COMPATIBILITY BETWEEN LOW-NOx EMISSIONS AND HIGH-COMBUSTION EFFICIENCY BY LEAN DIRECT INJECTION COMBUSTION

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ABSTRACT

This paper describes the low-emissions capability of lean direct injection, liquid-fueled burners for gas turbine combustors. The emissions characteristics of direct injection and premixed-prevaporized combustion systems were evaluated at inlet air temperatures of 450, 550 and 650 K and atmospheric pressure. The equivalence ratio was varied in a range from about 1.2 to values close to flammability limits. The effects of fuel nozzle capacity and mixing tube length were also investigated. In the direct injection system, a double swirler/fuel nozzle module was used to inject fuel spray and air into combustion chamber. Fuel was atomized into the circular swirling jet using a pressure fuel nozzle placed 15 mm upstream of the inner swirler channel exit. In the lean premixed-prevaporized combustion system, the swirler/fuel nozzle module was inserted between the double swirler/fuel nozzle module and the end wall of the combustor chamber for fuel prevaporization. At lean conditions, combustion was more complete for the direct injection combustion system than premixed-prevaporized combustion system, though both were capable of very low NOx emissions.

INTRODUCTION

Regulation standards for pollutants emissions from stationary gas turbines are increasingly stringent in industrial areas. The emissions from aircraft during take-off and landing are also regulated by ICAO and other regulatory agencies. In the near term, emissions during cruise will also be regulated for the protection of the global environments (Wesocky and Prather, 1991; Bahr, 1992). It is anticipated that the NOx emissions from high-speed aircraft should depress the ozone layer in the stratosphere (Ko et al., 1991). Thus, the developments of low-NOx combustion technologies have been a major concern of gas turbine and aero-engine combustor researchers.

Gas turbines are operated at excessive air-fuel ratios because of the thermal limitations of turbine blades. In swirl stabilized diffusion flame burners used for most gas turbine combustors, fuel rich regions are inevitably formed in the combustion zone even at fairly lean operating conditions. Excessively lean operation actually results in an appreciable decrease in NOx emissions, but at the same time, leads to unacceptably high levels of carbon monoxide (CO) and unburned hydrocarbons (HC) emissions. Thus, the NOX emissions index, being defined as grams of NOx as NO2 per kilogram of fuel, generally depends moderately on equivalence ratio provided that combustion is almost complete.

Flame tube evaluation tests of the performance of various low-NOx combustion concepts have shown that lean premixed-prevaporized (pre-vaporized) combustion has the potential of attaining very low NOx emissions. For natural gas-fueled stationary gas turbines of medium to large outputs, successful on-engine and field testsings of lean-premixed combustors have recently been reported (Aigner and Muller, 1992, Smith 1992; Etheridge, 1994). The success of lean-premixed combustion with natural gas is attributed to the fact that gaseous fuel mixes easily with combustion air. With liquid fuel, on the other hand, prevaporization process is required for preparing highly homogeneous mixtures. Under the combustor inlet conditions typical of high-pressure ratio gas turbines and high-speed aero propulsion engines, the time available for premixing and prevaporization without autoignition before entering the combustion chamber seems very short. Another limitation to the mixture preparation at high temperature and pressure environments is flash back into the fuel-air mixing zone.

In our earlier study on the low-NOx capability of premixed-
prevaporized combustion, it was found that lean combustion of fuel sprays using a double swirler/fuel nozzle module was, contrary to expectation, capable of attaining comparably low NOx emissions levels. The emissions of HC and CO were also lower than those with the premixed–prevaporized system. After Tacina (1992) the method of fuel preparation of the burner module is classified into the direct injection concept. The combustion and emissions characteristics of the burner are described in this paper, being compared with those measured for premixed–prevaporized combustion.

**BURNER CONFIGURATIONS**

In the present study, emissions and combustion characteristics were investigated mainly for two burner systems: direct injection combustion and premixed/prevaporized combustion. These burner systems are illustrated in Figure 1. The identical
premixing tube lengths (12 and 18 cm) were tested to vary the residence times of the atomized fuel in the inner channel are too short for prevaporization to be appreciable for the degree of fuel prevaporization.

In the premixed–prevaporized system, a mixture of fuel droplets and air, prepared by the double swirler/fuel nozzle module, enters a premixing tube, 30 mm in diameter, before entering the combustor chamber 80 mm in diameter. A conical flame holder of a full angle of 30 degrees is supported by struts with its base at the exit of the premixing tube. Two premixing tube lengths (12 and 18 cm) were tested to vary the degree of fuel prevaporization.

In the direct injection system, fuel droplets are dispersed not only by the inner jet but also by the outer swirling air flow over the cross sectional area of the combustion chamber. The residence times of the atomized fuel in the inner channel are too short for prevaporization to be appreciable for the combustor reference velocity tested (20 m/s). Flames are stabilized by a swirling flow as is the case with the conventional swirl stabilized spray burners. The most important feature of this burner configuration, compared with the conventional swirl stabilized burners is the high-velocity center jet. Formation of fuel–rich regions is suppressed in the combustion zone and mixing of fuel and air is promoted with this burner design.

The direct injection burner used in the present study was designed to promote mixing of fuel and air in the combustion zone by eliminating low-velocity regions, which are purposely prepared for broadening blow-off limits in the conventional swirl–stabilized burners. The high velocity of inner jet as well as insignificant fuel vaporization in the jet should be effective in preventing flash back. Additionally, the residence time in the inner channel is too short for autoignition to occur.

**TABLE 1. BURNER CONFIGURATIONS TESTED**

<table>
<thead>
<tr>
<th></th>
<th>Direct Injection</th>
<th>Premixed–Prevaporized</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Fuel Nozzle</strong></td>
<td>4.0</td>
<td>4.0</td>
</tr>
<tr>
<td><strong>Capacity, GPH</strong></td>
<td>2.5</td>
<td>2.5</td>
</tr>
<tr>
<td><strong>Mixing Tube Length, cm</strong></td>
<td></td>
<td>18 (1.3)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>12 (0.85)</td>
</tr>
</tbody>
</table>

( ) Residence Time in Mixing Tube, ms

double swirler/fuel nozzle modules are used for both systems. The double swirler is co-rotational and the opening areas of the inner and outer channels are the same. Liquid fuel is atomized into the swirling air flow using a solid-cone fuel nozzle (Delavan Inc., Type-B) placed in the inner channel. The nominal spray angle is 60 degrees so that the spray may not impinge on the wall of the inner channel. For both burner systems, pressure fuel nozzles of capacities of 2.5 and 4 gallons per hour were used to vary mean droplet size, on which the degree of prevaporization of the fuel should depend appreciably.

The burner configurations including fuel nozzle capacities and premixing tube lengths tested in the present study are summarized in Table 1.

In the premixed–prevaporized system, a mixture of fuel droplets and air, prepared by the double swirler/fuel nozzle module, enters a premixing tube, 30 mm in diameter, before entering the combustor chamber 80 mm in diameter. A conical flame holder of a full angle of 30 degrees is supported by struts with its base at the exit of the premixing tube. Two premixing tube lengths (12 and 18 cm) were tested to vary the degree of fuel prevaporization.

**EXPERIMENTAL PROCEDURE**

Emissions measurements were made at inlet air temperatures of 450, 550, and 650 K and at atmospheric pressure. The equivalence ratio was varied over a range from about 1.2 to values close to flame stability limits on the lean side. The combustor reference velocity, the velocity of air averaged over the cross sectional area of the combustion chamber, was 20 m/s. Thus, the mixture velocity in the premixing tube was about 140 m/s, and the residence times in the shorter and the longer mixing tubes were 1.3 ms and 0.85 ms, respectively. Autoignition of premixed fuel spray–air systems have been investigated at elevated pressures and temperatures using flowing autoignition test rigs (Marck et al, 1977; Spadaccini and Te Velde, 1982; Tacina, 1980; Hayashi, 1988). Though measured data are considerably scattered, ignition lags of about 1 msec were predicted for the typical combustor inlet conditions of an aero-engine of a pressure ratio of 30.

Gas sampling was made using a water-cooled, equal-area gas sampling probe positioned at 500 mm downstream of the combustor end wall. The probe had an additional single sampling hole, which was used for measuring gas compositions at the center of the combustor cross-section. To prevent condensation of water vapor in the sampling probe and the sampling line, the cooling water feed rate to the probe was controlled and the sampling line was electrically heated to a temperature of around 430 K. Gas samples were diluted with pure nitrogen using a capillary flow mixer before entering the gas analyzer. The mixing ratio of nitrogen to the sampled gas was influenced only slightly by the variation in the viscosity of sampled gas. Concentrations of O2, CO, CO2, HC, NO, and NOx were measured by the standard gas analysis procedures: chemiluminescence for NO, non-dispersive infrared absorption for CO and CO2, flame ionization for HC, and paramagnetic analysis for O2. The NO2–NO converter with stainless steel catalyst was calibrated using standard NO2 gas to confirm conversion efficiencies greater than 96%.

Combustion efficiencies were calculated from the measured gas compositions, assuming that the final combustion products at complete combustion were water vapor and carbon dioxide. Equivalence ratios were calculated not only from the measured air and fuel flow rates but also from the gas compositions measured for the area averaged gas samples. The deviations between the equivalence ratios by area-averaged sampling and by single-point sampling at the center of the combustor cross-section were less than 5% for all conditions where measurements were made by both sampling procedures. This suggests that at the sampling position the profiles of fuel–air ratio along the diameter were fairly uniform and that the samples with the area-averaged gas sampling probe should be representative of the combustion gas. The calculated equivalence ratios by the area-averaged gas analysis agreed with that from the measured fuel and air flow rates in accuracies better than 2% at lean conditions though the deviation increased to 5% at fuel–rich conditions. It was thought that most of the deviation at fuel–rich conditions should be attributed to the variation of mixing ratio of the capillary flow mixer, which depend to some extent...
on the compositions of sampled gas. In this paper, equivalence ratios based on the area averaged gas sampling are used for the presentation of the data obtained in the present study.

TEST RESULTS

FLAME STABILITY LIMITS

In the direct injection system, combustion air was almost equally split between inner and outer channels of the double swirler. Thus the equivalence ratio of the mixture jet issuing from the inner channel was about twice as the overall equivalence ratio. As a result, it was expected that the direct injection system would hold flames at much leaner conditions than the premixed-prevaporized system. Unexpectedly, blow-off of both systems occurred at about the same equivalence ratio for a given inlet air temperature. This observation suggests that in the direct injection burner, very rapid mixing of injected fuel with air should have occurred before reaching the combustion zone, otherwise flames could be sustained at very lean conditions as is the case with the conventional swirl-stabilized burners.

CO EMISSIONS

The effect of burner configurations on the CO emissions are shown in Figure 2. For both systems, the CO emissions levels decreased with decreasing equivalence ratio, the trend being generally in agreement with the CO equilibrium behaviors. It is clearly shown that, over the whole range of equivalence ratios and inlet air temperatures tested, the CO emissions levels measured for the premixed-prevaporized system are higher than those for direct injection system. The leaner was the mixture, the more appreciable was the difference between the emission levels of the two systems. The direct injection system achieved CO emissions levels less than 10 EICO at lean conditions for all inlet air temperatures. The effects of fuel nozzle capacity and premixing tube length were not consistent though they were small. The effect of inlet air temperature on CO emission levels was appreciable for the premixed-prevaporized system, but less appreciable for the direct injection system. This suggests that quenching was more appreciable for lean premixed-prevaporized combustor at lower inlet air temperatures. The low-CO emission levels measured for the direct injection...
Figure 3 shows emissions levels of unburned hydrocarbons as a function of equivalence ratio for different inlet air temperatures. The HC emissions levels with the direct injection system were lower than those with the premixed–prevaporized system at all inlet air temperatures, being lower than $10^{-1}$ EIHC over a wide range of equivalence ratio. The spray was enveloped by the outer swirling air jet and, therefore, impingement of fuel droplets on the combustion chamber wall was suppressed. In the premixed–prevaporized burners, some portion of the atomized fuel reached the wall and the re-atomization of the fuel was insufficient. Additionally, fuel droplets were likely to reach the combustion chamber wall. Thus, the HC emissions were higher at lower inlet air temperatures. For the premixed–prevaporized system, higher HC emissions levels were measured for the burner with the shorter premixing tube and the larger fuel nozzle capacity at inlet air temperatures of 450 and 550 K. At 650–K inlet air temperature, the emission levels were very low (less than $10^{-1}$ EIHC) over a wide range of equivalence ratios and the effects of fuel nozzle capacity and premixing tube length were insignificant, suggesting that prevaporization of atomized fuel should be fairly complete for the premixed–prevaporized system at the inlet air temperature.

The mean droplet size increases with increasing fuel nozzle capacity, and prevaporization of the atomized fuel is less for the shorter premixing tube length. The measured trends of HC emissions levels with fuel nozzle capacity and premixing tube length was not necessarily consistent. The effects of inlet air temperature on HC emissions were found to be appreciable. An increase in the inlet air temperature promoted prevaporization of fuel droplets on one hand and decreased quenching of HC oxidation on the other hand, resulting in lower emissions levels.

**NOx EMISSIONS**

The NOx emissions levels measured at inlet air temperatures of 450, 550 and 650 K are shown in Figure 4. The variation...
of NOx emissions levels with equivalence ratio for premixed-
prevaporized burners is similar to that for direct injection
burners. The emission levels decreased appreciably with
decreasing equivalence ratio, with the variations being more
than 10 times over the whole range of equivalence ratios tested.
Each curve for the NOx emissions data had a peak at equiv-

calence ratio very close to unity where flame temperatures should
be maximum, supporting the validity of area-averaged gas
sampling. At a given equivalence ratio, the level of NOx
emissions for the premixed-prevaporized system was lower than
that for the direct injection system. For the premixed-

prevaporized system, however, the burners with the shorter
mixing tube achieved the lowest NOx emissions levels. The
effect of nozzle capacity was small and inconsistent. Therefore,
the order of the measured NOx emissions levels was not always
consistent with the expected order of mixture homogeneity.
Detailed measurements of gas temperature and species concen-
trations in the combustion region and fuel prevaporization are
needed for the explanation of the observed order.

RELATION BETWEEN NOx EMISSIONS LEVELS AND
COMBUSTION EFFICIENCIES

In lean combustion, emissions of CO and hydrocarbons are
likely to be unacceptably high at lean conditions where ultra
low-NOx emissions are obtained. Therefore, a compromise
between NOx emissions levels and combustion efficiency is
needed in designing low-NOx combustors.

The emissions performance of low-NOx combustion systems
should be compared by examining the correlations between NOx
emissions levels and combustion efficiencies. In Figures 5, the
measured indices for NOx emissions, EINOx are plotted against
combustion efficiency. It can be seen from the correlation
curves that on the lean side, combustion efficiency increased
while EINOx decreased. Thus, both systems have the capability
of attaining low-NOx emissions levels without sacrificing
combustion efficiency. A comparison of the correlations
between NOx emissions and combustion efficiency, however,
clearly shows that the direct injection combustion system is
superior to the premixed-prevaporized combustion system in
that it can achieve very high combustion efficiency. The

![Figure 4. NOx Emissions for Direct Injection and Premixed-Prevaporized Combustion Systems](http://mechanicaldesign.asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1995/78804/V003T06A048/2406198/v003t06a048-95-gt-276.pdf)
difference between the characteristics of both systems was more significant at lower inlet air temperatures.

The EINOx—combustion efficiency correlations for diffusion combustion of premixed kerosene vapor–air mixtures are presented in Figure 5(d) for comparison. The diffusion combustion system was prepared by replacing the block attached to the premixed–prevaporized system with an axial vane swirler. Mixtures of fuel vapor and air were injected into the combustion chamber, being surrounded by a swirling air flow. With decreasing equivalence ratio, EINOx as well as combustion efficiency decreases. Similar correlations were observed with conventional swirl stabilized spray burners. A decreases in equivalence ratio on the fuel–lean side lead not only to a reduction in NOx emissions levels but also in combustion efficiency. Appreciably low NOx emissions were attained only at the expense of combustion efficiency. The low combustion

FIGURE 5. NOx EMISSIONS VS COMBUSTION EFFICIENCIES FOR DIRECT INJECTION, PREMIXED-PREVAPORIZED AND DIFFUSION COMBUSTION SYSTEMS
efficiency at fairly lean conditions is due to the quenching of CO and unburned hydrocarbon oxidation reactions by low temperature air from the swirler.

**NOx EMISSIONS LEVELS VS GAS TEMPERATURES**

It is well known that flame temperature is the major parameter that influences NOx emissions levels. The NOx emissions levels measured for some burner configurations are plotted against adiabatic equilibrium gas temperatures in Figure 6. The log of EINOx for the direct injection and the premixed-prevaporized systems was increasing linearly with adiabatic gas temperature. For the premixed-prevaporized system, fuel evaporation was promoted and mixtures were increasingly homogeneous at higher inlet air temperatures, and the slope was increasingly steep with increasing inlet air temperatures. The dependence on the gas temperature for the direct injection
A comparison between the present experimental data shows that the NOx emissions level with the direct injection burners was about as twice as that for premixed–prevaporized burners, and that the levels with the diffusion combustion system was, in spite of incomplete combustion, much higher at lean conditions. The low–NOx emissions levels achieved by lean premixed–prevaporized combustion can be attained by lean direct injection with higher combustion efficiencies when the operating condition is shifted to a leaner side.

All NOx emissions data for premixed–prevaporized combustion shown in the figure are well correlated to gas temperature by liner lines and their dependency on gas temperature was about the same: EINOx is proportional to \( \exp\left(8 \times 10^3 \frac{T_b}{K}\right) \), where \( T_b \) is gas temperature. However, there is a great discrepancy between the present and previous NOx emissions levels for premixed–prevaporized combustion. The kerosine–air mixtures in the present experiments were not homogeneous even at the highest inlet air temperature while the author's previous data were measured for completely vaporized octane–air mixtures in a flame tube of the "hot wall" construction using a single–hole gas sampling probe on the flame tube center. A probable reason for the low levels of the present NOx emissions data was the increased heat losses to the water–cooled combustion chamber wall. NOx formation might be appreciably depressed in the reaction region close to the combustion chamber wall and also downstream of the combustion zone. As a result, the effective residence time for NOx formation was surely shorter than the calculated value. It is noted that the NOx concentrations for area–averaged gas samples were by 20–30% higher than those for gas samples sampled at the center of the combustion chamber cross section, which seems consistent with the fact that equivalence ratio was higher for area averaged gas sampling than for single–point sampling at the center.

The low HC and CO emissions of the direct injection system are due to the increased mixing of fuel and air in the combustion zone. The idea was partly supported by the uniform profiles of gas temperature and species concentration in the combustion zone of the double swirler/fuel nozzle module measured using methane as a fuel.

The data with the direct injection system seem to be encouraging for developing low–NOx and low–CO emissions combustors for gas turbines of high–pressure and high–temperature operating conditions though the evaluation of the emission characteristics of the burners under actual conditions and the development of fuel–air control mechanism are needed.

CONCLUSIONS

The compatibility between low–NOx emissions and high–combustion efficiency (low CO and HC emissions) was demonstrated by using innovative double swirler/fuel nozzle...
module which directly injects kerosine spray into the combustion zone. When compared at the same equivalence ratio, combustion was more complete in the direct injection system than in the premixed prevaporized system, the difference being appreciable at lower inlet temperatures. When operated at leaner conditions than the premixed–prevaporized system the direct injection system achieved comparably low NOx emissions levels as well as higher combustion efficiency.

The log of EINOx for both systems was increasing linearly with adiabatic gas temperature. The dependence on the gas temperature for the direct injection system was as strong as for the most homogeneous mixtures in the lean–premixed–prevaporized systems in the present and previous studies. This was attributed to the rapid mixing of fuel spray and air in the combustion zone by the double swirler/fuel nozzle module. The effects of fuel nozzle capacity and mixing tube length on the levels of HC were generally consistent with the expected completeness of fuel prevaporization in the mixing tubes while those on NOx and CO were not.

REFERENCES


