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Injection of Lean Mixtures into Hot Burned Gas for Maintaining Low-NOx Emissions over an Extended Range of Fuel-Air Ratios in Prevaporized Combustion

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ABSTRACT

“Flameless combustion” of lean to ultra lean mixtures, supported by high-temperature burned gas, can resolve the dilemma between complete combustion versus ultra-low NOx emissions in gas turbine combustors.

The characteristic of NOx emissions and combustion in “lean-lean” two-stage combustion was investigated for premixed-prevaporized kerosene-air mixtures using a coaxial flow configuration. A circular jet of secondary kerosene-air mixtures of lean to ultra-lean compositions were injected into the primary hot burned gas prepared by combustion of lean mixtures on an annular perforated flame holder

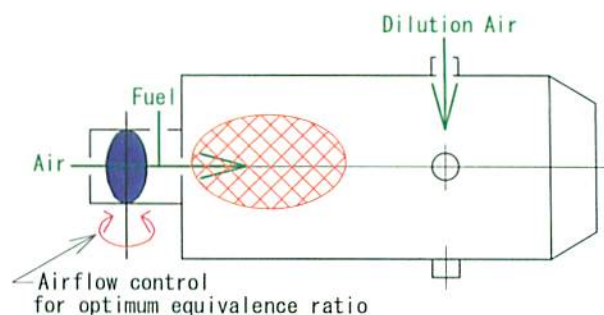
The progress of reaction and NOx formation in the combustion tube were measured for non-swirled and swirled secondary jets. The conditions required for complete combustion of the secondary mixtures were defined for various ratios of secondary to primary air flow rates.

INTRODUCTION

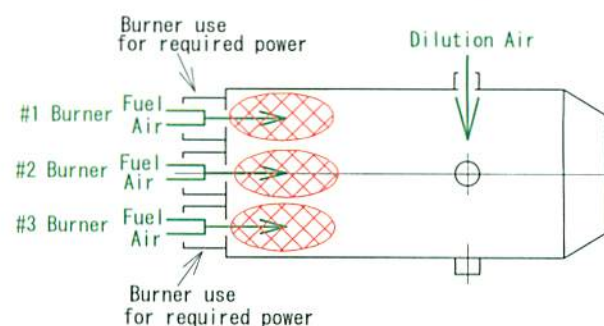
It is well recognized that lean premixed combustion is the most promising approach to ultra-low NOx emissions from gas turbines.^[1,2] In the conventional flame combustion, however, simultaneous achievement of ultra-low NOx emissions and complete combustion (low emissions of CO and unburned hydrocarbons) is only possible in a very narrowly limited range of fuel-air ratios. Thus, control of combustion zone equivalence ratio is required for achieving ultra-low NOx emissions in lean premixed flame combustion.

One approach to the equivalence ratio control is to modulate the air flow in proportion with fuel flow by means of a device such as valves. Accordingly, the mixture velocity varies, which is a big problem from the viewpoint of flame stability. A flame can be stabilized at mixture velocities in a range bounded by blow-off and flash-back. At lean conditions, the range is too narrow to cover most part of the operating range of gas turbines.

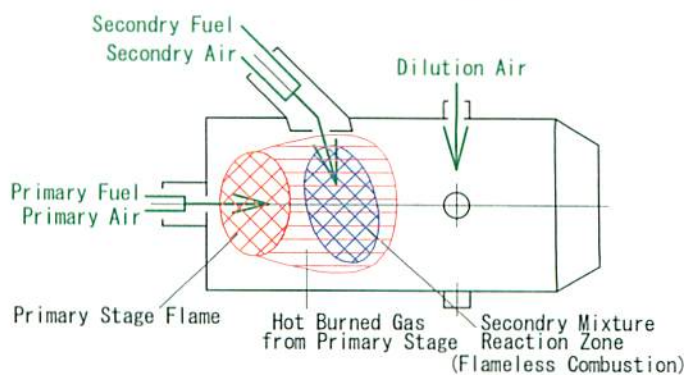
An alternative approach is fuel staging among multiple burners.^[3] The number of the burners in operation is increased or decreased in accordance with power. The equivalence ratio of the mixture to each burner can be maintained in a narrow range, being independent of the variation of overall equivalence ratio of the combustor. The control of fuel flow is easier and more reliable than that of air. Emissions of CO and hydrocarbons, however, increase stepwise when fuel injection into the next burner begins to increase



(a) Variable geometry (air staging)



(b) Multiple burners (fuel staging)



(c) “Lean-lean” two-stage combustion

FIG.1. Schematics of combustion concept.

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power.

Therefore, new innovative methods are required to achieve complete combustion while maintaining ultra-low NOx emissions over a wide range of engine operations. To meet this requirement, the present authors have proposed the use of "flameless thermal reactions" of very lean to ultra-lean mixtures supported by hot burned gas in "lean-lean" two-stage combustion.^[4] This concept is schematically shown in Fig. 1 (c), as compared with those of air staging by variable geometry Fig.1 (a) and fuel staging among multiple burners Fig.1 (b).

When the temperature of the mixture after mixing with the burned gas is higher than a threshold and the residence time in the reaction zone is longer enough, secondary lean mixtures can be completely reacted, even if they are leaner than the lower limit of inflammability at the combustor inlet temperature and pressure conditions. Mixing of the secondary mixtures with the primary burned gas is the critical for successful application of this concept, as previous researches shows.^[4]

In the present experimental study, the effects of swirl of the secondary mixture jets on the progress of reactions and NOx formation were investigated using kerosene in a model tubular gas turbine combustor. The conditions required for complete combustion of the secondary mixtures were investigated for different ratios of secondary to primary air flow rates. The present results clearly shows that "flameless combustion" of very lean to ultra-lean mixtures supported by hot burned gas is very advantageous in extending the limit of operating range to leaner side where ultra-low NOx emissions are achieved while maintaining complete combustion.

NOMENCLATURE

Ta_1 : temperature of primary mixture at entrance of combustion chamber, K.

Ta_2 : temperature of secondary air at entrance of combustion chamber, K.

Wa_1 : primary airflow rate, g/s.

Wa_2 : secondary airflow rate, g/s.

Wf_1 : primary fuel flow rate, g/s.

Wf_2 : secondary fuel flow rate, g/s.

ϕ_1 : equivalence ratio of primary mixture, -.

ϕ_2 : equivalence ratio of secondary mixture, -.

ϕ_t : overall equivalence ratio, -.

L : distance between probe hole and flame holder surface, mm.

U_1 : average velocity of primary air, m/s.

U_2 : average velocity of secondary air, m/s.

EXPERIMENTAL APPARATUS AND PROCEDURES

The major parts of the combustion system used in the present study are a combustor, a vaporizer/premixer unit for primary mixture preparation and air heaters. A detailed drawing of the combustor is shown in Fig. 2. A transparent quartz cylinder, 80 and 83 mm in inner and outer diameters, respectively, and 200 mm in length, is used for visualization of the phenomena in the flame tube. To prepare hot burned gas the primary stage, flames of prevaporized fuel-air mixtures are stabilized on a 13.5-mm thick annular flame holder, 80 and 27 mm in outer and inner diameters, made of a ceramic honeycomb plate whose blockage is about 71%. Thirty 5-mm diameter drilled holes are evenly distributed over its surface. Secondary mixture is injected from a 25-mm inner diameter quartz tube, 1.0 mm thick and 200 mm long, coaxially placed in the flame tube and penetrating into the primary hot burned gas through the center hole of the flame holder. The exit of the mixture injection tube is positioned 27 mm above the flame holder surface. Secondary fuel is atomized in the mixture injection tube by means of a twin-fluid atomizer, 12 mm in outer diameter, positioned coaxially in the secondary mixture injection tube. The atomizer has a fuel nozzle of 0.4 mm in diameter, connected to a straight tube of 2 mm in inner diameter and 0.5 mm in thickness, is

concentrically positioned in the convergent air nozzle at the end of the atomizer. The distance from the exit of the injection tube to the tip of the atomizer is adjustable to vary the degree of prevaporization of the secondary fuel. The axial swirler on the twin-fluid atomizer, shown in the photo on the left bottom of Fig. 2, is removable.

Figure 3 shows a schematic illustration of the air and fuel flows and a photo of the combustion system of the experimental apparatus. The primary air from a blower is heated and then supplied to the vaporizer/premixer unit for primary mixture preparation. Its flow rate is measured by an orifice flow meter upstream of the heater. The primary fuel is injected into the preheated air stream at the inlet of the vaporizer/premixer unit by using a pressure swirl atomizer of a nominal flow rate of 0.5 gallons/h at an injection pressure of 100 psi. A cone made of perforated metal sheet, a full angle of 15 degrees, is coaxially positioned in the cylinder of the vaporizer/premixer unit. The air jets from the holes on the cone are effective for preventing atomized fuel from adhering to the cylinder wall. A pair of glass windows are fitted to the wall of the cylinder in an opposing position. A laser beam is passed across the cylinder through the glass windows to check whether the fuel droplets are completely vaporized. If vaporization remains incomplete, scattering of the laser beam by the fuel droplets is visible. A static mixer is placed between the exit of the vaporizer/premixer unit and the inlet of the combustor so that homogeneous mixtures can be supplied to the combustor.

The secondary air is supplied from a rotary compressor via a settling chamber. A very small portion of the secondary air, about 0.2g/s, enters the twin fluid atomizer, bypassing the heater, for atomizing the secondary fuel while the rest enters the secondary mixture injection tube of the combustor to mix with the atomized fuel after preheated by the electric heater. A mass-flow meter measures each airflow rate.

In the present experiment, kerosene was used as a fuel for primary and secondary mixtures. The twin-fluid atomizer used for the secondary fuel atomization was axially positioned so that the tip of the atomizer was 155 mm upstream of the exit of the secondary injection tube. The temperatures of primary mixture and secondary air, Ta_1 , Ta_2 were fixed at 500 K and the primary airflow rate, Wa_1 , was also fixed at 4.0 g/s through the present experiment. The secondary air flow rates, Wa_2 , were 4.0, 6.0 and 8.0 g/s.

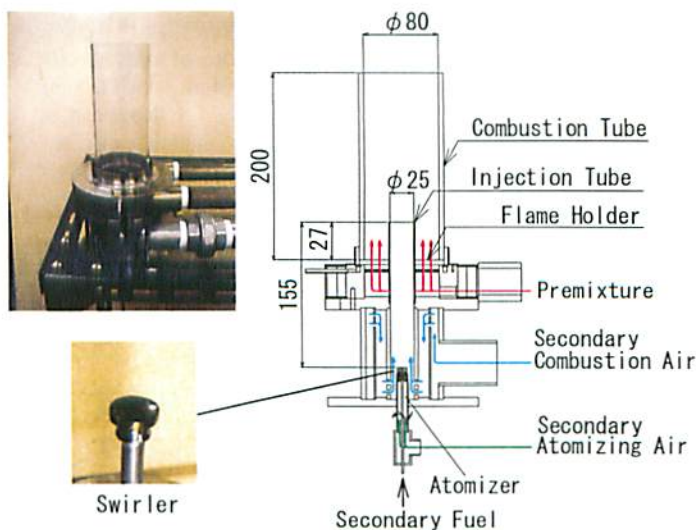


FIG.2. Schematic and photograph of experimental combustors.

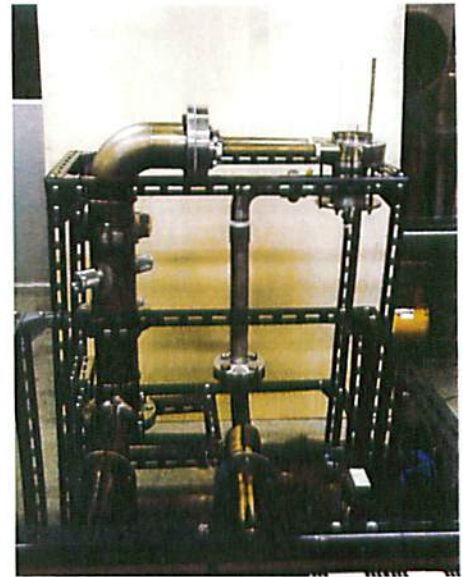
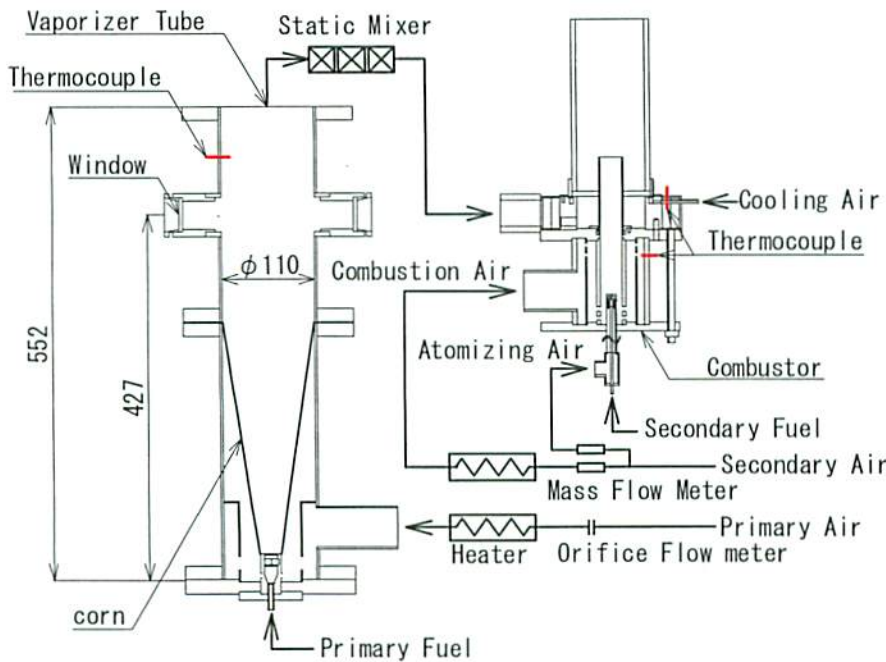


FIG.3. Schematic and photograph of experimental apparatus.

It was confirmed that in the vaporizer/mixer unit and in the secondary mixture injection tube, the atomized fuel were completely vaporized. It was found that swirling of the secondary mixture jet produced by the swirl vanes surrounding the atomizer significantly enhanced the mixing of the primary burned gas and the secondary mixture. Therefore, most runs were conducted for swirled secondary mixture jets.

A single-hole gas-sampling probe attached on a two-dimensional traversing rail was used to enable spatially resolved burned gas sampling in the combustion tube while an X-shaped gas-sampling rake with 36 holes in total, placed at a distance of 190 mm from the surface of the flame holder, was used to evaluate the total emissions from the combustor. The compositions of gaseous species in the sampled gas were determined by the standard gas analysis procedures: NO was measured by chemiluminescence, O₂ was measured by paramagnetic pressure difference, CO and CO₂ were measured by non-dispersed infrared absorption and HC (as CH₂) was measured by flame ionization detection.

EXPERIMENTAL RESULTS

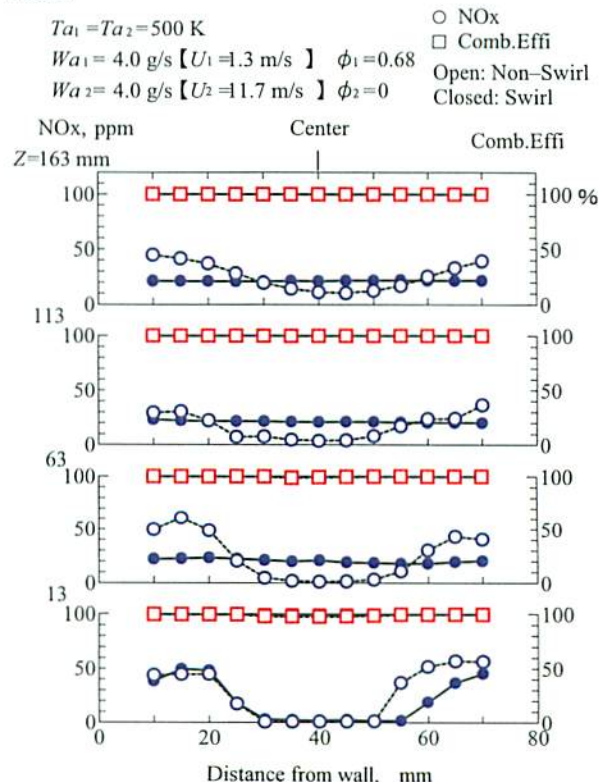
Progress of mixing and reaction of secondary mixture jets

In order to investigate the effects of swirl imparted to the secondary mixture jets, the diametral profiles of combustion efficiency and NO_x concentration for non-swirling and swirling jets of air and mixtures were measured at the fixed primary mixture equivalence ratio $\phi_1 = 0.68$ by using the single-hole gas-sampling probe at axial distances of 13 to 113 mm in 50 mm steps from the injection tube exit.

Figure 4 shows diametral profiles of combustion efficiency and NO_x concentration for simple and swirling air jets. The NO_x formed in the primary stage are transported from the periphery toward the center, with the profiles becoming uniform with increasing axial position. A comparison of the NO_x concentration profiles at different axial positions for non-swirling and swirling jet reveals that swirl to the jet drastically enhance the mixing with the burned gas. The combustion efficiency is 100% for the simple jet at any axial positions while a very small inefficiency is seen in the central region at 13 and 63 mm axial positions for the swirling jets, representing quenching of the combustion reactions of the primary mixture by the low-temperature swirling jet.

The secondary mixture jet, injected into and mixed with the burned gas from the primary stage, initiates reactions (combustion)

in the secondary stage. In addition to the measurements of the profiles of concentration and combustion efficiency along a flame tube diameter, luminous reaction zones in the secondary stage were photographically recorded for various secondary equivalence ratios in the range of 0.6 to 0.1 for the primary equivalence ratio of 0.68 as above. A comparison of the photographed luminous zones for non-swirling and swirling secondary mixture jets, shown in Fig. 5, shows that the swirl imparted to the secondary mixture jet has a significant influence on the developments of the secondary reaction zone. In the case of no swirl (on the left of Fig. 5), slender conical flames starting from the exit of the mixture injection tube are visible.


 FIG.4. Local NO_x concentrate and combustion efficiency in combustion chamber vs. distance from wall.

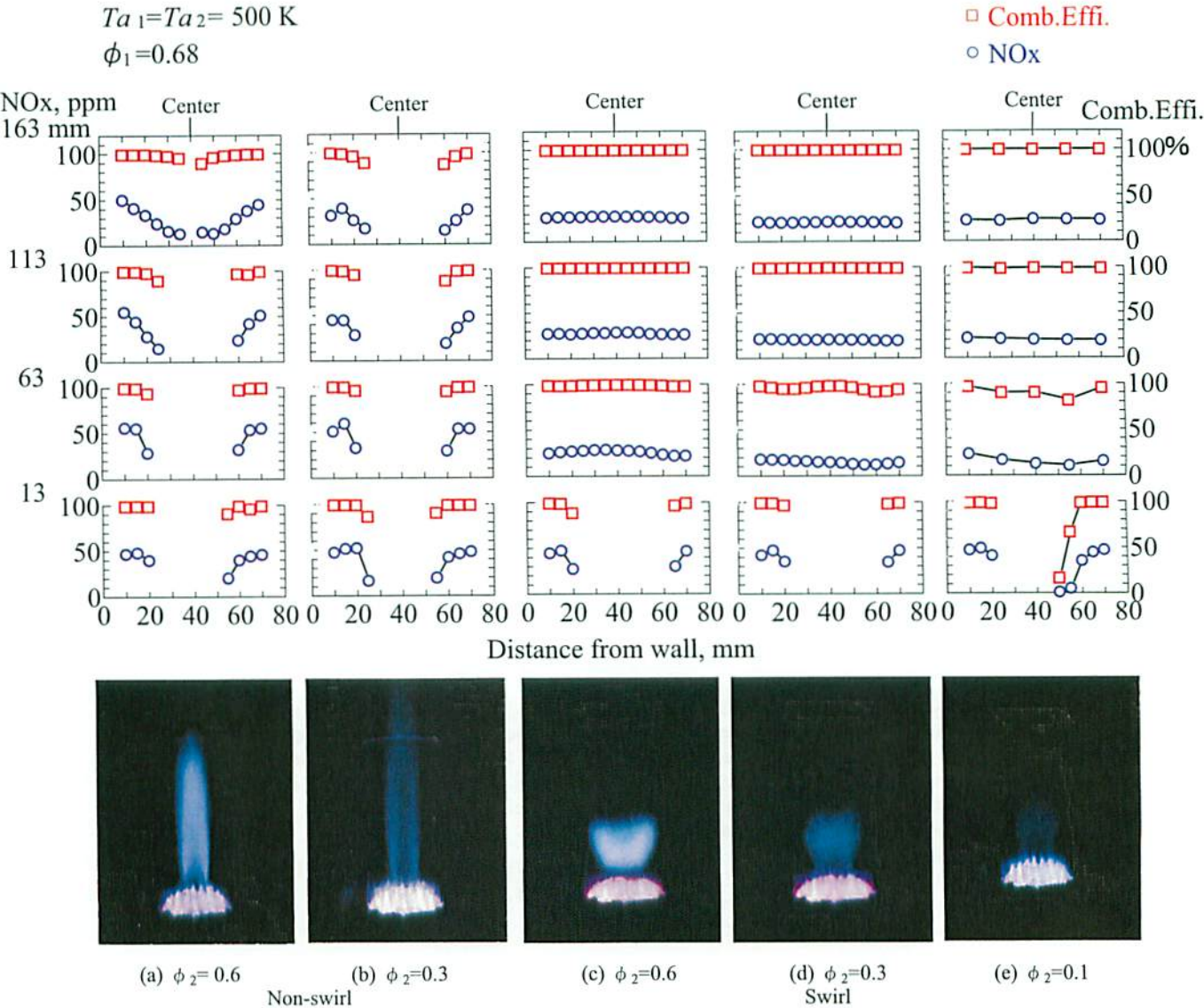


FIG.5. Effects of swirl and secondary mixture equivalence ratio, ϕ_2 , on flame shape and diametral NOx concentration and combustion efficiency profiles.

Imparting swirl to the secondary jets make the luminous zone much wider but shorter, as shown on the right in Fig 5. Swirl-enhanced mixing of the secondary jet with burned gas promotes the reactions of the secondary mixture. The luminescence of the reaction zone is decreasing in both cases as the secondary mixture becomes leaner.

The graphs on the upper part of Fig.5 show the NOx concentration and combustion efficiency profiles measured at four axial distances from the injection tube exit. Gas sampling was made in the regions where THC concentration was no more than 10000 ppm.

The plots of combustion efficiency data shows that the reaction of a non-swirling secondary mixture jet is limited to the periphery of the jet where the mixture and the primary burned gas mix. The tip of the unburned mixture reaches the exit of the combustion tube even when the mixture is combustible ($\phi_2=0.6$)

When swirl is imparted to the jets, mixtures of equivalence ratios as low as 0.1 can be completely reacted well before leaving the flame tube. "Flameless combustion" supported by hot gas has proceeded at axial positions promoted. A comparison of the NOx concentration profiles for the swirled and non-swirled secondary mixture jets shows that swirl was very effective in enhancing mixing. A comparison of the combustion efficiency profiles for swirled and no-swirled secondary mixture jets shows that swirl-enhanced mixing of the secondary mixture with burned gas resulted in faster reactions of the secondary mixture.

Emissions and combustion efficiency

Two-stage combustion vs. single-stage combustion

Typical NOx emissions and combustion efficiency for two-stage combustion are shown in Fig 6 as functions of overall equivalence ratio, together with those for single-stage combustion. For convenience, subscales for secondary mixture equivalence and calculated combustor exit gas temperature are drawn on the bottom and top in the figure, respectively.

In this and following figures, NOx emissions levels are expressed in parts per million at the 15% residual O₂ concentration. The correction for different residual oxygen concentrations enables us a direct comparison of the amounts of NOx formed in the primary and secondary stages since the magnitudes remain unchanged before and after dilution.

In the single stage combustion, just air was injected into the primary stage burned gas from the secondary injection tube and, hence, an increase in overall equivalence ratio was only brought by an increase in the fuel flow into the primary air. On the other hand, in the two-stage combustion, the fuel flow to the secondary air was increased while maintaining the primary mixture equivalence ratio at 0.68.

In the single-stage combustion, NOx emissions increase very steeply with increasing fuel flow rate or equivalence ratio. It is estimated that the gas temperatures in the primary combustion zone were varied in the 1650-1980 K range in the runs. The well-known thermal mechanism for NOx formation explains this steep increase in NOx emissions with increasing fuel flow rate in the single-stage combustion. With decreasing fuel flow rate, the combustion efficiency finally began to drop at $\phi_1=0.29$ ($\phi_1=0.58$) since the mixture equivalence ratio for the primary mixture, ϕ_1 , was approaching the lower limit of inflammability for the inlet mixture temperature. This is a characteristic inherent in lean premixed combustion.

In the two-stage combustion, as the secondary fuel increases, the NOx emissions level decreases from the value for non secondary fuel injection ($\phi_1=0.68$; $\phi_2=0$) to a minimum at around $\phi_2=0.4$ and then increases. When the secondary mixture equivalence ratio reaches the fixed primary mixture equivalence ratio of 0.68, the level of NOx emissions is almost equal to the level for air injection from the secondary injection tube, i.e. $\phi_1=0.68$ ($\phi_2=0$). A further increase in the secondary fuel flow rate makes

the gas temperatures in the secondary stage higher than 1830 K at $\phi_1=0.68$ and results in a steep increase in NOx emissions level. The combustion efficiency is more than 99.8% in the whole range of equivalence ratios tested. The "lean-lean" two-stage combustion, thus, has the potential to realized complete combustion over a wide range of equivalence ratios while maintaining very low NOx emissions levels. In the single-stage combustion, an increase in the combustor exit gas temperature by 190 K (from 1140 K to 1330 K) resulted in a five times increase in NOx emissions. In the two-stage combustion, however, very low NOx emissions were maintained together with complete combustion even when the combustor exit temperature increased by as much as 600 K (from 1230 K to 1830 K). This feature of the "lean-lean" two-stage combustion is very attractive since enlarging the low NOx operating range of gas turbines is very much requested.

Emissions at increased secondary air flow rates

The effects of secondary airflow rate, W_{a2} , on the NOx emissions and combustion efficiency were shown in Fig. 7. W_{a2} was increased from 4.0 g/s to 6.0 and 8.0 g/s whereas the primary equivalence ratio was fixed at $\phi_1 = 0.68$ as before. A comparison of the NOx data point for no secondary fuel at $W_{a2}=4.0$ g/s in Fig. 7 and the equivalent one in Fig. 6 suggests that the value of ϕ_1 for the runs whose data are shown in Fig. 7 was by a bit larger than that for the runs whose data are shown in Fig. 6. This is the major reason for explaining the noticeable difference (several ppm) in the baseline NOx levels shown in Figs. 6 and 7.

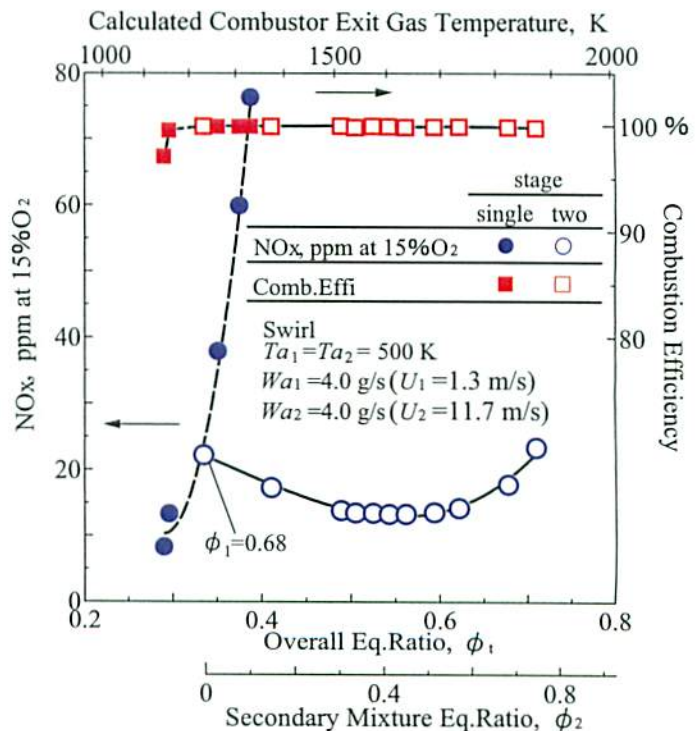


FIG.6. NOx emissions and combustion efficiency vs. overall and secondary mixture equivalence ratios for single-stage and two-stage combustion.

As for combustion efficiency, drops in combustion efficiency were seen for very lean secondary mixtures when the secondary air flow rate is doubled. Since the combustion of the primary mixture was complete, the dent on the combustion efficiency-overall equivalence curve was due to incomplete reaction of the secondary mixtures. As equivalence ratio of the secondary mixture increases at a given secondary air flow rate, the reaction in the secondary zone is enhanced and becomes completion before reaching the combustor exit. The initial temperature of the secondary mixture after diluted with hot burned gas is a critical factor in initiating reactions of the secondary fuel, especially for ultra-lean mixture compositions.

The initial gas temperatures for $\phi_1=0.68$ in the second stage were approximately estimated, by assuming complete mixing of the hot burned gas and the secondary mixture but neglecting the contribution of the secondary fuel, as 1260, 1130 and 1040 K for $W_{a2}=4.0$, 6.0 and 8.0 g/s, respectively. It may be said that the minimum temperature for achieving complete combustion of the secondary mixture is around 1250 K.

The NOx levels for a given secondary air flow rate is once decreasing from the level achieved in the single-stage combustion (with air dilution) and then increasing with increasing secondary fuel flow rate. The minimum NOx level is lower at larger secondary air flow rate. It is partly explained by a shorter residence time in the combustor on NOx formation and partly by early quenching of the NOx formation in the primary burned gas due to enhanced mixing at a larger secondary air flow rate.

The dotted lines in the figure represent combustion efficiency and NOx concentrations (at 15% O₂) calculated from the values obtained for no secondary fuel conditions assuming that the primary stage burned gas is just diluted by the secondary air and that the secondary fuel remains unburned. It is natural that because of reaction of the secondary fuel, the measured combustion efficiency data points are above the dotted line for each secondary air flow rate. As for NOx emissions, the measured data points for two-stage combustion are below the dotted line except for those at greater overall equivalence ratios. This supports the idea that the secondary fuel reduces some NOx produced in the primary stage into species other than NO or NO₂. A similar reduction of NOx by hydrocarbon fuels in combustion environments was previously reported.

Figure 8 shows the concentrations of monoxide, CO, and hydrocarbons, HC (as CH₂), emissions in the exhaust gas at the combustor exit as functions of overall equivalence ratio. The conditions are the same as those where the data shown in Fig. 5 were obtained. At overall equivalence ratios greater than about 0.7, the CO concentrations are increasing with overall equivalence ratio, in contrast to very low concentrations of hydrocarbons. The level of CO is less dependent on the secondary air flow rate. These trends are consistent with the predictions based on the equilibrium consideration. An adequate introduction of cooler dilution air into the burned gas or expansion of the gas in the turbine passage of a gas turbine may sustain the recombination of the CO so that high combustion efficiency results

The CO concentrations peak at overall equivalence ratios where combustion of the secondary mixtures is incomplete due to lower temperatures of the secondary mixtures after mixed with the primary burned gas. The CO concentration is much higher than the hydrocarbons concentration at the maximum secondary air flow rate of 8 g/s. A longer residence time in the combustor could reduce their levels but the effective residence times in the turbine passage are too short for most CO to be oxide before reaching the turbine exit.

Emissions for leaner primary mixtures

Figure 9 shows the NOx emissions and combustion efficiency at leaner primary mixture equivalence ratios: $\phi_1=0.59$ was selected for $W_{a2}=4.0$ and 6.0 g/s while $\phi_1=0.62$ for $W_{a2}=8.0$ g/s to secure

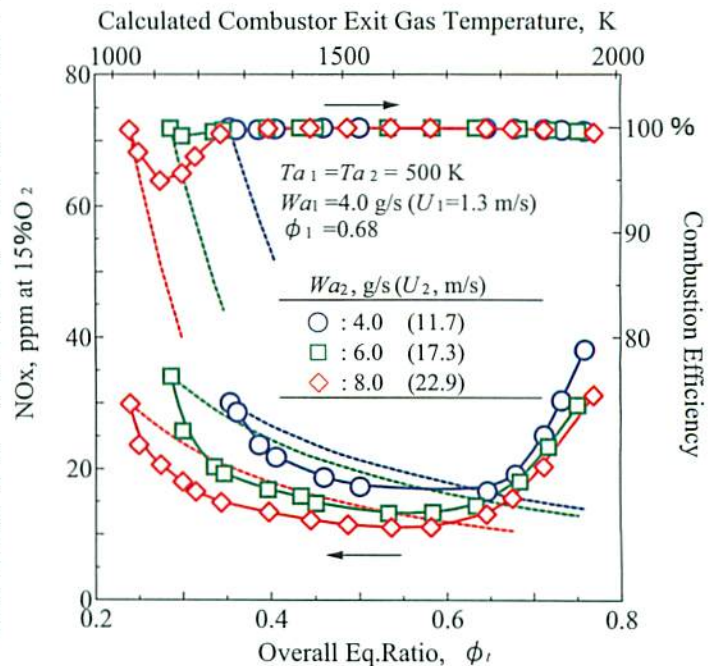


FIG.7. NOx emissions and combustion efficiency vs. overall equivalence ratio, ϕ_1 for different secondary airflow rates, W_{a2} , in two-stage combustion.

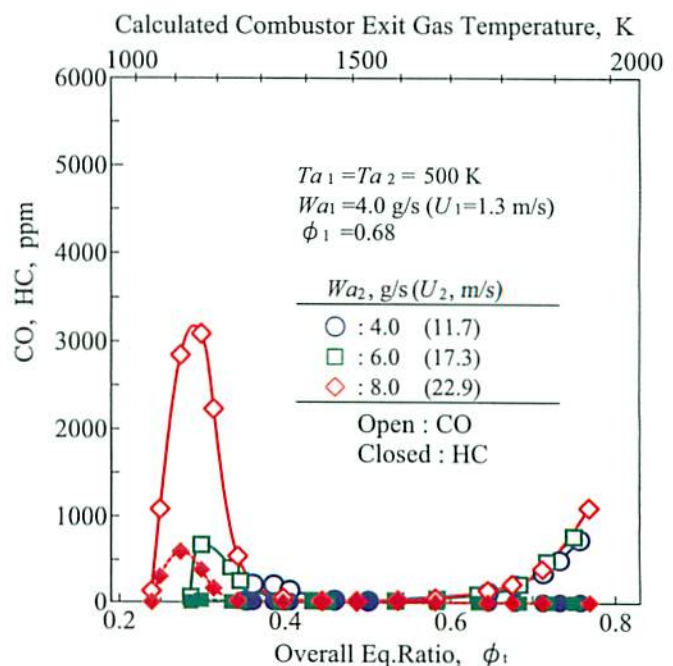


FIG.8. CO and HC emissions and combustion efficiency vs. overall equivalence ratio, ϕ_1 for different secondary airflow rates, W_{a2} , in two-stage combustion.

flame stability at higher mixture velocity. The combustion in the primary zone was incomplete, with overall combustion efficiencies being 97-98% and the NO concentrations were of single digit. The temperatures of the burned gas from the primary stage after diluted with the secondary air ($\phi_2=0$) flow rates of 4.0, 6.0 and 8.0 g/s were calculated: 1140 and 1040 K at $W_{a2}=4.0$ and 6.0 g/s, respectively, for the $\phi_1=0.59$ primary mixture and 980 K at 8.0 g/s for the $\phi_1=0.62$ primary mixture. These temperatures for $\phi_1=0.59$ are by 100 degrees lower and that for $\phi_1=0.62$ by 60 degrees lower than the temperatures for $\phi_1=0.68$.

It is clearly seen that reactions of very lean secondary mixtures are initiated by mixing with the hot burned gas from the primary stage, as before in Fig. 7. The dent on the combustion efficiency curve for each secondary air flow rate is, however, bigger than that in Fig. 7, where the data for $\phi_1=0.68$ are plotted. This is because of the lower temperatures of the primary burned gas. The effects of the secondary air flow rate on the drop in combustion efficiency at very lean secondary mixtures are consistent with those in Fig. 7. It is seen that for all secondary air flow rates, combustion becomes complete when the equivalence ratio of the secondary mixture is increased so that the final gas temperature can be as high as 1350K. The threshold gas temperature for complete combustion is 1250 K for $\phi_1=0.68$ as previously mentioned, suggesting the temperature of the secondary mixture after mixed with the primary hot burned gas is a critical parameter in determining the optimum air split between the primary and secondary stages for enlarging the operating range of complete combustion.

The NOx emissions are less than 10 ppm as long as the secondary mixture is leaner than the primary mixture and less than 5 ppm for the range $0.3 < \phi_2 < 0.6$. The behavior of NOx emissions level with overall equivalence ratio in the two-stage combustion is similar to that in Fig. 7: the NOx level is decreasing from the value for the single-stage combustion to a minimum and finally steeply increasing with increasing secondary fuel. It is also seen as before that the NOx emissions level approaches to the level achieved by the single-stage combustion when the overall equivalence ratio equals to that of the primary mixture or when the secondary mixture is identical to the primary mixture in fuel-air ratio. Dilution of secondary combustible mixture with burned gas produced by combustion of another mixture of the same fuel-air ratio as the secondary mixture is neutral in NOx formation.

The NOx levels increase steeply with an increase in overall equivalence ratio in the range $\phi_1 > 0.65$ where gas temperatures are higher than 1800 K, with the order of NOx levels for the three secondary flow rates being in the inverse order of secondary flow rates. This trend is also observed for $\phi_1 < 0.65$, though the NOx levels for the three secondary flow rates are distributed in a very narrow band.

Figure 10 shows the concentrations of monoxide and hydrocarbons emissions in the exhaust gas at the combustor exit as functions of overall equivalence ratio. Except for the lean side, typically $\phi_1 < 0.4$, the dependences of the concentrations of these species on overall equivalence ratio are very similar to those in Fig. 8. This also support the explanation based on the equilibrium of the burned gas.

The concentrations of HC at $0.2 < \phi_1 < 0.4$ are, in contrast to the data shown previously in Fig 8, as high as CO concentration for each secondary air flow rate. Their levels are much higher than those in Fig. 8. A lower temperature of the burned gas resulting from a reduced primary equivalence ratio leads to a lower initial temperature of the secondary mixture just mixed with the primary burned gas. Initiating reactions at lower mixture temperatures is difficult even if the reduction in the initial temperature due to the reduced primary mixture equivalence ratio is compensated for by increasing the equivalence ratio of the secondary mixture.

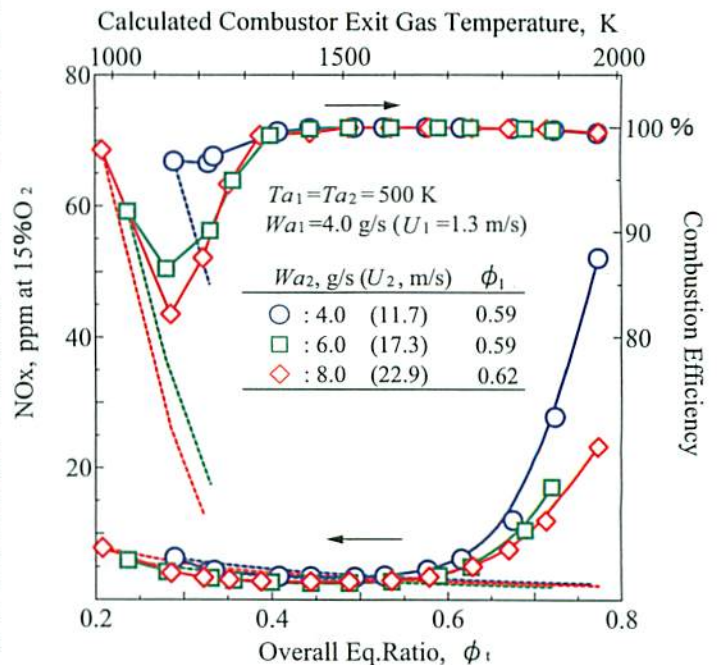


FIG.9. Ultra-low NOx emissions achieved by operation at lower primary mixture equivalence ratio, ϕ_1 .

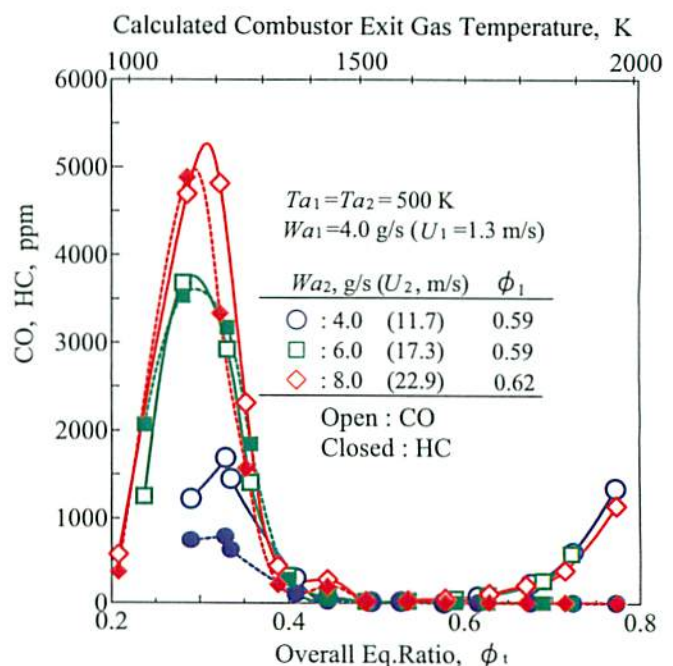


FIG.10. CO and HC emissions vs. overall equivalence ratio at lower primary mixture equivalence ratio, ϕ_1 .

Combustor design consideration

One of the most appropriate applications of the present two-stage combustion concept is for low-NO_x combustors of constant speed gas turbines. Since the total air flow rate is almost constant over the whole range of power in this application, the fuel flow to the primary stage can be kept constant at the value at idle operation over the range of loads and just the secondary fuel flow is increased or decreased to modulate power. The total NO_x emissions remain very low as long as the secondary mixture is leaner than the primary mixture. The lean limit for operation in term of overall equivalence ratio moves to the leaner side as the secondary air flow rate relative to the primary air flow rate increases. Both complete combustion and very low NO_x emissions can be achieved over the whole range of operation for moderate secondary air flow rates. For larger secondary air flow rate combustion efficiency actually may drop at overall equivalence ratios close to idling conditions where no secondary fuel is injected. It wouldn't be a big problem since most gas turbines are operated not at loads close to idle but at loads ranging from half to full.

CONCLUSIONS

The following are the major conclusions of the present study on the NO_x emissions in the reaction of lean secondary mixtures injected into and mixed with hot burned gas from the primary stage.

1. The use of thermal reaction of ultra-lean to moderately lean secondary mixtures supported by its mixing with hot burned gas from the primary stage is very advantageous in extending the range of ultra-low NO_x emissions and complete combustion.
2. Imparted swirl to the secondary mixture jet significantly enhances its mixing with the burned gas from the primary stage and hence its reaction.
3. The NO_x concentration in the exhaust remains in the single-digit ppm range, corrected at 15% O₂ when the equivalence ratio of the primary mixtures is around 0.6.
4. The reaction of the secondary mixture doesn't increase total NO_x emissions level as long as it is leaner than the primary mixture.
5. The reaction of secondary fuel becomes completion when the temperature of the secondary mixture after mixing with the primary burned gas is higher than about 1300 K partly depending on the primary equivalence ratio.
6. Some NO_x produced in the primary stage is reburned faster than the production of NO_x by the reaction of the secondary mixtures at exit gas temperatures lower than about 1800 K.

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