit tube wall, resulting in significant error in PIV measurements as the particles may not follow the flow streamlines.

After the model was validated against available experimental data, the inlet conditions were modified to match the experiments reported here. For each stream (main and cross), the mixture fraction of methane, oxygen, and nitrogen were set to match a specific condition (equivalence ratio). Table 2 gives the mass flow rates and calculated jet forces for each case.

Figure 7 shows contours of the velocity magnitude for both the non-premixed and air dilution cases at different equivalence ratios. The velocity profiles for different cases shows the effect of increased cross jet force (momentum) with dilution on the flowfield. Adding air to the fuel jet resulted in a much higher velocity (more than two folds as in the case of $\phi(\text{Phi})=0.8$, 220m/s vs 97 m/s) and increased mass emanating from the jet, to result in increased jet force (momentum), that changed the flowfield in the combustor.

This increased momentum also affected mixing inside the combustor. Figure 8 shows contours of methane mixture fraction (by mass) for the same cases as that shown in Fig. 7. The increased velocity and mass of the cross jet significantly enhanced mixing inside the combustor to result in a more favorable methane mixture fraction distribution.

The mixture fraction prior to ignition is of importance as it dictates the combustion behavior and resulting emissions. One can infer the mixture fraction prior to ignition through comparing the methane mixture fraction (obtained numerically) for different cases at the reaction initiation point (inferred here from the OH* chemiluminescence). Note that the actual methane mixture

fraction will be different under reactive conditions (due to the recirculation of hot reactive species rather than non-reacting flow as shown in numerical simulations), however, the comparison holds between different cases (premixed, non-premixed, and dilution). Figure 9 shows the methane mixture fraction distribution and the OH* chemiluminescence for the selected cases.

A comparison of methane mixture fraction for different cases prior to reaction initiation provides the difference between different fuel introduction methods. In dilution cases, the reaction starts at a location with a methane mixture fraction of about 0.06. This is much lower than the methane mixture fraction for the non-premixed case (0.12) and closer to the premixed mixture fraction of 0.0392 for the same equivalence ratio (0.7). The same trend was also found at other equivalence ratios. Such lower mixture fraction at the reaction zone is attributed to better mixing between air and fuel for the dilution case. This enhanced mixing resulted in lower emissions for the dilution cases as seen in Figures 3 and 4. The closer this methane mixture fraction is to the premixed case, the lower the emissions.

5. Conclusions

A new air and fuel mixing approach, wherein the fuel is diluted with air while portion of the fuel is introduced in the main air jet, has demonstrated NO emission comparable to those encountered in premixed combustion mode. These emissions were about 40% that of NO emissions under non-premixed combustion mode. Fuel dilution, characterized by high cross jet momentum, not only demonstrated NO emission reduction compared to non-premixed mode with comparable emissions to the of premixed mode combustion, but also lower CO emissions compared to non-premixed combustion. Such new mixing technique eliminates the possibility of

flame flashback and combustion instability as the equivalence ratio in the jets is outside the methane air flammability limit. OH* intensity distribution showed that the combustion zone is affected by the cross jet momentum and equivalence ratio. This was further demonstrated through numerical simulations, where the results showed that the methane mixture fraction prior to ignition plays an important role on emissions. The emissions were improved with the mixture fraction closer to the premixed case (prior to ignition). This mixture fraction stems from the mixing strategy within the combustor. The ability of air dilution mixing technique to demonstrate near zero emission shows the importance of mixture preparation effects on achieving distributed combustion conditions for ultra-low NO emissions, akin to premixed combustion mode, but without any drawbacks of flame flashback and combustion instability that accompany premixed combustion.

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Figure Captions

Figure 1. Schematic diagram of the Combustor used

Figure 2. Schematic of air dilution in the experimental facility

Figure 3. NO emission for premixed, non-premixed, and air dilution case

Figure 4. CO emission for premixed, non-premixed, and air dilution case

Figure 5. OH* chemiluminescence intensity distribution for both non-premixed and dilution cases

Figure 6. A comparison of numerical and experimental results. Contours of velocity magnitude at mid plane of the combustor (bottom left). Experimentally measured and numerically simulated velocity profiles normalized by inlet velocity at locations L1 (right top), L2, (right middle), and L3 (right bottom)

Figure 7. Contours of velocity magnitude for non-premixed and dilution cases at different equivalence ratios [m/s]

Figure 8. Contours of methane mixture fraction (by mass) for non-premixed and dilution cases at different equivalence ratios

Figure 9. OH* Chemiluminescence (indicating reaction zone) and methane mixture fraction for non-premixed and dilution cases at $\phi=0.7$

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Case	Air (Top) Stream			Fuel (Cross) Stream			Overall Equivalence Ratio
	Air flow (SLPM)	CH ₄ flow (SLPM)	Φ	Air flow (SLPM)	CH ₄ flow (SLPM)	Φ	
Premixed	137	11.5-7.2	0.8-0.5	-	-	-	0.8-0.5
Non-Premixed	137	-	-	-	11.5-7.2	-	0.8-0.5
Dilution	119.9-124	2.5-1.9	0.2-0.14	17.1-12.8	9-6.7	5	0.8-0.6

Table 2. Mass flow rates and jet forces for different configurations

Case	Phi	Main Jet			Cross Jet			Force Ratio
		Air [Kg/s]	Fuel [Kg/s]	Jet Force [Kg.m/s ²]	Air [Kg/s]	Fuel [Kg/s]	Jet Force [Kg.m/s ²]	
Non-Premixed	0.8	0.002683	0	0.2473	0	12.5E-5	0.012	20.35
	0.7	0.002683	0	0.2473	0	10.9E-5	0.0093	26.58
	0.6	0.002683	0	0.2473	0	9.38E-5	0.0068	36.18
Air Dilution	0.8	0.002354	2.71E-5	0.1943	0.000335	9.75E-5	0.0737	2.63
	0.7	0.002393	2.36E-5	0.2	0.000293	8.51E-5	0.0564	3.55
	0.6	0.002436	2.03E-5	0.207	0.0000251	7.31E-5	0.04135	5