Fuel Injection Effects on Distribution Reaction in a High Intensity Combustor

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Abstract

The role of dual location fuel injection (versus single injection) is examined for improved mixture preparation with enhanced reaction distribution in the combustor that offers reduced emissions. A cylindrical combustor was used at a combustion intensity of 36MW/m³.atm and heat load of 6.25 kW. Three different configurations were examined for the effect of dual location fuel injection using methane as the fuel. NO reduction of 48% was achieved with fuel injected at two locations versus single location at an overall equivalence ratio of 0.7. The OH* Chemiluminescence intensity distribution with dual location fuel injection showed the reaction zone to shift further downstream that provided longer fuel mixture preparation time prior to ignition under favorable fuel distribution conditions. The longer mixing time helped to improve mixture preparation with lower emissions. The NO* chemiluminescence signatures supported the results obtained on reduced NO emission. Mean to maximum OH* signal showed that dual location fuel injection enhanced distributed reaction at certain fuel

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distributions between the two locations (fuel injection ratio lower than 70%) for all the three configurations examined. Increase in air injection velocity from 46m/s to 102m/s showed up to 85% NO reduction utilizing dual location fuel injection without any increase in CO emission. Increase in air preheats from 300K to 600K reduced the extent of NO reduction. Dual location fuel injection provided improved reaction zone distribution in the combustor for all the experimental conditions reported here. Correlations are provided that describes the NO_x emission as function of fuel injection ratio.

Nomenclature

CDC: colorless distributed combustion

Ø: total equivalence ratio

FIR: fuel injection ratio is defined as the amount of fuel to an injector to the total amount of fuel to the combustor was used $FIR = \frac{F_i}{\sum_i^N F_i}$ where F_i represents fuel flow rate through injector (i)

V_{air}: Air velocity

OR: operational range is defined as the range of minimum to maximum fuel injection ratio operating range of FIR in which the emissions concentrations are lower than (single injection) 100% FIR's emissions concentrations

Ø_s: secondary equivalence ratio

a.u: arbitrary unit

T_{air}: air injection temperature

NO_{min.}: it is minimum corrected NO (corrected at 15% O₂) achieved by dual location fuel injection configuration

45 **NO_{single}**: it is the corrected NO (corrected at 15% O₂) concentration achieved by single location
fuel injection configuration at FIR 100%

1. Introduction

Stationary gas turbines are used worldwide for electricity production due to their higher
performance and reduced emission. However, environmental regulations recommend achieving
50 ultra-low emission levels from all combustion process. Novel combustion strategies are required
to reduce emission of harmful pollutants. Natural gas operated power generation plants are more
favorable than other fossil fuel fired power generation. Nitrogen oxides (NO_x) are produced
from the burning of fuels in gas turbine power plants. NO_x affects human health, cause acid rain
and photochemical smog. It reacts with ammonia and moisture forming nitric acid and related
55 particles. Carbon monoxide is produced from incomplete combustion. Significant progress has
been made to mitigate nitric oxide (NO) and carbon monoxide (CO) formation and emission.
Efficient natural gas power plant results in almost 50-60% less CO₂ emission than a typical coal
burning power plants. CO₂ is a greenhouse gas that causes global warming with primary source
being the combustion of fossil fuel in power plants.

60 The thermal field uniformity throughout the combustion zone is a key factor to mitigate
NO_x and other pollutants emission. Avoidance of the local hot spots within the combustion
chamber help enhance the life time of the engine. To reduce the NO_x, CO, soot and unburned
hydrocarbons, mixing between air and fuel must be enhanced prior to the mixture ignition. A
promising strategy to provide ultra-low emissions and better combustion specification for gas
65 turbine combustors is colorless distributed combustion (CDC). The CDC has shown uniformly
distributed reaction region throughout the entire combustion zone with ultra-low emission (near
zero concentrations) [1-6]. The CDC shares similarities with high temperature air combustion

(HiTAC) [6] except for the very short residence time available in CDC and significantly higher thermal intensities. The reactive gases with high thermal energy are internally entrained and mixed with air inside the combustor. Entrainment of hot reactive gases from within the combustor dilutes the oxygen concentration and increases the oxidizer temperature that allows avoidance of peak flame temperatures (hot spots) in the combustion zone and reduces the thermal NO_x formation. The CDC incorporates unique mixing of hot reactive gases with the air and/or fuel to improve thermal field uniformity. Besides, CDC provides low noise stable combustion without any need for flame stabilizers. In distributed combustion, the flame signatures are invisible as compared to conventional flame (hence colorless distributed combustion, CDC) and the reaction zone is distributed.

Ignition, wide burning range, high combustion efficiency, and low pollutants emission are some of issues that must be considered with single fuel injection combustion in gas turbines. Therefore, multiple different strategies have been developed to achieve low NO_x emission. Hot gases entrainment has shown to introduce ultra-low emissions and high combustion efficiency. This entrainment and the subsequent mixing occurring just prior to ignition are substantial components to foster distributed reactions. Distributed reaction is characterized by a lower reaction rate throughout the entire combustion volume which reduces the temperature rise. In contrast normal flames have concentrated flame reaction front with higher reaction rate that result in local hot spots and high NO_x emission. Enhanced mixing between the hot gases and air dilutes the oxygen concentration to result in lower reaction rate. Distributed reaction avoids thin reaction zone and results in a uniform temperature distribution which mitigates thermal NO_x formed from the Zeldovich mechanism (thermal NO_x mechanism) [7-13].

90 Multipoint injection promotes uniform mixture distribution, and a small multi-zone
burning provides shorter burning time, both resulting in reduced NOX formation [14]. Moreover,
multi-location fuel injection is expected to provide far more uniform distributed reaction zone in
the combustors and reduced emissions. Multi-location fuel injection is expected to improve
mixing prior to ignition. The mixing strategy between air and fuel jets in cross flow has a
95 substantial role to reduce overall emissions in both can and annular type gas turbine combustors
[14, 15] wherein jet in cross flow mixing is used. Several investigations have been performed on
the cross flow mixing of air jets into hot jets that form the fuel-rich primary zone due to its wider
range of applicability, such as in gas turbine combustors [14, 15]. For confined combustion the
degree of mixing is affected by geometry of the jets and cross flow, jet-to-mainstream density
100 and momentum-flux ratios. In confined subsonic cross flow, the primary interest has been on the
jet-to-mainstream momentum flux ratio and this ratio must be determined [14]. The Rich-
burn/Quick-mix/Lean-burn (RQL) combustor has been proposed to minimize the formation of
oxides of nitrogen (NO_x) in gas turbine combustors. The goal here is to rapidly convert the
overall fuel-rich mixture to fuel-lean mixture, thus avoiding near stoichiometric combustion and
105 minimize NO_x formation that results from high temperatures [14-15].

Previous investigations have been performed on (RQL) combustor to study the effect of
mixing rich fuel main stream with penetrating jets. Authors [14] used a cylindrical reactors and
different numbers of penetrating jets (8, 12, 14 and 22). The experiments performed under non-
preheating and preheating conditions showed that the NO_x concentrations in the center region for
110 the 8-hole module were significantly lower than that from the other three modules due to good
penetrations achieved with the 8-hole module case. High concentrations of NO_x were observed
in the wake of the jets near to the cylinder walls. This was attributed to high temperatures and

longer residence times caused by jet induced recirculation [14]. Confined mixing between air and fuel (propane) [16] was studied in a cylindrical combustor, using 10 round orifices attached to the combustor side walls. The temperature and emissions were recorded upstream using different number of injectors [16]. A lean-direct injection (LDI) has been used wherein the fuel was injected directly into the combustion zone at overall lean equivalence ratio mixtures. A multi-injection mode was used that had 49 injectors (having the same equivalent size as that of a single injector [17]. The results from the multiplex NO_x emission was compared with the Multi Point Integrated Module (MPIM) [17, 18] obtained in a previous study. The results from the multiplex showed approximately the same NO_x levels as that obtained from the 25 point MPIM combustor. However, the NO_x levels were higher using the 36 point MPIM [17]. In staged combustion, methane /air flames were investigated for the effects of pressure, flame temperature, fuel-to-air ratio and fuel staging ratio since these parameters have a major effect on NO_x formation [19]. The effect of different pilot [20] injection configuration on flame stability, emission and flame location were inspected using swirl burner. Different secondary injection (pilot fuel, premixing of pilot fuel with air, air injection only) were examined for different locations of the pilot lance at different total equivalence ratios. The aim was to control the pressure pulsations and NO_x emission by changing the fuel and air repartition inside the burner at constant total equivalence ratio at a given heat load. A multi injection pilot of air located upstream of the recirculation zone showed good results in stabilizing the combustor [20]. For most of the aforementioned investigations, multi-location fuel injection was used under premixed condition. Non-premixed combustion provides stable combustion without the presence of any combustion instability in comparison to premixed combustion. Subsequently, considerable efforts are made to reduce pollutants emission produced from non-premixed burning of fuels.

This paper investigates the effect of using three different configurations of dual location fuel injection in a high intensity gas turbine combustor under non-premixed mode. The role of air injection velocity and air preheats are also examined. Emissions of NO and CO were measured at the combustor exit to determine the extent of overall emission reduction. Global NO* chemiluminescence intensity distribution was performed to visualize the regions of NO emission from within the combustor. Reaction zone distribution was examined using OH* chemiluminescence.

2. Experimental Setup

2.1 *The high intensity combustor*

A cylindrical stainless steel combustor was used as the test combustor and it was insulated internally with a cylindrical ceramic insert to reduce heat losses through the combustor walls, see fig. 1. The combustor ends were sealed on one side using stainless steel plate (having zirconia on the inside) while the other end had a quartz plate that provided optical access to the flame for chemiluminescence diagnostic. The zirconia coating reduced the thermal losses from the combustor. The exhaust tube was extended inside the combustor to enhance residence time of reactants for improved mixing and lower CO emission [3]. Air was injected tangentially at mid-point along longitudinal length of the combustor at temperature of 300K. The combustion behavior was examined under preheated air conditions (at $T_{\text{air}}=600\text{K}$) that simulated more realistic gas turbine combustion conditions which have high air temperatures due to polytropic compression of air in compressor prior to the combustion chamber. Fuel jets were located at the middle of the combustor height, see fig. 1. Methane (CH_4), which represents major component of natural gas (that contains 97% methane), was used as a fuel herein. Methane has high ignition temperature, and low flame speed [21].

2.2 Fuel injection system

The combustor had four fuel injection locations (0, 1, 2, and 3) as shown by red arrows in fig. 1. Fuel introduced from (0, 2, and 3) locations are called secondary while the fuel introduced from location number 1 is called primary (main) fuel. Figure 1 shows the four different locations for the fuel injectors examined here. For dual location fuel injection, three different combinations were utilized having a combination of main injector (F1) and a secondary injector located at F0, F2 or F3 to provide combinations of (F1F0, F1F2, and F1F3). In each combination, the amount of fuel from each injector was varied while keeping the total amount of fuel constant so as to maintain the same overall equivalence ratio (ϕ). The fuel injection ratio (FIR) is the amount of fuel supplied through the main injector (F1) to the total amount of fuel injected to the combustor.

2.3 Instrumentation

Air and fuel flow rates were controlled using choked flow orifice system. Precision stainless steel orifices were utilized to choke the flow of gas with upstream pressure controlled to supply the required mass flow rate of the gas. The upstream air pressure was controlled using pressure regulators to maintain a steady pressure and avoid any oscillations in pressure from the compressor turning on and off periodically. The upstream pressure for both air and fuel was maintained between (± 0.2 psi) of the desired pressure. A piezo resistive digital pressure gauge with an accuracy of (± 0.25 psi) was utilized to maintain the desired pressure.

Gas analyzers were used to determine the emissions. The NO concentration was measured using a NO–NO_x chemiluminescent gas analyzer; CO using the non-dispersive infrared method and O₂ (used to correct the NO and CO emissions to standard 15% oxygen concentration) using galvanic cell method. Before experiments the gas analyzers were calibrated

for NO, CO and O₂. The combustor was allowed to run for 20-30 minutes before recording any data for assuring operation under steady state conditions. The data were repeated at least three times for each configuration. The experiments were repeated at least twice to ensure consistency and repeatability of data.

The OH* chemiluminescence intensity distribution was captured using an ICCD camera fitted with an interference filter (centered at 307 nm) that provided visualization of the flame reaction zone under various conditions. The NO* chemiluminescence intensity distribution from within the flames was obtained using an ICCD camera fitted with a narrow band filter (centered at 248 nm $\pm$ 15nm). OH* signal can be detected at 284 which is outside the NO* signal bandwidth. The gain of the camera for these images was set to 150 with one accumulation per image and an exposure time of 100ms.

High intensity combustion is investigated here for the effect of dual location fuel injection using three different fuel injection combinations. The effect of single and dual location on NO and CO emissions fuel injection along with OH* and NO* chemiluminescence distribution were recorded and compared. Dual location injection was also examined for achieving distributed reaction zone as compared to single injection. The combustor performance was also investigated with configuration F1F2 and F1F3 using 600K air preheats. The F1F0 configuration mixed portion of the fuel with air to provide the mixture outside the flammability limits. Air preheats for F1F3 combination was not examined to avoid pre-ignition of the resulting mixture. The effect on NO, CO emission OH*-NO* chemiluminescence distribution and near distributed reaction conditions were examined.

3. Results and Discussion

Three different configurations of fuel injection (F1F2, F1F3 and F1F0) were investigated at two air injection velocities of 46 and 102m/s. Two air injection velocity used were 46m/s at normal 300K air temperature (or ~92m/s at 600K) and 102m/s at 300K (or 205 m/s at air preheated to 600K). These velocities were selected based on one of the author (AKG) prior experiences on high temperature air combustion technology development for low NO_x emission [6]. F1F2 configuration was examined with V_{air} of 46m/s in earlier publication [9]. However, it was added to current results to show comparison between the three fuel configurations together. In addition, F1F2 was examined at different velocity than previously published results [9]. Contemporary stationary gas turbine combustors operate at thermal intensities of 15 to 25 MW/m³.atm [25]. The cylindrical Combustor used here was at a combustion intensity of 36MW/m³.atm and heat load of 6.25 kW so that the results presented here are relevant to contemporary gas turbine combustion conditions. Higher thermal intensity will reduce the combustion volume. Our main focus was to seek lower emissions at higher thermal intensity.

3.1 Effect of air injection velocity on different dual location fuel injection configuration

Effect of increasing the air injection velocity from 46 to 102m/s on NO and CO emission, and OH*, NO* chemiluminescence signatures for the even reaction distribution zone inside a high intensity combustor was examined under dual location fuel injection condition for different fuel configurations.

3.1.1 F1F2 dual fuel location configuration

F1F2 scheme refers to the fuel distribution between inlet injection ports F1 and F2. The primary fuel injector position (F1) was kept fixed in all the experiments while the secondary injector position was changed between 2 or 3 (for non-premixed case) and location 0 (for premixed case). The fuel injection ratio (FIR) represents fraction of primary fuel flow rate in F1

injector to the total fuel flow rate injected to the combustion space, i.e., $FIR = F1/(F1+F_i)$, where $i=2, 3$ or 0 . Fuel injection ratio (FIR) varied from 100% to 50%. Results are presented at three equivalence ratios ($\phi = 0.6, 0.7$ and 0.8) for constant air flow rate and total amount of fuel
 230 changed to alternate global equivalence ratio with respect to different conditions.

The results showed lower emissions at lower equivalence ratios for both the single and dual location fuel injection using methane as the fuel, see Fig. 2. This was attributed to the direct effect of lower adiabatic flame temperature at lower equivalence ratios with less thermal NO formation according to Zeldovich or thermal mechanism. Reaction (1) needs high activation
 235 energy (E_a) to be initiated so lower adiabatic flame temperatures at leaner equivalence ratios would result in reduced NO emission [21]. NO emission formed in most of the methane flames results from atmospheric nitrogen fixation by extended Zeldovich mechanism (thermal mechanism).

The results depicted NO reduction with moderate decrease in FIR at injector 1, see fig. 2. Beyond this point, the NO emission started to increase, surpassing that of single injection (100% FIR) until FIR of 50%. This reveals that fuel injector location is important to control NO
 245 emission and reaction distribution. This observed behavior was consistent for the three equivalence ratios at two different air velocities investigated herein. For the same ϕ , increase in air injection velocity (V_{air}) from 46 to 102 m/s enhanced the entrainment rate which aided to reduce the NO emission level as shown in fig.2.

Air admitted to the combustor was at normal temperature of 300K for the two different velocities investigated here. At $V_{\text{air}}=46$ m/s, $\phi = 0.6$, 28% NO emission reduction was obtained with fuel injection ratio changed from single injection (100% in injector no.1) to FIR of 90%, see fig. 2. However, at $V_{\text{air}}=102$ m/s and $\phi=0.6$, 57.5 % NO reduction was obtained at FIR of 80%. At $\phi= 0.7$, NO reduction was approximately 24% and 46%, for air velocities of 46m/s and 102m/s, respectively. Further increase in ϕ resulted in even less NO emission reduction. At $\phi = 0.8$, the reduction was 13% and 17% at two air velocities of 46m/s and 102m/s, respectively. This is attributed to the direct role of higher adiabatic flame temperatures at higher equivalence ratio 0.8. This reveals that air injection velocity must be appropriately adjusted at higher equivalence ratios so as to achieve higher reactive gases entrainment. Ricou and Spalding [22] investigated the effect of diameter of free air jet flow under non-reacting condition on the entrainment ratio. They provided a correlation between jet diameter and the entrainment ratio achieved. Higher jet velocity with smaller diameter jet increased the entrained gas flow rates. Their obtained results support the gained NO reduction to provide the direct role of higher air injection velocity.

The results showed that the maximum reduction percentage in NO emission with dual location fuel injection approximately doubled with increase in V_{air} from 46 to 102 m/s at equivalence ratios (ϕ) of 0.6 and 0.7. However, at ϕ of 0.8 the change in NO reduction was only 4%. The maximum reduction was found to shift at FIR of 80% instead of 90% with change of velocity from 46 to 102m/s. The reasons for these observations are discussed in the results on OH* and NO* chemiluminescence section.

The F1F2 combustor configuration had an operating range (OR) of FIR with emissions lower than 100% FIR (single injection), see fig. 2. At ϕ of 0.6, the FIR operation ranges for air

injection velocity of 46 and 102 m/s were respectively ($78\% \leq \text{FIR} \leq 100\%$) and ($60\% \leq \text{FIR} \leq 100\%$), approximately. However, at ϕ of 0.7, the operation ranges (OR) were approximately ($73\% \leq \text{FIR} \leq 100\%$) and ($52\% \leq \text{FIR} \leq 100\%$) at the two air injection velocity of 46 and 102 m/s respectively. wider OR at ϕ of 0.6 and 0.7, was obtained when the velocity was increased which contributed to higher NO emission reduction percent. At ϕ of 0.8, the operation range was approximately ($50\% \leq \text{FIR} \leq 100\%$) and ($70\% \leq \text{FIR} \leq 100\%$) at air injection velocity of 46 and 102m/s, respectively. For OR at 46m/s it is preferred not to operate at FIR lower than 55% since the emissions became similar at all the equivalence ratio examined which indicated a diffusion flame behavior.

Effect of utilizing dual location fuel injection on the CO emission was depicted in figure 3. The CO emission decreased with decrease in equivalence ratio (0.8 to 0.7). However, decrease in equivalence ratio from 0.7 down to 0.6 caused lower flame temperature and consequently less CO to CO₂ conversion. Reducing FIR (having dual fuel location entry instead of single fuel entry) of 100% to 80% resulted in significant reduction in CO was for ϕ of 0.8 and 0.7. Further reduction in FIR increased the CO emission up to FIR of 50%. Increase in air velocity did not affect the CO emission trends significantly except for ϕ of 0.6 wherein the CO emission increased at higher velocity (102 m/s).

Previous publication [9] showed the effect of dual location fuel injection (F1F2) on the OH* chemiluminescence intensity distribution in the combustor at $V_{\text{air}}=46\text{m/s}$. That could also help to correlate the change in reaction zone intensity with change in FIR and that is compared with recently obtained results of higher air injection velocity in fig. 4. For F1F2 with V_{air} of 46m/s [9], at FIR of 100% the reaction zone was of crescent shape distribution located diametrically opposite to the fuel injection location. The reaction intensity and crescent thickness

decreased with decrease in equivalence ratio. The flame shifted downstream with decrease in equivalence ratio. This observed trend was consistent for the results obtained at FIR of 100, 90 and 80% at $V_{air}=46\text{m/s}$ (see different rows for the selected FIR in Fig. 4 existing in previously published data [9]).

The flame shifted downstream influenced the CO emission. Oxidation of CO can be described using the following reaction steps [21]:

The shift of flame to further downstream location increased the available mixing time to promote reaction (4). However, reaction (4) has a limited contribution in production of CO_2 due to slow occurrence. The more realistic CO oxidation is through reaction (6) producing H atoms that react with O_2 forming OH and O (reaction 7). These produced radicals feed back into oxidation reaction (5). These two reactions support the production of CO_2 and suppress CO [21]. The aforementioned mixing time promotion and reaction steps (4-7) interpret the CO reduction that occurred with change in FIR from 100% to 80% for $\phi=0.8$ and 0.7 at any examined velocity. Meanwhile low temperature at ϕ of 0.6 hindered CO to CO_2 formation which agreed well with the dramatically increased CO results as shown in fig. 3.

Figure 4 provides a comparison of OH^* chemiluminescence with air injection velocity increased from 46 to 102m/s at $\phi=0.6$. Increase in air injection velocity from 46 to 102m/s provided lower OH^* chemiluminescence concentration and stretched the flame in the combustor so that flame occupied over longer distances in combustor with improved uniformity of

chemiluminescence signal distribution. Different scales are used to illustrate variation in flame thickness with increase in air flow velocity. This was significant at FIR of 80% and can be related to the maximum NO reduction (57.5%) reported in fig. 2 using F1F2 configuration from the enhanced entrainment which is attributed to higher air injection speed. Figure 4 also revealed that increasing the FIR to 70% with $V_{air}=102\text{m/s}$ increased the flame thickness as compared to that obtained with lower velocity (46m/s). Thus higher injection velocity provided improved distribution flame in the combustor.

Figure 5 shows the ratio of mean to maximum OH^* intensity (a.u) distribution at different FIRs of 90%, 80% and 70% for two different air injection velocities of 46 and 102m/s. Figure 5 supports the obtained results in fig. 4 with respect to enhanced distributed combustion using dual location fuel injection. The higher the value of intensity ratio, the more was the colorless distributed combustion. The results show that the reaction is more distributed in the combustor at ϕ of 0.6; however at higher ϕ the flame is less distributed. The higher air injection velocity of 102 m/s showed improved distributed combustion in comparison to lower velocity of 46m/s.

NO^* chemiluminescence was also investigated to determine the behavior of NO with change in FIR at an equivalence ratio of 0.6. Figure 6 shows the recorded NO^* chemiluminescence signal at a wavelength of 248 nm. These results can be compared with the OH^* chemiluminescence data shown in fig.4 obtained at $V_{air}=46\text{ m/s}$, $\phi=0.6$ and $100\% \leq \text{FIR} \leq 70\%$. The OH^* signal was higher than NO^* signal by one order of magnitude for the same exposure time and camera gain. The NO^* and OH^* filters had the same efficiency so as to seek a direct comparison of signals together. The OH^* and NO^* scales are different to better detect the signal distribution. NO^* signal coincided with the maximum OH^* signal. Also the NO^* signal

decreases with reduction of FIR from 100% to 90% or 80% which coincided well with the drop obtained in measured NO data reported in fig. 2.

3.1.2 F1F3 fuel location configuration

This configuration used F1 as a primary fuel inlet and F3 as a secondary fuel inlet, the
345 inlets position is shown in fig1. Fuel injection ratio (FIR) was varied from 100% to 50%. Results were presented at three equivalence ratios ($\phi = 0.6, 0.7$ and 0.8) for constant air flow rate but with the total amount of fuel changed that provided different global equivalence ratios at for the examined conditions. Figure 7 showed measured NO emission (ppm at 15% O_2) with secondary fuel injection at F3 location. NO emission concentration against FIR showed similar behavior at
350 all the equivalence ratios (ϕ) examined using two different air injection velocities (V_{air}), see fig. 7. The results showed similar behavior of NO trends to that observed with the F1F2 arrangement and adiabatic flame temperature had identical influence on the NO trends for different equivalence ratios reported here. The results showed a decrease in NO with decrease in FIR from injector 1; the reduction was favorable with FIR between 90 to 56%. Thereafter, NO emission
355 increased exceeding that from single injection (FIR=100%). At $V_{air}=46\text{m/s}$, ϕ of 0.6, the results shown in figure 8 revealed a 32% reduction in NO emission with the injection changed from single injection (100% in injector no.1) to 90% in no. 1 and 10% in no. 2). For $V_{air}=46\text{m/s}$, at ϕ of 0.7, the results showed 30% NO reduction with FIR of 80%. The results showed less (19%) NO reduction at higher equivalence ratio ($\phi = 0.8$) with FIR of 90%. Increase in air injection
360 velocity (V_{air}) to 102m/s at ϕ of 0.6, 0.7 resulted in lower NO emission and NO reduction of some 50% and 56%, respectively. Increase in air injection velocity from 46m/s to 102m/s at ϕ of 0.8 reduced the emission level. The NO emission maximum reduction was 18% with dual location fuel injection. The F1F3 showed lower NO emission. This was attributed to closer

location of secondary injector to the air jet which fostered more entrainment and enhanced
365 mixing between air, secondary fuel, hot gases and primary fuel.

Operational range (OR) provides the preferable range of FIR providing lower NO
emission and better distributed combustion using duel location fuel injection ($FIR < 100$)
compared to single fuel location fuel injection (i.e., $FIR = 100\%$). The F1F3 scheme provided a
wider and more favorable operational range (OR) of FIR, at ($\emptyset$ of 0.6 and 0.7). At $V_{air} = 46\text{m/s}$,
370 OR of $70\% \leq FIR \leq 100\%$ and $67\% \leq FIR \leq 100\%$ were obtained at $\emptyset$ of 0.6 and 0.7, respectively,
shown in fig. 8. At $V_{air} = 46\text{m/s}$, $\emptyset$ of 0.8, the OR is $58\% \leq FIR \leq 100\%$, similar to that found for
F1F2 arrangement. At $V_{air} = 102\text{m/s}$, the OR is $50\% \leq FIR \leq 100\%$, $55\% \leq FIR \leq 100\%$ and
 $65\% \leq FIR \leq 100\%$ for $\emptyset$ of 0.6, 0.7 and 0.8, respectively. The F1F3 showed wider OR than F1F2
and this is attributed to better mixing in F1F3 case. The F1F3 arrangement promoted mixing
375 between the entrained reactive gases, secondary fuel from F3 and fresh air. Being closer to the
air injector, secondary fuel (injection at F3) mixes with entrained gases and then this mixture
entrained gases into the high momentum air jet. The weak momentum of the secondary jet
followed the hot reactive gases to penetrate the entrained reactive gas flow to result in higher NO
reduction.

380 The CO emission using F1F3 configuration were similar to that obtained using F1F2
configuration, see Fig 9. The CO emission concentration decreased with decrease in equivalence
ratio from 0.8 to 0.7 since this reduction in equivalence ratio is accompanied by increase in
available air, present in the air fuel mixture, to support complete combustion, see fig. 8. Further
reduction in equivalence ratio (0.7 to 0.6) led to increase in CO emission due to lower flame
385 temperatures and less energy available for complete CO to CO_2 conversion as the flame
approached the lean limits [21]. At $V_{air} = 46\text{m/s}$, $\emptyset$ of 0.6, CO emission decreased with change in

FIR from 90 to 50%. The trends on CO at equivalence ratios of 0.7 and 0.6 were similar. As the FIR was changed from 100% to 80%, significant reduction in CO was observed at 0.8 equivalence ratio; further reduction in FIR increased CO emission, even above the single injection configuration. At higher air injection velocity of 102m/s and $\phi=0.6$ trend showed similar behavior but the CO emissions were higher than at 46m/s. With high velocity (102m/s) and ϕ of 0.7 and 0.8 CO trends were similar to that at low velocity (46m/s) with the same CO concentration at all FIR.

OH* chemiluminescence intensity distribution in the combustor for F1F3 arrangement is described in fig. 9. A change in FIR distribution 100% to 90% shifted the reaction zone further downstream than that with 100%. The F1F3 arrangement provided lower chemiluminescence signal intensity than F1F2; however, flame distribution was similar to that of F1F2 arrangement. At FIR=80%, the reaction zone extended slightly upstream with no change in the position downstream (as compared to FIR of 90%). More stretching of the flame was observed with FIR reduced to 70% which indicated better flame distribution. Two flame zones were observed with FIR=70% that provided enhanced reaction distribution. The first started just after secondary fuel injector. The second flame zone was extended upstream of the primary fuel injector. This can be promoted by having different distributed flame zones through multiple fuel injection (non-premixed) locations in a combustor. Non-Premixed mode provides stable combustion. The goal was to promote distributed reaction inside the combustor and avoid the hot spots which lead to higher NO emission. FIR of 70% provided enhanced distributed combustion with low NO emission.

Figure 10 illustrated the effect of increasing the air injection velocity to 102m/s on the OH* chemiluminescence. At FIR of 90% and 80%, increase in air injection velocity from 46m/s

410 to 102m/s resulted in flame signatures at the same position for all the examined equivalence ratios with the crescent stretched further downstream with the increase in air injection velocity. At FIR of 70% and ϕ of 0.6, increase in velocity did not change the flame location but increased the OH* concentration. At FIR of 70% and ϕ of 0.8 and 0.7, increase in air injection velocity caused the flame to shift downstream, see figs. 9 and 10.

415 The mean OH* to maximum OH* intensity ratio was utilized to determine the extent of distributed reaction. Figure 11 shows the intensity ratio for different FIR at two different air injection velocities. For the same FIR, increase in air injection velocity from 46 to 102m/s maintained similar intensity ratio from the F1F3 configuration.

3.1.3 F1F0 fuel injection configuration

420 F1F0 arrangement was investigated at different equivalence ratios in which the secondary fuel (F0) was supplied through the air injector location with the secondary equivalence ratio (for air and F0 mixture) so as to reside outside the flammability limits of methane ($0.46(\text{lean}) < \phi < 1.65(\text{rich})$) [23] so that premixed mixture is not combustible.

Figure 12 shows the calculated values for the secondary equivalence ratios at different
425 total equivalence ratios ($\phi = 0.6, 0.7, 0.8$) for different FIR= (100%: 50%); maximum secondary $\phi = 0.4$ was achieved at $\phi = 0.8$ & FIR= 50% which is lower than lean flammability limits of methane at atmospheric pressure [23]. The purpose of this experiment is to mix secondary fuel with air and have out of flammability limits mixture. Further decrease in FIR from 50% would result in a combustible premixed mixture since the secondary equivalence ratio would be within
430 the flammability limits. F1F0 arrangement showed different behavior than F1F3 and F1F2 configurations. The NO emission decreased significantly when the FIR was varied from 100% to

50% at all the equivalence ratios, see fig. 12. NO reduction of 41.5, 48 and 35.5% was obtained with FIR reduced from 100% to 50% at $\phi=0.6$, 0.7 and 0.8, respectively with $V_{\text{air}}=46\text{m/s}$.

This reduction was attributed to better mixing between secondary fuel and main air jets as the secondary jet is essentially premixed with air. Reducing FIR at any equivalence ratio reduced the emissions since reduction in FIR increased secondary fuel flow rate with more fuel premixed with the air. Increase in air injection velocity fostered NO reduction. At FIR of 50% and ϕ of 0.6, 0.7 and 0.6, the NO reductions obtained were 85%, 70% and 55%, respectively.

The CO emission with FIR at different equivalence ratios (ϕ) is shown in figure 13. The data shows significant reduction in CO with FIR decreased from 100% to 70%. Thereafter the CO start to increase with higher values at FIR=50%. This increase is attributed to lower temperatures that resulted in lack of conversion from CO to CO₂. The observed CO trends are similar to that from F1F2 and F1F3 geometries.

OH* chemiluminescence with primary fuel supplied through F1 and secondary fuel and air mixture (that resided outside the flammability limits) supplied through F0 showed similar behavior to F1F2 arrangement. The signal intensity was similar to the F1F2 arrangement. Premixing between the secondary fuel and air before the combustor promoted flame distribution. In this arrangement the (secondary fuel+ air) charge supplied to the combustor mixed rapidly with the hot reactive combustion gases. The OH* images showed enhanced reaction zone that was more than that observed for the other two configurations (F1F2 and F1F3). A shift in flame further downstream led to more ignition delay and better mixing. The better mixing reflected higher NO reduction from this configuration, see fig. 14. At lower equivalence ratio the flame moved further downstream than higher equivalence ratios which indicate longer time for mixing prior to ignition and to support improved reaction zone distribution.

455 The results on the effect of increase in air injection velocity (V_{air}) from 46 to 102m/s on
OH* chemiluminescence using the F1F0 configuration is shown in fig. 15. For all examined FIR
and ϕ , increase in V_{air} increased the thickness of the flame crescent and also the crescent
extended (especially for ϕ of 0.6). The NO emission reduction is attributed to strong mixing
between secondary fuel and air in premixed inlet and increase in entrainment ratio from
460 increased air velocity. The OH* chemiluminescence concentration reduced with reduction in air
velocity. NO* chemiluminescence showed that reducing FIR reduced NO* signal to result in NO
emission reduction, see fig. 15.

3.2 Effect of air preheating on dual location fuel injection configuration

3.2.1 Effect of Change in air injection temperature on dual location fuel injection

465 Effect of increase in air injection temperature on the NO and CO emission was examined
on the OH* chemiluminescence and the reaction zone distribution inside the combustor under the
dual location fuel injection. Figure 16 shows the maximum NO reduction. The results on NO
emission depicted the behavior under different FIR conditions at equivalence ratios of 0.8, 0.7
and 0.6 using air preheats of 600K. The maximum NO reduction (percentage) was reduced
470 compared to reduction achieved with air at 300K. This is attributed to high adiabatic flame
temperature which means high available energy required to initiate reaction (1), in Zeldovich
mechanism, leading to higher NO emission. The OH* chemiluminescence showed similar
behavior on flame distribution, with the expected early onset of flame with air preheats. The
flame distribution improved with FIR reduced from 100% to 70%. Reduced NO emission and
475 flame distribution reveals that dual fuel location injection is also favorable with air preheats.

3.2.2 Effect of Change in air injection temperature on dual location fuel injection

Figure 17 shows the effect of increasing the air injection temperature on NO reduction and reaction distribution for F1F3 configuration. The results show the NO emission behavior under different FIR at equivalence ratio of 0.8, 0.7 and 0.6 using 600K air preheats. Maximum
 480 NO reduction achieved was 14%, 7% and 17% at equivalence ratios of 0.8, 0.7 and 0.6, respectively. This arrangement provided higher reduction than the F1F2 arrangement due to close position of secondary injector F3 to the air jet which promotes reactive gases entrainment.

3.3 Correlations describing dual location fuel injection

3.3.1 Non-premixed dual location fuel injection correlations

485 Correlations to describe NO emission as a function of fuel injection ratio (FIR) can be viewed for F1F2 and F1F3 arrangements. Equations (8 and 9) below were used to correlate the ratio of NO to minimum NO concentration with FIR at equivalence ratio of $\phi=0.6$ & $T_{air}=300K$, see figure 18:

$$\left. \frac{NO}{NO_{min.}} \right|_{F1F2} = 14.774[FIR]^2 - 27.415[FIR] + 13.897 \quad (8)$$

$$\left. \frac{NO}{NO_{min.}} \right|_{F1F3} = 20.868[FIR]^2 - 35.547[FIR] + 16.122 \quad (9)$$

The above relations have a regression co-efficient of 0.9706 and 0.9995 which indicates a very good fit. Equations (8 & 9) are valid for the F1F2 and F1F3 arrangements at equivalence ratio of 0.6 for FIR ranging from 100% to 50%. The aforementioned correlations 8 and 9 can be merged in one correlation that can describe NO/ ($NO_{min.}$) as function of FIR for both F1F2 and
 495 F1F3 configurations see fig. 19 and eqn. 10.

$$\left. \frac{NO}{NO_{min.}} \right|_{F1F2 \& F1F3} = 17.821[FIR]^2 - 31.481[FIR] + 15.01 \quad (10)$$

Knowing the NO concentration of single non-premixed combustion can provide the concentration of NO at different FIRs. The combustor operation conditions can then be

determined based on the given correlation for non-premixed dual location fuel injection, see fig.

500 20. Two correlations are given below in eqn. 11 and 12:

$$\left. \frac{NO}{NO_{single}} \right|_{F1F2} = 10.651[FIR]^2 - 19.76[FIR] + 10.019 \quad (11)$$

$$\left. \frac{NO}{NO_{single}} \right|_{F1F3} = 14.333[FIR]^2 - 24.415[FIR] + 11.073 \quad (12)$$

Furthermore, if the data for the two fuel configurations is combined, a single relation can be obtained to predict the corrected ratio of NO/ (NO_{single}) emission for different fuel injection ratios

505 (FIR) as shown in fig. 21 and eq. 13.

$$\left. \frac{NO}{NO_{single}} \right|_{F1F2 \text{ \& } F1F3} = 12.492[FIR]^2 - 22.09[FIR] + 10.546 \quad (13)$$

3.3.2 Premixed dual location fuel injection correlation

To verify that the newly defined fuel injection ratio (FIR) parameter is also suitable for predicting the NO, a NO correlation that included the effect of fuel staging ratio was defined,

510 see Eq. (14) [19][24]:

$$\left. \frac{NO}{NO_{ref.}} \right|_{F1F0} = A \times \exp (B \times \alpha) \quad (14)$$

Where, A and B are constants which were determined from experiments, α is the fuel staging ratio and has a similar definition to the fuel injection ratio (FIR). Therefore, the general eqn. 14 can be used in current experiments to obtain equation that predicts NO_x with respect to a reference NO_x concentration (NO_{ref.}). Two reference concentrations were used NO_{min.} and NO_{single}, as illustrated in fig. 22.

$$\left. \frac{NO}{NO_{min.}} \right|_{F1F0} = \exp(0.986FIR - 0.5379) = 0.584 \exp(0.986FIR) \quad (15)$$

$$\left. \frac{NO}{NO_{single}} \right|_{F1F0} = \exp(0.986FIR - 1.0733) = 0.342 \exp (0.986FIR) \quad (16)$$

4. Conclusions

The dual location fuel injection has shown to provide improved distributed reaction combustion in a cylindrical high intensity combustor with ultra-low emission. Three different fuel injection schemes are reported here (F1F2, F1F3, and F1F0) at two different air injection velocity of 46 and 102 m/s and temperature of 300K and 600K. Fuel injection ratio ($FIR1 = F1/F1 + Fi$, where $i=0, 2, 3$) varied from single fuel injection at 100% to dual location fuel injection (at 50%) at 3 different equivalence ratio of $\phi=0.8, 0.7, 0.6$ have been reported. The results showed 48% NO reduction using F1F0 configuration at fuel injection ratio of 50%. However, maximum NO reduction for F1F2 and F1F3 cases was 28% and 32%, respectively. Reduced emission is from improved mixing with fuel injection from two injection ports. Change in air injection velocity from 46m/s to 102m/s enhanced NO reduction (57.5%, 56% and 85%) using F1F2, F1F3 and F1F0 configurations, respectively. Larger entrainment achieved at higher velocity reduced the NO emission significantly. NO* chemiluminescence distribution showed the same trend of measured NO emission for all the configurations examined here showed decreased NO emission until reaching certain FIR, depending on the configuration used; thereafter the NO emission increased.

OH* chemiluminescence distribution showed improved distributed combustion with dual location fuel injection than single injection location. The OH* chemiluminescence provided a significant reduction in signal intensity for F1F3 configuration. The mean OH* to maximum OH* signal provided an indication on the extent of reaction zone distribution inside the combustor. The results showed that F1F2 arrangement provided improved distributed reaction in the combustor. Dual location fuel injection provided improved distribution at higher equivalence ratio of 0.8. Air preheats from 300K to 600K, showed similar behavior on emission and OH* chemiluminescence distribution. However, the reduction obtained was less because of increased

adiabatic flame temperature. Multi-location fuel injection with proper design can provide enhanced NO reduction to foster distributed reaction in high intensity combustors. The experimental results and preliminary correlations provided here assist in determining NO_x emission as a function of fuel injection ratio (FIR) in combustors, albeit they require validation over a large range of conditions for their wider use.

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Revised Figure 18

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Fig. 20

Fig. 21.

Fig. 22

Figure Caption:

Fig. 1. Cross-section of the cylindrical combustor (non- premixed with 3 fuel injection locations) and (premixed with 1 fuel injection location).

Fig. 2. NO emission at different fractions of fuel injected from injector one for scheme F1F2 at two different air injection velocities.

Fig. 3. CO emission at different fuel injection ratios (FIR) for scheme F1F2 for two different air velocities.

Fig. 4. OH* Chemiluminescence intensity distribution at different FIR (90%-70%) at equivalence ratio (ϕ) of 0.6 for F1F2 arrangement.

Fig. 5. Comparison of mean to maximum intensity ratio of OH* chemiluminescence signal (distribution index) at different dual fuel schemes for V_{air} of 46 and 102m/s.

Fig. 6. NO*Chemiluminescence intensity distribution for F1F2 arrangement at different FIR (100%-70%), (ϕ) of 0.6 and V_{air} of 46 and 102m/s.

Fig. 7. NO concentration at different fractions of fuel injected to one injector for scheme F1F3.

Fig. 8. CO emission at different fuel injection ratios (FIR) for F1F3 configuration.

Fig. 9. OH* Chemiluminescence intensity distribution at different FIR (90%-70%) at equivalence ratio (ϕ) of 0.6, 0.7 and 0.8 for F1F3 arrangement and $V_{air}=46\text{m/s}$.

Fig. 10. OH* Chemiluminescence intensity distribution at different FIR (90%-70%) at equivalence ratio (ϕ) of 0.6, 0.7 and 0.8 for F1F3 arrangement and $V_{air}=102\text{m/s}$.

Fig. 11. Mean to maximum intensity ratio of OH* chemiluminescence signal (distribution index) for different dual fuel injection schemes at V_{air} of 46m/s and 102m/s.

Fig. 12. NO concentration at different fractions of fuel injected to one injector for F1F0 scheme two different air injection velocities.

Fig. 13. CO emission at different fuel injection ratios (FIR) for scheme F1F0 two different air injection velocities.

Fig. 14. OH* Chemiluminescence intensity distribution at different FIR at (ϕ) of 0.6, 0.7 and 0.8 for F1F0 arrangement and $V_{air}=46\text{m/s}$.

Fig. 15. NO* and OH* Chemiluminescence intensity distribution at different FIR at (ϕ) of 0.6, 0.7 and 0.8 for F1F0 arrangement and $V_{air}=102\text{m/s}$.

Fig. 16. Maximum NO emission reduction (percentage) using dual location fuel injection at different ϕ and $T_{air}=600\text{K}$ for F1F2 configuration.

Fig. 17. Maximum reduction percentage in NO emission using dual fuel location injection at different ϕ and $T_{air}=600K$ for F1F3 configuration.

Fig. 18. Variation of emission ratio $NO/ (NO_{min.})$ with change of FIR at $\phi=0.6$ for F1F2 &F1F3.

Fig. 19. Variation of $NO/ (NO_{min.})$ with change of FIR at $\phi=0.6$ for non-premixed dual location fuel injection.

Fig. 20. Variation of emission ratio $NO/ (NO_{single})$ with change of FIR at $\phi=0.6$ for F1F2 &F1F3.

Fig. 21. Variation of $NO/ (NO_{single})$ with change of FIR at $\phi=0.6$ for non-premixed dual location injection.

Fig. 22. $NO/ (NO_{single})$ & $NO/ (NO_{min.})$ with change of FIR at $\phi=0.6$ for premixed dual location injection F1F0.