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PERFORMANCE OF A REDUCED NO_x DIFFUSION FLAME COMBUSTOR FOR THE MS5002 GAS TURBINE

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ABSTRACT

This paper describes a reduced NO_x diffusion flame combustor that has been developed for the MS5002 gas turbine. Laboratory tests have shown that when firing with natural gas, without water or steam injection, NO_x emissions from the new combustor are about 40% lower than NO_x emissions from the standard MS5002 combustor. CO emissions are virtually unchanged at base load, but increase at part load conditions. The laboratory results were confirmed in 1997 by a commercial demonstration test at a British Petroleum site in Prudhoe Bay, Alaska.

The standard MS5002 gas turbine is equipped with a conventional, swirl stabilized diffusion flame combustion system. The twelve standard combustors in an MS5002 turbine are cylindrical cans, approximately 27 cm (10.5 inches) in diameter and 112 cm (44 inches) long. A small, annular, vortex generator surrounds the single fuel nozzle that is centered at the inlet to each can. The walls of the cans are louvered for cooling, and contain an array of mixing and dilution holes that provide the air needed to complete combustion and dilute the burned gas to the desired turbine inlet temperature.

The new, reduced NO_x emissions combustor (referred to as a "lean head end", or LHE, combustor) retains all of the key features of the conventional combustor; the only significant difference is the arrangement of the mixing and dilution holes in the cylindrical combustor can. By optimizing the number, diameter, and location of these holes, NO_x emissions were substantially reduced. The materials of construction, fuel nozzle, and total combustor air flow were unchanged.

The differences in NO_x emissions between the standard and LHE combustors, as well as the variations in NO_x emissions with firing temperature, were well correlated using turbulent flame length arguments. Details of this correlation are also presented.

INTRODUCTION

The simple cycle MS5002 gas turbine is a mechanical drive model which is widely used in pumping applications. The MS5002 turbine has two shafts (a high pressure turbine powers the air compressor and a low pressure turbine drives the desired load), a pressure ratio of about 8.5:1, a compressor discharge temperature of about 290°C (555°F), and a net power output of about 28 MW. NO_x emissions from an unabated, natural gas fired MS5002B turbine equipped with standard combustors are about 100 ppmv (on a dry, 15% O_2 basis) at base load and ISO conditions. The base load combustor exit temperature for an MS5002B is 953°C (1747°F) at ISO conditions. The MS5002C turbine is an uprated version of the MS5002B unit with a 39°C (70°F) increase in firing temperature.

Increasingly stringent environmental regulations continue to drive emissions reductions from almost all combustion turbines. Dry low NO_x (DLN) combustion systems (that is, systems that do not require water or steam injection) remain one of the most popular methods of reducing emissions, because of the demonstrated poten-

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tial for high reliability, minimal impact on turbine performance, and extremely low emissions. These systems routinely achieve emissions of 25 ppmv NO_x (on a dry, 15% O_2 basis) or less, usually through lean premixing of the fuel and air to eliminate high temperature, stoichiometric combustion zones. One drawback is that lean premixed combustion systems are often more complex and, therefore, more costly than the older style, diffusion flame combustors they replace.

Although DLN combustors are generally preferred for new units, there are applications in which the low emissions (and high cost) of lean premixed combustion systems are not essential. For example, when modifying an existing turbine to increase turbine power output and/or efficiency (often by increasing the pressure ratio and/or turbine inlet temperature), frequently the only requirement is that emissions do not increase after the modifications. Increasing the pressure ratio and/or turbine inlet temperature will tend to increase NO_x emissions, but since the available modifications to existing units are often relatively minor, the resulting increases in emissions are typically small (10 - 15%). In these situations, there exists a need for low cost combustion system modifications with relatively modest emissions reductions.

To fill this need, GE has been developing lean head end (LHE) combustors. Although somewhat similar in appearance to the standard combustors they replace, NO_x emissions from LHE combustors are about 40% lower. Nevertheless, combustion occurs in a swirl-stabilized turbulent diffusion flame in both combustors.

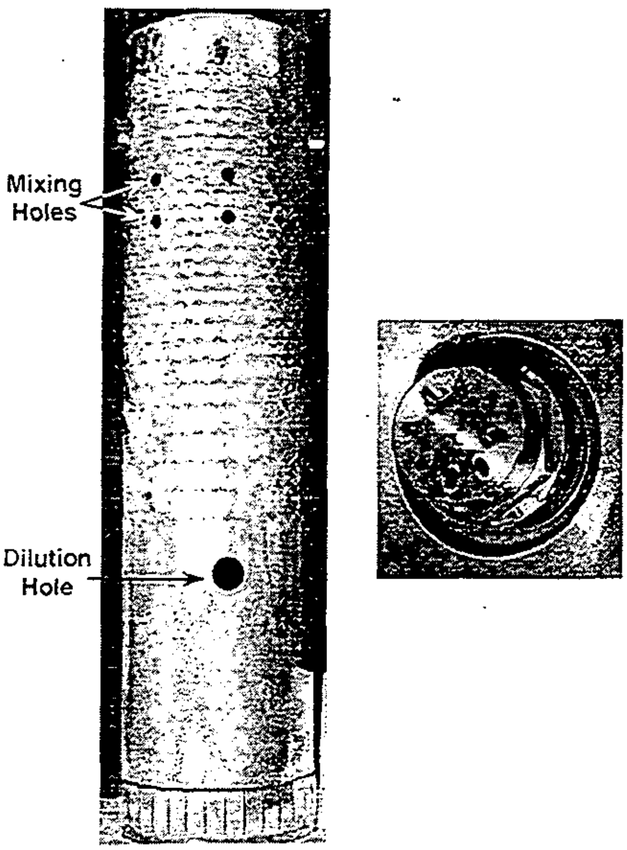


Fig. 1 Standard MS5002 combustor liner (left) and gas fuel nozzle (fight)

The essence of the LHE concept is best understood by comparing the standard and LHE MS5002 combustors. The MS5002 turbine is equipped with twelve cylindrical can-type combustors (see Figure 1). Each standard (and LHE) combustor is approximately 27 cm (10.5 inches) in diameter and 112 cm (44 inches) long. A small, annular, vortex generator surrounds the single fuel nozzle that is centered at the inlet to each can. The walls of the cans are louvered for cooling, and contain an array of mixing and dilution holes that provide the air needed to complete combustion and dilute the burned gas to the desired combustor exit temperature.

The MS5002 LHE combustor retains all of the key features of the conventional combustor; the only significant difference is the arrangement of the mixing and dilution holes in the cylindrical combustor can. The number and diameter of the mixing holes have been increased; both the mixing holes and the dilution holes have been moved closer to the fuel nozzle, resulting in a more fully aerated diffusion flame. This reduces both the turbulent flame length and the time in the flame spent at stoichiometric conditions, thereby reducing NO_x emissions. For a given firing temperature, the total combustor air flow and the overall fuel/air ratio in the combustor are unchanged.

Development of an LHE combustor for the MS5002 turbine began in early 1997. After a series of laboratory tests at GE Corporate Research and Development, a turbine test was successfully conducted in September 1997 at a British Petroleum site in Prudhoe Bay, Alaska. To minimize unit-to-unit variation, the same turbine was tested with both standard combustors and LHE combustors. LHE combustors are now being developed for the simple cycle MS3002 turbine series.

LABORATORY TESTING AND RESULTS

The laboratory test stand used for development of the MS5002 LHE combustor is shown in Figure 2 and has been described previously (Maughan et al., 1994). The combustor liner is housed in a 40.5 cm (16 inch) diameter pressure vessel, which is supplied with up to 7.3 kg/s (16 lb_m/s) of preheated, non-vitiated, oil-free air. Air enters the vessel through 24 holes uniformly distributed around the circumference of an air distribution plenum. The air flows over a simulated transition piece and continues upstream, towards the combustor liner and fuel nozzle, mimicking the reverse-flow configuration of the MS5002 turbine. The vessel pressure is set by an orifice plate which has an effective area approximately equal to one-twelfth (since there are 12 combustors) of the total first stage nozzle area in an MS5002 turbine. The backpressure orifice is cooled by a water spray injected into the exhaust gas just upstream of the orifice. Exhaust gas is sampled with an uncooled stainless steel probe located upstream of the water spray and analyzed for CO, CO_2, O_2, and NO_x with continuous emissions analyzers. The gas sample probe extends across a diameter of the duct, and has inlet holes equally spaced along its entire length. Using this arrangement, the sampled gas represents an integrated average across the duct.

Results from laboratory tests of the standard and LHE MS5002 combustors are shown in Figures 3 and 4. Each combustor was tested at two different total air flow rates and pressures, because the total air flow through an MS5002 combustor (about 9.5 kg/s, or 21 lb_m/s at base load) exceeds the capacity of the laboratory air compressors. The variations in air flow between the lab tests of the standard and lean head end combustors are due to day-to-day variations in the settings of the laboratory air compressors. Since lab emissions must be scaled to full pressure regardless of the air flow, no attempt was made to exactly duplicate the air flow from day to day. In general, we find laboratory NO_x emissions are proportional to P^n, where P is the absolute pressure and $0.3 < n < 0.5$. Through testing at two different air flow rates and pressures, we can determine the pressure exponent n for each combustor, and scale lab NO_x emissions to full flow and pressure. This result in not surprising; Lefebvre (1999) recently summarized several studies in which NO_x from diffusion flame combustors was found to be proportional to P^n. Laboratory CO emissions are generally found to be insensitive to total flow rate and pressure, and do not require scaling.

Figure 3 shows that NO_x emissions decrease as the fuel flow and combustor exit temperature decrease. This is typical behavior for diffusion flame gas turbine combustors. At the base load combustor exit temperature for an MS5002B turbine (953°C or 1747°F at ISO conditions) and at similar pressures, lab NO_x emissions decreased from about 94 ppmv (dry, 15% O_2) for the standard combustor to about 58 ppmv (dry, 15% O_2) for the LHE combustor, a decrease of 38%. The percent reduction in NO_x emissions was even larger at part load conditions. A regression analysis of the data in Figure 3 showed that NO_x emissions from the standard combustor were proportional to $P^{0.36}$, while NO_x emissions from the LHE combustor were proportional to $P^{0.40}$.

CO emissions from laboratory tests of the standard and LHE combustors are shown in Figure 4. As expected, CO increases as the fuel flow and combustor exit temperature decrease. LHE CO emissions were substantially higher than CO emissions from the standard combustor at part load conditions, but were similar at base load MS5002C conditions. Figure 4 also shows that laboratory CO emissions are insensitive to the total air flow rate and pressure.

FIELD TESTING AND RESULTS

The first field test of a simple cycle, natural gas fired MS5002 turbine equipped with LHE combustors was conducted during September, 1997, at a British Petroleum site in Prudhoe Bay, Alaska. An MS5002B turbine equipped with standard combustors was tested just before a scheduled outage. LHE combustors were installed during the outage, and the same unit was tested again just after being restarted. During both tests, data points were collected at both base load and part load conditions. The lower load limit was determined by the operating limits of the load compressor being driven by the MS5002B turbine, not by the turbine itself. Turbine operating parameters such as compressor inlet temperature, compressor discharge temperature, fuel flow rate, etc., were collected from the turbine data logging system. After the tests were completed, these parameters and the measured natural gas composition at the site (see Table 1) were used to calculate the combustor exit temperature for each field test point.

As might be expected from the lean head end design, the fuel flow rate required for ignition increased by approximately 5% after the LHE combustors were installed. No problems with lean blow out, crossfiring, or stability were detected during startup or acceleration. Part load and base load operations were also uneventful.

A gas sample probe was inserted into an existing flanged port on the turbine exhaust stack and positioned along a stack diameter. The sample probe had 3 mm (1/8 inch) diameter inlet holes, spaced approximately 30 cm (12 inches) apart, along its entire length. In this manner the gas sample approximated an integrated average across the exhaust stack. The sample gas was dried and sent to a portable gas analyzer (an Environmental Equipment, Inc. IMR 2800-P) which measured concentrations of CO, O_2, NO, and NO_2. Results from the field tests are shown in Figures 5, 6, and 7.

As expected from the laboratory tests, base load NO_x emissions from the LHE combustor were about 36% lower than NO_x emissions from the standard combustor, with the percent reduction increasing at part load conditions (see Figure 5). The dashed lines in Figure 5 rep-

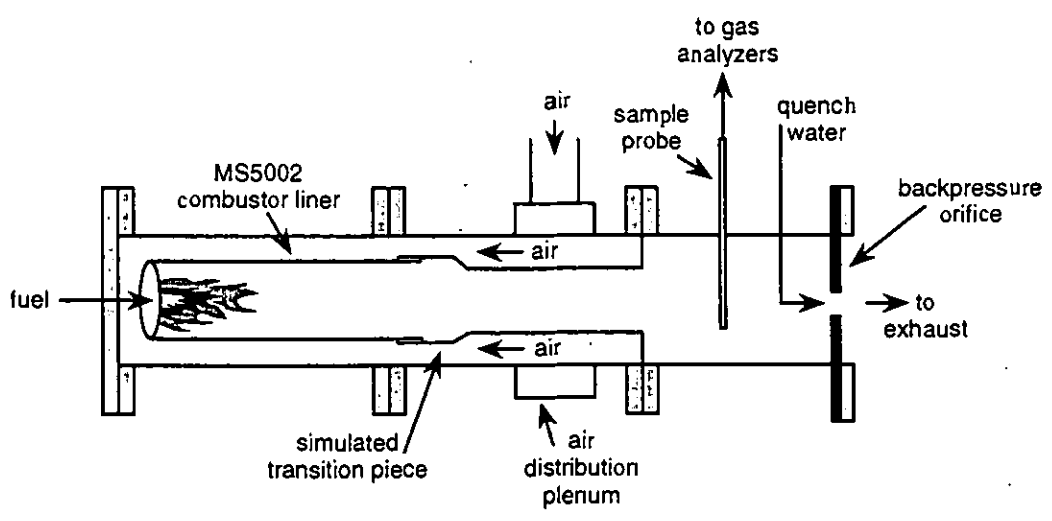
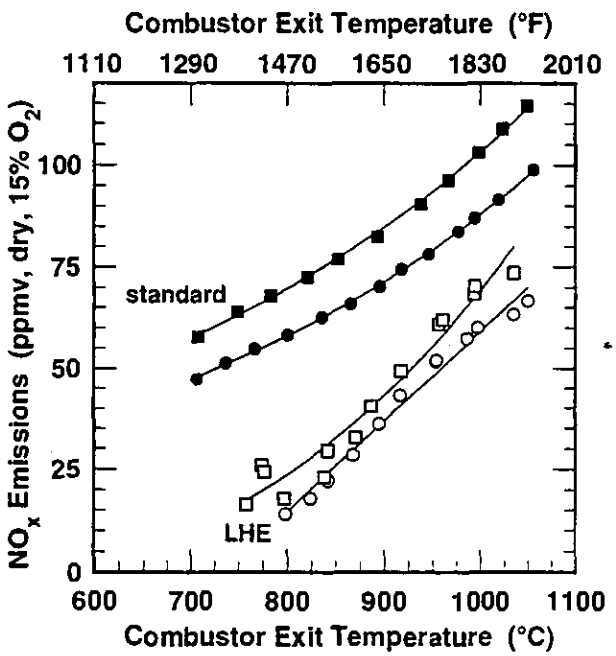


Fig. 2 The laboratory MS5002 combustor test stand

resent the NO_x emissions expected in the field, based upon the laboratory data from the standard and LHE combustors. Reduced pressure lab NO_x emissions were first scaled to full field pressures using the P^n pressure dependence determined in the lab tests. Additional corrections were then applied to account for the differences in combustion air temperature and fuel composition between the lab and the field.

To apply these corrections, the stoichiometric flame temperature was calculated for each lab and field test point. Flame temperatures were calculated using *CET89*, the NASA chemical equilibrium code (McBride, 1989). Stoichiometric flame temperatures were calculated because in an diffusion flame, most of the NO_x can be assumed to have formed through the thermal NO_x reaction pathway at the stoichiometric fuel/air interface. If this assumption is correct, the rate of



Symbol	Combustor	Air Flow (kg/s)	Pressure (bar)
■	standard	5.4	5.0
●	standard	3.3	3.1
□	LHE	6.0	5.4
○	LHE	4.2	3.8

Fig. 3 Laboratory measurements of NO_x emissions from standard and LHE combustors at reduced total flows. Combustion air temperature = 294 ± 3°C for all points. Combustor exit temperature for the MS5002B turbine is 953°C at base load ISO conditions.

NO_x formation should follow an Arrhenius-type rate expression of the form:

$$\frac{d}{dt}[NO_x] \propto \exp\left\{\frac{-E_{act}}{R\,T_{stoic}}\right\} \qquad (1)$$

where

$^d/_{dt}[NO_x]$ = NO_x formation rate [moles/(unit volume) (time)]
E_{act} = activation energy for thermal NO_x formation [energy/mole],
R = ideal gas constant [energy/(mole) (K)], and
T_{stoic} = stoichiometric flame temperature [K].

Assuming the residence time in the flame zone is the same in both the lab and the field, Eq. 1 can be used to derive an expression relating the difference in NO_x emissions between the lab and the field to the difference in stoichiometric flame temperature:

$$\frac{NO_{x,\,field}}{NO_{x,\,lab}} = \frac{\exp\left\{-E_{act}/R\,T_{stoic,\,field}\right\}}{\exp\left\{-E_{act}/R\,T_{stoic,\,lab}\right\}} \qquad (2)$$

Rearranging (2) gives

$$NO_{x,\,field} = NO_{x,\,lab}\,\exp\left\{\frac{-E_{act}}{R}\left[\frac{T_{stoic,\,lab} - T_{stoic,\,field}}{T_{stoic,\,lab} \times T_{stoic,\,field}}\right]\right\} \qquad (3)$$

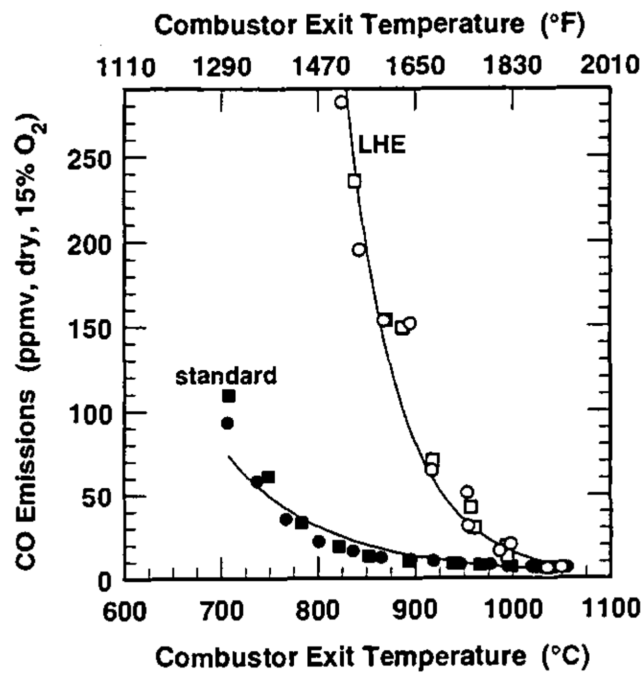


Fig. 4 Laboratory measurements of CO emissions from standard and LHE combustors at the same conditions as shown in Fig. 3

Table 1 Field and laboratory natural gas compositions

Species	Concentration (mole %) Field Test	Laboratory
N_2	0.6	0.6
CH_4	76.0	95.8
CO_2	12.2	0.7
C_2H_6	6.2	2.2
C_3H_8	3.2	0.7
isobutane	0.4	<0.1
n-butane	0.9	<0.1
isopentane	0.2	<0.1
n-pentane	0.2	<0.1
C_6+	0.1	<0.1
Higher Heating Value (kJ/kg)	40,650	53,700

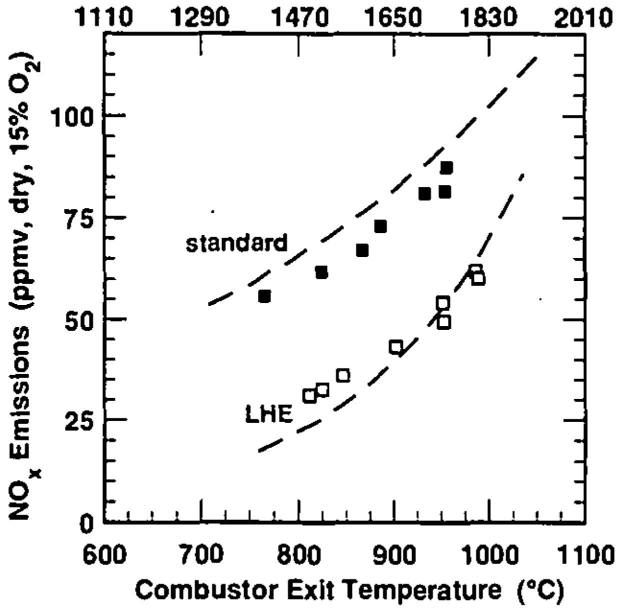


Fig. 5 NO_x emissions from an MS5002B turbine equipped with standard and LHE combustors. Compressor discharge temperature = 269 ± 9°C for all points. Symbols represent field measurements, while dashed lines represent expected values, based on laboratory measurements corrected to field fuel composition, pressure, and compressor discharge temperature.

Equation 3 was used to project field test NO_x emissions from the laboratory measurements (the dashed lines in Figure 5). E_{act} was taken to be 76 kcal/mole based on Hanson and Salimian's 1984 review.

At the same combustor exit temperature, stoichiometric flame temperatures during the field test were 25 - 35°C (45 - 60°F) lower than stoichiometric flame temperatures in the lab. The difference was due in part to the lower compressor discharge temperature in the field (due to the low ambient temperature), and in part due to the lower heating value of the field test natural gas (see Table 1). Figure 5 shows that after applying corrections for pressure and stoichiometric flame temperature, the overall agreement between the lab and field NO_x measurements is fairly good.

Figure 6 shows that most of the emitted NO_x was in the form of NO at base load, with the fractional contribution of NO_2 increasing at part load conditions. This result was not surprising, since CO emissions also increase at part load. Recent studies (e.g., Feitelberg and Correa, 1999) have indicated that if significant concentrations of CO are present, NO can convert into NO_2 relatively rapidly at typical turbine exhaust temperatures. CO emissions measured during the field test are shown in Figure 7. Comparing Figures 4 and 7 shows that the laboratory test stand slightly over-predicts field CO emissions. This over-prediction may be caused by differences in the time-temperature history of the burned gas downstream of the combustor. In the lab, the burned gas is rapidly quenched in the gas sample probe. There may be a greater opportunity for CO burnout to occur during the initial stages of expansion in the turbine than in the gas sample probe.

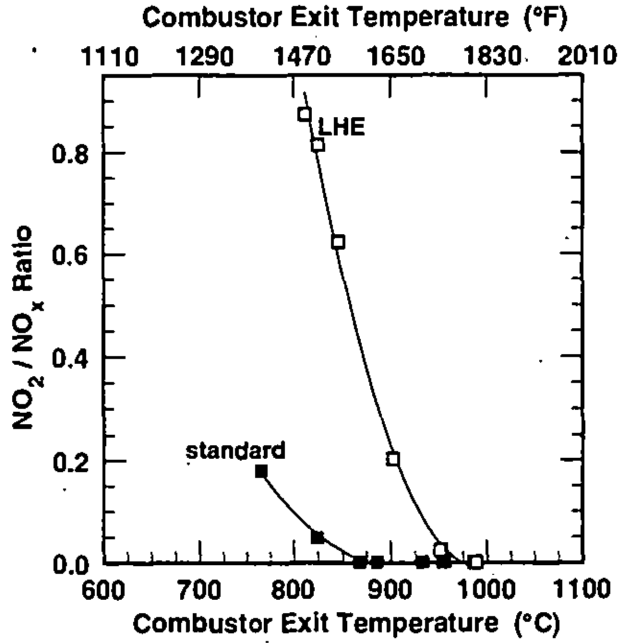


Fig. 6 Ratio of NO_2/NO_x in the MS5002B field test exhaust gas at the same conditions as shown in Fig. 5

DISCUSSION

The LHE MS5002 combustor liner design demonstrates the trade-off between NO_x emissions reductions and increases in CO emissions. At a given firing temperature, the standard and LHE combustors have the same overall fuel/air ratio, but different fuel/air ratios in the primary zone (that is, the zone within the combustor that is closest to the fuel nozzle). Increasing primary zone aeration decreases NO_x emissions but also increases CO emissions. Additional increases in primary zone aeration, beyond the LHE design presented here, would begin to significantly increase base load CO emissions. In this regard, the LHE MS5002 combustor represents a design optimized for the minimum possible NO_x emissions with negligible increase in base load CO emissions.

To aid in the development of LHE combustors for other turbines, we have developed a semi-empirical model which relates liner features to NO_x emissions. This model can be used as a tool for rapidly evaluating the impact of combustor liner design alternatives on NO_x emissions. The starting point for the model is the well known relationship (see, for example, Beér and Chigier, 1983) between flame length (L_f), fuel flow rate (W_{fuel}), and primary air flow rate ($W_{primary\ air}$) for a confined, turbulent diffusion jet flame:

$$L_f \propto \frac{W_{fuel}}{W_{primary\ air}} \tag{4}$$

The key hypotheses in the model are: (1) most of the NO_x is formed at the stoichiometric fuel/air interface, and (2) residence time in the stoichiometric flame zone is proportional to flame length. If these hypotheses are correct, then we should be able to infer a stoichiometric flame zone residence time (τ) from the measured NO_x emissions. Furthermore, this residence time should be proportional to fuel flow rate and inversely proportional to the primary air flow rate.

The first step in applying the model is to infer τ from the NO_x emissions measured in the lab. NO_x formation rates in stoichiometric flame zones were estimated by calculating NO_x emissions from stoichiometric, perfectly stirred reactors (PSRs) at the same pressure and inlet temperature. PSR calculations were performed using the Chemkin II package of programs and subroutines (Glarborg et al., 1986; Kee et al., 1989) and the set of ~250 elementary chemical reactions recommended by Michaud et al. (1992). The calculated relationship between PSR residence time and NO_x emissions for one set of lab conditions (the solid squares ■ in Figures 3 and 4) is shown in Figure 8. Using this approach, a residence time τ can be found that corresponds to each data point shown in Figure 3.

The second step in applying the model is to determine the primary air flow rate. Figure 9 shows the axial distribution of effective area, expressed as a percentage of the total effective area, for both the standard and LHE combustors. Effective areas of distinct liner features (louvers, mixing holes, etc.) were measured separately in a flow stand at atmospheric pressure and temperature. The cumulative effective area at each axial location was then determined by a simple

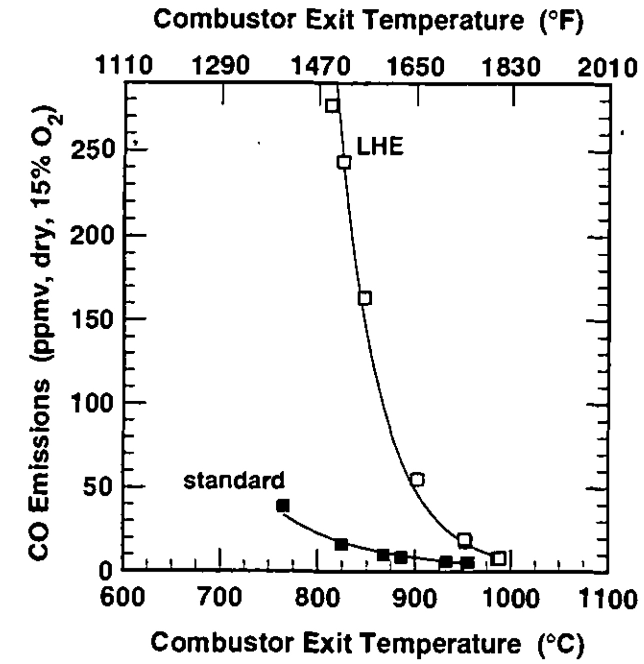


Fig. 7 CO emissions from an MS5002B turbine equipped with standard and LHE combustors at the same conditions as shown in Figs. 5 and 6

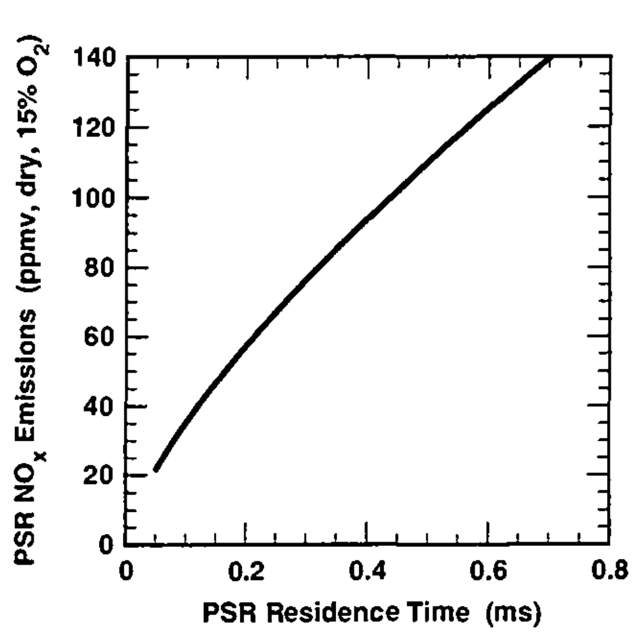


Fig. 8 Calculated NO_x emissions from a stoichiometric PSR for the high air flow rate, standard combustor conditions shown in Fig. 3 (P = 5.0 bar, combustion air temperature = 294°C)

linear combination of the liner features and open areas upstream of that location. The step increases in effective area shown in Figure 9 correspond to mixing holes and dilution holes. The gradual increases in effective area in other portions of the curves correspond to regions of the liner with louvers, but no holes. For reasons that will be shown below, the primary air will be defined as the air that enters the liner within one combustor diameter (27 cm or 10.5 inches) of the fuel nozzle. With this definition, primary aeration increased from about 51% for the standard combustor to about 69% for the LHE combustor. Using this definition, and the known test conditions, $W_{fuel}/W_{primary\ air}$ can be calculated for each data point shown in Figure 3.

Figure 10 shows the resulting linear correlation between residence time τ and $W_{fuel}/W_{primary\ air}$ for the standard and LHE combustor laboratory test NO_x data. The correlation is remarkably good ($R^2 = 0.99$) if primary air is defined as described above. Figure 11 shows the value or R^2 that would be calculated for a range of definitions of the primary zone. The model is most nearly linear, as predicted by the hypotheses outlined above, when the primary zone is defined as the region of the combustor within one combustor diameter (27 cm or 10.5 inches), of the fuel nozzle.

One somewhat surprising observation from this model is that a single definition of primary zone is suitable for both the standard and LHE combustors. Since the same definition of the primary zone is acceptable for both combustors, the proportionality constant implied by Eq. 4 is about the same for both combustors. Significant changes in the number, location, and diameter of the holes in the liner, especially holes located in the primary zone, might be expected to lead to different values of this proportionality constant for each combustor. Figure 10 shows that this is not the case, and the modeling approach outlined here is relatively insensitive to these aerodynamic difference.

The value of this model is that by measuring NO_x emissions from a single liner design over a range of fuel flow rates, the effect of varying the hole diameter and location (and, therefore, the primary zone aeration) can also be evaluated. Other, more radical liner modifications would probably not be well fit with this approach. Modifications that resulted in significant changes to the aerothermal pattern, such as partially premixing the fuel and air, or dramatically changing the swirl number, would fall into this category.

The methodology for applying the model is fairly straightforward. Figures 8 and 10 can be used to estimate NO_x emissions from combustor liners with designs similar to the designs presented here, with primary aerations between 50 and 70% and over a range of combustor exit temperatures. First, the cumulative effective area at the 27 cm (10.5 inch) axial location must be estimated for the proposed design. $W_{fuel}/W_{primary\ air}$ can then be calculated. Figure 10 can then be used to estimate the residence time, and Figure 8 indi-

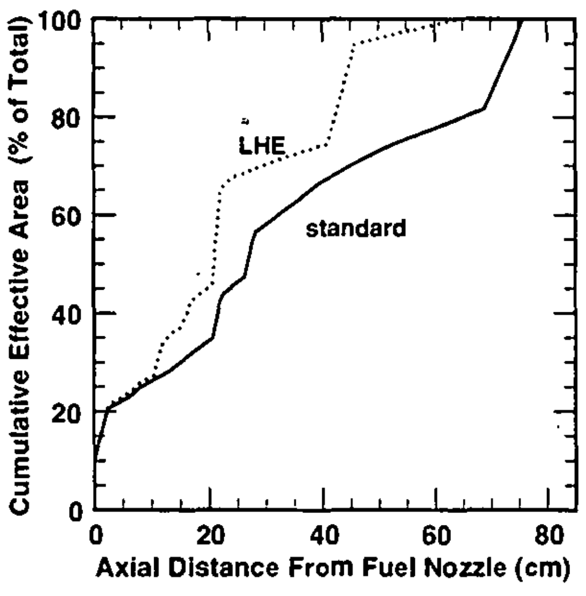


Fig. 9 Axial distribution of effective area in the standard and LHE MS5002 combustor liners

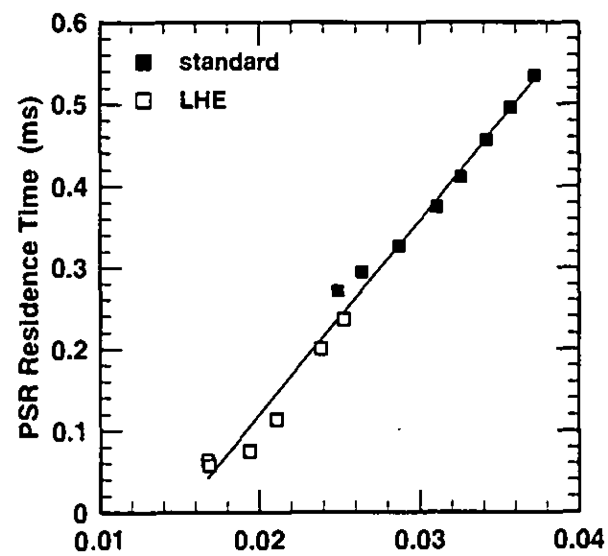


Fig. 10 Correlation between PSR residence time and $W_{fuel}/W_{primary\ air}$ when the primary air is defined as the air entering the combustor through the first 27 cm of the combustor liner. Data points correspond to the high air flow rate, standard and LHE combustor laboratory NO_x measurements shown in Fig. 3. The straight line is a best fit through the data. With this definition of primary air flow, $R^2 = 0.99$.

cates the expected NO_x emissions. Of course, if the pressure and/or inlet temperature are changed, then Figure 8 must be recalculated at the new conditions. This model is semi-empirical because the slope of the line in Figure 10 will also change if the pressure and/or inlet temperature change. Radical changes to the aerothermal pattern are also likely to change the slope of the line. Since the slope cannot be predicted *a priori*, NO_x emissions data will be required from at least one liner design, either standard or LHE, to determine the slope of the line.

CONCLUSIONS

A diffusion flame combustor has been developed with ~40% lower NO_x emissions than the standard combustor for the natural gas fired, simple cycle MS5002 turbine. This new lean head end (LHE) combustor design offers modest NO_x reductions with a negligible increase in base load CO emissions, and at a substantially lower cost than lean premixed combustors. The LHE combustor is intended primarily for the retrofit market, where moderate NO_x reductions are needed to offset NO_x increases that would otherwise occur when uprating an existing unit.

Reduced flow laboratory development tests are in reasonably good agreement with the results from the first field test. The differences in NO_x emissions between the standard and LHE combustors, as well as the variations in NO_x emissions with firing temperature, were well correlated with a semi-empirical model that was developed using turbulent flame length arguments. This model relates combustor liner features to NO_x emissions and can be used as a rapid design tool. GE is currently adapting the LHE combustor concept to the MS3002 series of gas turbines.

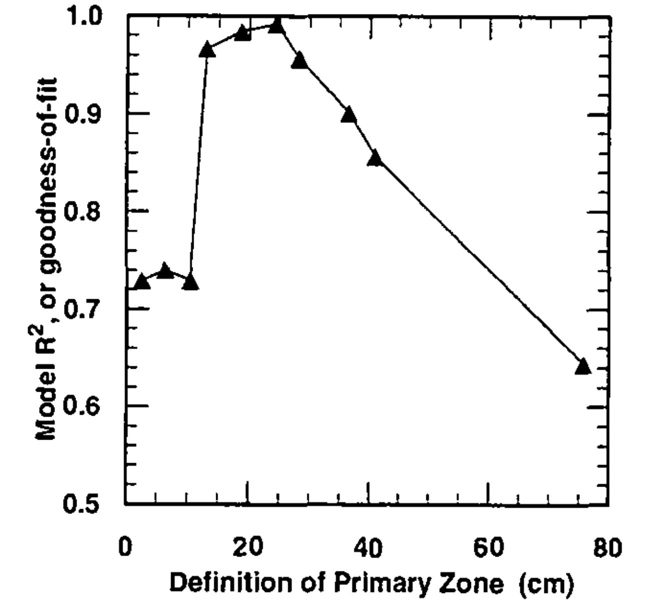


Fig. 11 Model goodness-of-fit for various definitions of the primary zone. The correlation between PSR residence time and $W_{fuel}/W_{primary\ air}$ is most nearly linear when the primary air is defined as the air entering the combustor through the first ~27 cm (10.5 inches) of the combustor liner. To generate this Figure, the data points in Fig. 10 were recalculated with the range of primary zone definitions shown here, and then refit to a straight line.

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