

# A Novel Low NO<sub>x</sub> Lean, Premixed, and Prevaporized Combustion System for Liquid Fuels

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*Dry low emission (DLE) systems employing lean, premixed combustion have been successfully used with natural gas in combustion turbines to meet stringent emission standards. However, the burning of liquid fuels in DLE systems is still a challenging task due to the complexities of fuel vaporization and air premixing. Lean, premixed, and prevaporized (LPP) combustion has always provided the promise of obtaining low pollutant emissions while burning liquid fuels, such as kerosene and fuel oil. Because of the short ignition delay times of these fuels at elevated temperatures, the autoignition of vaporized higher hydrocarbons typical of most practical liquid fuels has been proven difficult to overcome when burning in a lean, premixed mode. To avoid this autoignition problem, developers of LPP combustion systems have focused mainly on designing premixers and combustors that permit rapid mixing and combustion of fuels before spontaneous ignition of the fuel can occur. However, none of the reported works in the literature has looked at altering fuel combustion characteristics in order to delay the onset of ignition in lean, premixed combustion systems. The work presented in this paper describes the development of a patented low NO<sub>x</sub> LPP system for combustion of liquid fuels, which modifies the fuel rather than the combustion hardware in order to achieve LPP combustion. In the initial phase of the development, laboratory-scale experiments were performed to study the combustion characteristics, such as ignition delay time and NO<sub>x</sub> formation, of the liquid fuels that were vaporized into gaseous form in the presence of nitrogen diluent. In the second phase, a LPP combustion system was commissioned to perform pilot-scale tests on commercial turbine combustor hardware. These pilot-scale tests were conducted at typical compressor discharge temperatures and at both atmospheric and high pressures. In this study, vaporization of the liquid fuel in an inert environment has been shown to be a viable method for delaying autoignition and for generating a gaseous fuel stream with characteristics similar to natural gas. Tests conducted in both atmospheric and high pressure combustor rigs utilizing swirl-stabilized burners designed for natural gas demonstrated an operation similar to that obtained when burning natural gas. Emission levels were similar for both the LPP fuels (fuel oils 1 and 2) and natural gas, with any differences ascribed to the fuel-bound nitrogen present in the liquid fuels. An extended lean operation was observed for the liquid fuels as a result of the wider lean flammability range for these fuels compared to natural gas. [DOI: 10.1115/1.2904889]*

## Introduction

Traditionally, spray diffusion combustors have been employed in gas turbines that operate on liquid fuels such as fuel oils 1 and 2. However, this diffusion mode of operation tends to produce unacceptable levels of NO<sub>x</sub> emissions. The current technology for burning liquid fuels in gas turbines is to use water and/or steam injection with conventional diffusion burners. Emission levels for a typical "state of the art" gas turbine, such as GE 7FA burning fuel oil 2 in diffusion mode with water/steam injection, are 42 ppm NO<sub>x</sub> and 20 ppm CO [1]. Water/steam injection has a dilution and cooling effect, lowering the combustion temperature and thus lowering NO<sub>x</sub> emissions. However, at the same time, water/steam injection is likely to increase CO emissions as a result

of local quenching effects. Thus, the "wet" diffusion type of combustion system for liquid fuels must trade off NO<sub>x</sub> emissions for CO emissions.

In recent years, stringent emission standards have made lean, premixed combustion more desirable in power generation and industrial applications than ever before since this combustion mode provides low NO<sub>x</sub> and low CO emissions without water addition. Lean, premixed combustion of natural gas avoids the problems associated with diffusion combustion and water addition. Thus, lean, premixed combustion is the foundation of modern dry low emissions (DLE) gas turbine combustion systems. When operated on natural gas, DLE combustion systems provide NO<sub>x</sub> and CO emissions of 25 ppm or less with no water addition. However, these systems cannot currently operate in a premixed mode on liquid fuels because of autoignition and flashback within the pre-mixing section.

Plee and Mellor [2] characterized autoignition of the fuel/air mixture in the premixer as an important factor that causes flashback in practical combustion devices. Autoignition of the fuel/air mixture occurs before the main combustion zone when the ignition delay time of the fuel/air mixture is shorter than the mean residence time of the fuel in the premixer. Autoignition especially

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Contributed by the International Gas Turbine Institute of ASME for publication in the JOURNAL OF ENGINEERING FOR GAS TURBINES AND POWER. Manuscript received September 24, 2007; final manuscript received October 5, 2007; published online May 30, 2008. Review conducted by Dilip R. Ballal. Paper presented at the ASME Turbo Expo 2007: Land, Sea, and Air GT2007, Montreal, Canada, May 14–17, 2007, Paper No. GT2007-27552.

occurs with the higher-order hydrocarbon fuels, such as fuel oils, which have shorter ignition delay times compared to natural gas [3]. The short ignition delay times of vaporized higher hydrocarbons have been proven difficult to overcome when burning in a lean, premixed mode.

Nevertheless, in order to overcome high  $\text{NO}_x$  levels produced by spray combustion, gas turbine designers still desire to use lean, premixed, and prevaporized (LPP) combustion. Several approaches have been reported in the literature [4–12] to overcome flashback and autoignition in the premixers of LPP combustors. These approaches attempt to achieve low  $\text{NO}_x$  emissions by designing premixers and combustors that permit rapid mixing and combustion before spontaneous ignition of the fuel can occur. In most of the work reported on LPP combustion systems in the literature, the fuel is directly sprayed into the premixer so that the liquid fuel droplets vaporize and mix with air at lean conditions. Typically, swirlers with multiport liquid fuel injection systems are employed for better fuel/air mixing [7]. However, unlike these attempts to alter hardware, there has been no reported work on altering fuel combustion characteristics in order to delay the onset of ignition in lean, premixed combustion systems.

In this study, vaporization of the liquid fuel in an inert environment has been shown to be a technically viable approach for LPP combustion. As described in this paper, a patented fuel vaporization and conditioning process [13] was developed and tested to achieve low emissions ( $\text{NO}_x$  and CO) comparable to those of natural gas while operating on liquid fuels, without water or steam addition. In this approach, liquid fuel is vaporized in an inert environment to create a fuel vapor/inert gas mixture, LPP gas, with combustion properties similar to those of natural gas. A premature autoignition of the LPP gas was controlled by the level of inert gas in the vaporization process. Tests conducted in both atmospheric and high pressure test rigs utilizing typical swirl-stabilized burners (designed for natural gas) found an operation similar to that achieved when burning natural gas. Emission levels were similar for both the LPP gas fuels (fuel oils 1 and 2) and natural gas, with any differences in  $\text{NO}_x$  emissions ascribed to fuel-bound nitrogen present in fuel oil 2. Also, tests showed that the LPP combustion system helps to reduce the  $\text{NO}_x$  emissions by facilitating stable combustion even at very lean conditions when using liquid fuels. An extended lean operation was found for the liquid fuels due to the wider lean flammability range for these fuels compared to natural gas. An added advantage of the fuel vaporization and conditioning process is the ability to achieve fuel interchangeability of a natural-gas-fired combustor with liquid fuels.

The initial phase of development of the LPP system described in this work involved laboratory-scale experiments to study combustion characteristics, such as ignition delay time and  $\text{NO}_x$  formation, of the liquid fuels that are vaporized into gaseous form in the presence of nitrogen diluent. In the second phase of this program, the LPP system [13] was commissioned to perform large-scale pilot experiments on atmospheric pressure and high pressure test rigs. Fuel oils 1 and 2 were chosen as typical liquid fuels for the tests. In this study, fuel oil 1 refers to commercially available kerosene and fuel oil 2, known as diesel fuel 2, is similar to commercially available home heating oil. Also, for a comparative study, experiments were performed with *n*-heptane, which is a widely studied liquid fuel. The chemical and physical specifications of the fuel oils are listed in Table 1. The chemical analysis shows that when compared to fuel oil 1, fuel oil 2 has a higher content of aromatics as well as fuel-bound nitrogen and sulfur.

### Effect of Diluent on Ignition Delay Time

As discussed above, one of the causes of flashback in gas turbine combustors is the premature autoignition of the fuel in air during premixing. One way to avoid autoignition is to extend the induction period, known as the ignition delay time, of the fuel/air mixture by reducing the formation of combustion radicals that are

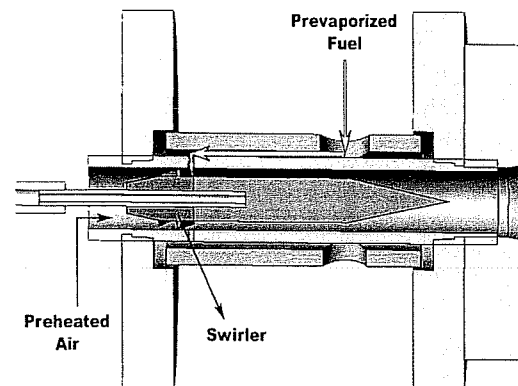
**Table 1 Chemical and physical properties of fuel oils 1 and 2 used in the experiments**

	Fuel oil 1	Fuel oil 2
API gravity at 60°F	45.7	37.6
Specific gravity at 60°F	0.7985	0.8373
Viscosity at 40°C (cS)	1.04	2.6
Molecular weight	150	182
Paraffins (vol %)	83.6	79.3
Aromatics (vol %)	15.7	19.8
Olefins (vol %)	0.7	0.9
C (wt %)	85.78	86.83
H (wt %)	14.15	13.09
N (wt %)	0.02	0.04
S (wt %)	0.03	0.03
H/C molar ratio	1.91	1.85
Chemical formula	$\text{C}_{11}\text{H}_{21}$	$\text{C}_{13}\text{H}_{24}$

necessary for autoignition of the mixture at a given temperature. In hydrocarbon oxidation, during the induction period, the diluent plays an important role as a third-body enhancer for recombination reactions and slows down the radical formation [14]. These effects, in turn, extend the induction period so that the ignition process will be delayed. Therefore, increasing the diluent fraction of the fuel/oxidant mixture will increase the ignition delay time. In the present work, the effect of diluents on the ignition delay time was investigated for vaporized liquid fuels, such as fuel oils and *n*-heptane.

The ignition delay time experiments were performed in an atmospheric pressure flow reactor, which consists of a premixing section and a plug-flow test section. The fuel was vaporized into gaseous form in a preheated nitrogen environment before being mixed with air in the premixing section, which is located at the entrance to the reactor. Figure 1 shows the cross-sectional view of the premixing section. The premixing section opens into the plug-flow test section via an expanding duct in order to prevent the separation of flows at the entrance of the test section. The test section is a  $52 \times 2$  in.<sup>2</sup> alumina tube placed inside a furnace with three independently controlled zone heaters.

One of the difficulties in any flow reactor experiment is quickly mixing the fuel and oxidant at the entrance to the reactor [15,16]. In this work, the rapid mixing of the vaporized fuel with air was achieved by using a swirler followed by an expanding duct that opened into the test section. A photomultiplier coupled with a narrow band filter (centered at 430 nm) was placed at the end of the test section to detect  $\text{CH}^*$  emissions. The time between the



**Fig. 1 Schematic of the premixing section of the flow reactor used for ignition delay time measurements**

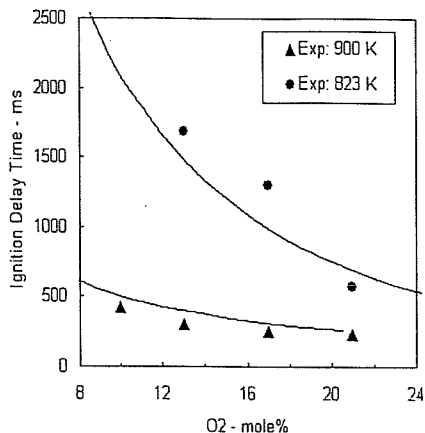


Fig. 2 Atmospheric pressure ignition delay time measurements of stoichiometric *n*-heptane/O<sub>2</sub>/N<sub>2</sub> mixture as a function of the inlet O<sub>2</sub> composition. Key: symbols, experimental data; lines, ignition delay time model predictions using the detailed kinetic model of Curran et al. [18].

injection of fuel vapor into the mixing section and the detection of the CH\* emission signal was designated as the ignition delay time. The plug-flow test section was maintained at a constant initial reactor temperature. The flow reactor pre-mixer temperature was maintained at 600 K in order to simulate the conditions in a gas turbine pre-mixer. A detailed description of the flow reactor facility can be found in a previous publication [17].

Figure 2 shows the ignition delay time as a function of O<sub>2</sub> mole % in the inlet stream for *n*-heptane at 823 K and 900 K. The equivalence ratio of the fuel/oxidant mixture was maintained at stoichiometric conditions. The experimental measurements are compared to ignition delay time model predictions using the detailed kinetics mechanism of Curran et al. [18]. As can be seen in the figure, the effect of diluent on ignition delay time increases with decreasing inlet temperature. For example, for the *n*-heptane measurements shown in Fig. 2, the ignition delay time is proportional to  $[X_{O_2}]^{-0.80}$  at 900 K and to  $[X_{O_2}]^{-2.2}$  at 823 K, where  $[X_{O_2}]$  refers to the mole % of O<sub>2</sub> in the inlet stream to the test section of the flow reactor.

Figure 3 compares the ignition delay time of both vaporized fuel oils with *n*-heptane as a function of O<sub>2</sub> mole % at the inlet to the flow reactor at 900 K. Both fuel oils 1 and 2 show ignition

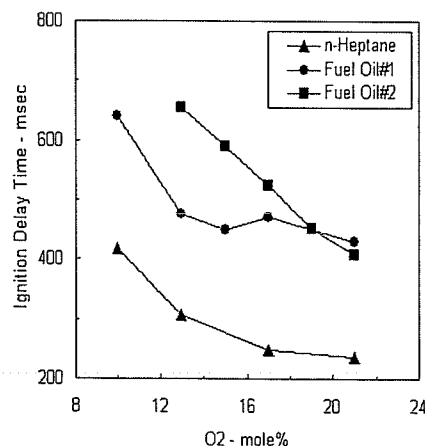


Fig. 3 Comparison of ignition delay time measurements of *n*-heptane, fuel oil 1, and fuel oil 2 as a function of the inlet O<sub>2</sub> composition at 900 K inlet temperature, 1 atm pressure, and 1.0 equivalence ratio

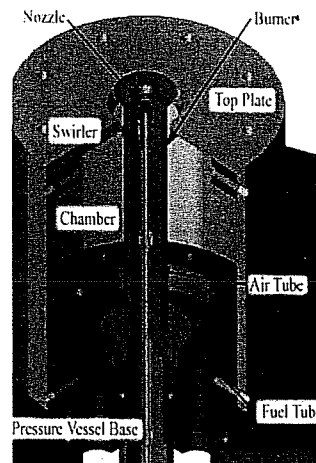


Fig. 4 Cutaway view of the bench-scale, high pressure swirl burner used to investigate NO<sub>x</sub> emissions from the LPP combustion

delay times longer than those of *n*-heptane due to the presence of aromatic hydrocarbons in the fuel oils. Fuel oil 1 shows a similar O<sub>2</sub> dependency on the ignition delay time as that of *n*-heptane. However, the effect of O<sub>2</sub> on the ignition delay time is higher for fuel oil 2 compared to *n*-heptane. For example, for the ignition delay time measurements shown in Fig. 3, the *n*-heptane ignition delay time is proportional to  $[X_{O_2}]^{-0.8}$ , while the fuel oil 2 ignition delay time is proportional to  $[X_{O_2}]^{-1.0}$ .

#### Effect of Pressure on NO<sub>x</sub> Formation

In the LPP system design, the liquid fuel is vaporized in an inert environment to create a gaseous fuel. This gaseous fuel is then premixed with air and burned in a manner similar to lean, premixed natural gas combustion. However, fuel oil, even as a gaseous fuel, is expected to produce slightly higher NO<sub>x</sub> than natural gas fuel, as most fuel oils contain fuel-bound nitrogen. In this work, a series of experiments was performed over a range of pressures up to 10 atm in a bench-scale, high pressure swirl burner to investigate the NO<sub>x</sub> production from the fuel oils when supplied as vaporized liquid fuels in an inert environment. To compare the formation of NO<sub>x</sub> to the vaporized fuel oils, experiments were also performed with natural gas.

The bench-scale LPP combustion system consists of a swirl-stabilized burner where a radially injected gaseous fuel stream is mixed with primary air before the fuel/air mixture reaches the burner surface. A cutaway view of the high pressure swirl burner is shown in Fig. 4. A stand-alone vaporizing system for liquid fuels is directly connected to the fuel port of the swirler to supply the liquid fuel in gaseous form. Nitrogen was used as the diluent for vaporization. The burner was placed in a pressure vessel with a jacket cooling system along the wall. The pressure was varied from 1 atm to 10 atm. The air flow was varied from 1 g/s to 10 g/s with an input power of approximately 1 kW to 15 kW, respectively, for natural gas fuel. A water-cooled sample probe was placed just above the flame zone to draw exhaust gas samples into analyzers to measure CO, CO<sub>2</sub>, and NO<sub>x</sub> emissions. Measurements of CO and CO<sub>2</sub> were obtained using nondispersive infrared (NDIR) ray absorption analyzers, while NO<sub>x</sub> was measured using the chemiluminescence detection method.

Experiments were performed to investigate the effect of diluent on NO<sub>x</sub> formation by varying the N<sub>2</sub> to fuel molar ratio in the vaporizer in a range of 2–10. During the experiments, nitrogen was preheated to 573 K (300°C) to vaporize the liquid fuel in the vaporizer. Figure 5 shows the NO<sub>x</sub> and CO measurements as a

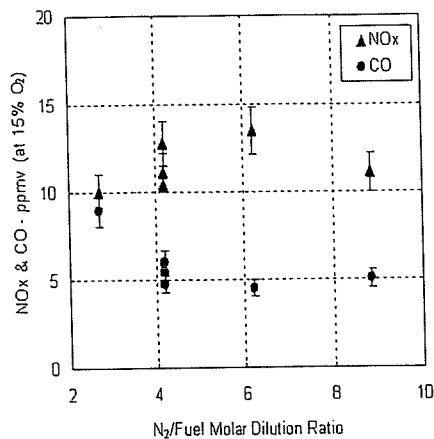


Fig. 5 The effect of  $N_2$  to fuel dilution ratio on  $NO_x$  and CO for fuel oil 2 at 5 atm and 0.6 equivalence ratio

function of diluent to fuel molar ratio for fuel oil 2 at 5 atm and at an equivalence ratio of 0.6. In Fig. 5, CO and  $NO_x$  measurements show that the diluent  $N_2$  to fuel molar ratio does not have any significant effect on CO and  $NO_x$  emissions above a ratio of 4.

Figure 6 shows  $NO_x$  measurements as a function of equivalence ratio for methane, fuel oil 1, and fuel oil 2 at several pressures. Due to the presence of fuel-bound nitrogen, the fuel oils produce higher  $NO_x$  than methane. For methane, the  $NO_x$  measurements increased from 2 ppm to 5 ppm when the equivalence ratio was increased from 0.55 to 0.75 at 2 atm. For fuel oil 2, the  $NO_x$  increased from 5 ppm to 18 ppm when the equivalence ratio was changed from 0.5 to 0.7 at 8 atm. Fuel oils 1 and 2 have 0.02 wt % and 0.04 wt % fuel-bound nitrogen, respectively, as listed in Table 1. Thus, potentially, the fuel-bound nitrogen can produce approximately 4 ppm and 10 ppm of  $NO_x$ , respectively, for fuel oils 1 and 2 at a 0.6 equivalence ratio, assuming that all the fuel-bound nitrogen atoms are converted to  $NO_x$ . At an equivalence ratio of 0.6, methane produced approximately 2 ppm  $NO_x$  at 5 atm, as shown in Fig. 6. The measured  $NO_x$  emissions for fuel oils 1 and 2 were 6 ppm and 10 ppm, respectively, at the

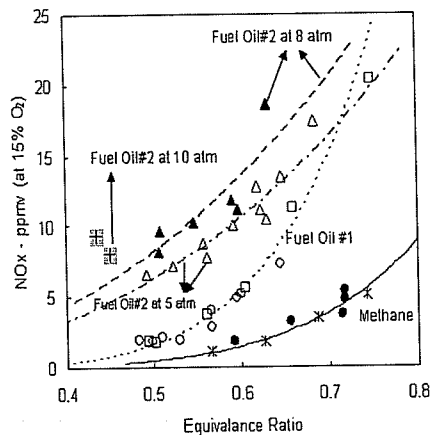


Fig. 6 Comparison of  $NO_x$  measurements for methane and fuel oils 1 and 2 as a function of equivalence ratio at varying pressures. Key: methane at 2 atm (\*) and 5 atm (●) pressures; fuel oil 1 at 3 atm (◇), 4 atm (□), and 5 atm (○) pressures; fuel oil 2 at 5 atm (△), 8 atm (▲), and 10 atm (+) pressures. The lines indicate the trends. The fuel to  $N_2$  molar dilution ratio was 1–5.

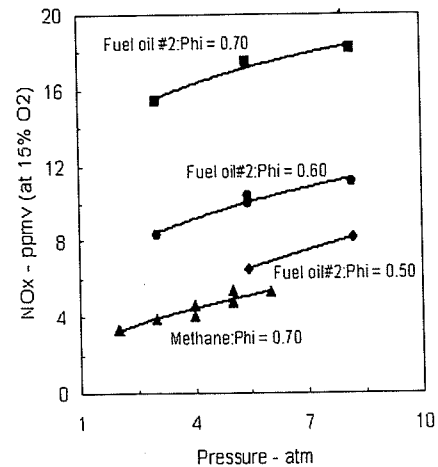


Fig. 7 Comparison of  $NO_x$  measurements for methane and fuel oil 2 as a function of pressure. The fuel to  $N_2$  molar dilution ratio was 1–5.

same conditions. Thus, the difference in  $NO_x$  between the methane and fuel oils can entirely be accounted for by the  $NO_x$  that comes from the fuel-bound nitrogen.

The effects of pressure and equivalence ratio on the  $NO_x$  formation for fuel oil 2 are presented in Fig. 7. The results are also compared to those of methane. The effect of pressure on the  $NO_x$  formation shows a similar trend for fuel oil 2 as for methane. When the equivalence ratio was increased from 0.5 to 0.7, the formation of  $NO_x$  was increased by a multiple of approximately 2 for fuel oil 2, independent of pressure. Three factors contribute to the increase in  $NO_x$  for fuel oils as the equivalence ratio is increased: (1) the amount of fuel-bound nitrogen available for  $NO_x$  conversion is also increased; (2) the increase in flame temperature creates more thermal- $NO_x$  formation; (3) the prompt- $NO_x$  contribution increases as the availability of CH-like radicals increases as the mixture gets richer. The last two factors also contribute to the increase in  $NO_x$  for methane as the equivalence ratio is increased, as shown in Fig. 6. The difference in  $NO_x$  between methane and fuel oil 2 is equivalent to the  $NO_x$  produced by the fuel-bound nitrogen.

From the chemical composition analysis shown in Table 1, the contribution of fuel-bound nitrogen to  $NO_x$  would be 12 ppm for fuel oil 2 at 0.7 equivalence ratio. In Fig. 7, the measured  $NO_x$  for fuel oil 2 at 0.7 equivalence ratio is approximately equal to the  $NO_x$  produced by methane plus 12 ppm fuel  $NO_x$ . These results indicate an apparent quantitative conversion of fuel-bound nitrogen to  $NO_x$  at these lean conditions.

The results given in Fig. 7 show that  $NO_x$  formation in methane and fuel oil 2 exhibits a slight positive dependence on pressure. The effect of pressure on  $NO_x$  emissions during lean, premixed combustion is influenced by many factors [19–23]: degree of fuel-air mixing, equivalence ratio, residence time, and inlet temperature. Leonard and Stegmaier [19], Steele et al. [20], and Bhargava et al. [21] studied the influence of pressure on  $NO_x$  formation in lean, premixed systems but reached differing conclusions. Leonard and Stegmaier [19] concluded that  $NO_x$  emissions are independent of pressure for a perfectly premixed system with a flame temperature in the range from 1700 K to 1900 K. Lean, premixed combustion experiments of Bhargava et al. [21], performed from 7 atm and 27 atm, showed that the pressure effect on  $NO_x$  is negligible up to 0.50 equivalence ratio. However, equivalence ratios larger than 0.5 showed a positive pressure dependence on  $NO_x$ . Bhargava et al. [21] indicated that equivalence ratio and absolute pressure are also important factors that determine the effect of pressure on  $NO_x$ . Conversely, Steele et al. [20], who performed high pressure jet-stirred reactor experiments from

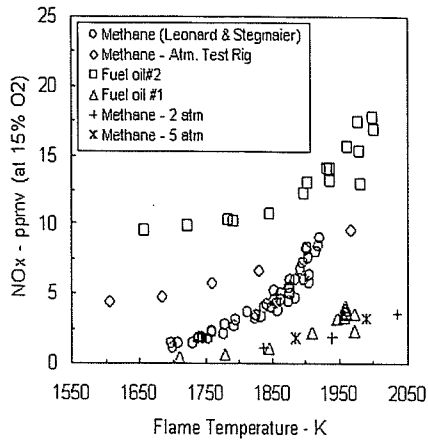


Fig. 8  $\text{NO}_x$  data obtained in the atmospheric pressure test rig for methane, fuel oil 1, and fuel oil 2. The data are compared to the methane data of Leonard and Stegmaier [19] as well as to the  $\text{NO}_x$  data obtained in the high pressure swirl burner described in Fig. 6 at 2 atm and 5 atm.

1 atm to 7 atm, reported a slightly negative pressure dependence on  $\text{NO}_x$ . The results of Steele et al. may be due to the absence of a significant plug-flow residence time in the downstream of the jet-stirred reactor used in their experiments. Thus, based on the experimental data [19,21] reported in the literature and the experimental results discussed in this paper, it can be concluded that  $\text{NO}_x$  formation in a gas turbine combustor is independent of pressure or has a slight positive pressure dependence at very lean conditions. However,  $\text{NO}_x$  emissions will increase with pressure for higher equivalence ratio mixtures.

### Pilot-Scale Testing

The next stage of development of the LPP system involved pilot-scale testing to study emissions and combustion characteristics, such as flame stability and lean blowout (LBO) limits, on realistic turbine hardware. A scaled-up version of the lean, premixed, swirl-stabilized burner used for the high pressure bench-scale tests was designed, built, and tested. The design of this scaled-up burner is similar to that found in modern DLE gas turbine combustion systems [1,24–27]. This large-scale test facility was able to supply up to 0.6 kg/s flow rate of air for the atmospheric pressure tests.

Atmospheric pressure tests were performed with combustion air at typical gas turbine compressor discharge temperatures of 600–620 K. Figure 8 shows  $\text{NO}_x$  emission data obtained in the atmospheric pressure, swirl-stabilized burner for methane, fuel oil 1, and fuel oil 2. Figure 8 also compares the natural gas  $\text{NO}_x$  measurements of Leonard and Stegmaier [19], as well as the  $\text{NO}_x$  data at 2 atm and 5 atm of pressure obtained in the high pressure swirl burner described in Fig. 6. The experimental data of Leonard and Stegmaier [19] was measured between 1 atm and 30 atm. The  $\text{NO}_x$  measurements shown in Fig. 8 for fuel oil 2 are approximately the same as those observed in the high pressure swirl burner experiments. The same figure also shows that fuel oil 1 produced relatively lower  $\text{NO}_x$ , while natural gas gave higher  $\text{NO}_x$  when compared to the high pressure burner results.

### Single Gas Turbine Burner Testing

The next step in the pilot-scale testing was to perform combustion tests on actual turbine hardware at both atmospheric and high pressure conditions. A Solar Turbines Centaur 50 gas turbine fuel nozzle was used for all real hardware tests. This natural gas nozzle was used for the vaporized liquid fuel (LPP gas) tests without any modifications. Tests were conducted at single nozzle, full load conditions for a Centaur 50, a Taurus 60, and a Taurus 70 gas

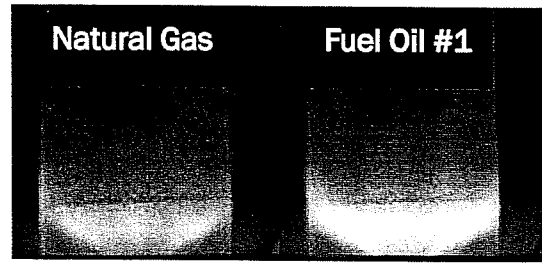


Fig. 9 Comparison of natural gas and fuel oil 1 flames at atmospheric pressure for Centaur 50 fuel nozzle at full load conditions

turbine [27]. During the gas turbine burner tests, the liquid fuel was supplied in gaseous form from the vaporizer.

The testing involved a study of emissions and combustion characteristics, such as flame stability and LBO limits. Both the atmospheric pressure and high pressure tests were performed at typical compressor discharge temperatures. For the high pressure tests, typical compressor discharge pressures were also used. Figure 9 shows a representative atmospheric pressure flame structure for natural gas and for fuel oil 1 from a Centaur 50 fuel nozzle at full load conditions. As can be seen in the figure, the LPP flame with fuel oil 1 exhibits a very similar flame structure and color to that of the natural gas flame.

Figures 10 and 11 show the results of atmospheric pressure testing of a single gas turbine fuel nozzle at Centaur 50 full load conditions for three fuels. Pre-vaporized fuel oil 1 and fuel oil 2 run as LPP gas both show low  $\text{NO}_x$  and CO emissions comparable to those of DLE combustion systems fired on natural gas. The figures show that these low  $\text{NO}_x$  and low CO emissions are simultaneously achieved. As discussed earlier, the primary difference between natural gas and LPP gas  $\text{NO}_x$  emissions can be attributed to the fuel-bound nitrogen present in the fuel oils. Also, during the testing, no flashbacks were observed at any of the test conditions when operating on the fuel oils using the LPP system, and a stable flame was easily maintained when switching fuels from natural gas to LPP gas and back again.

Actual turbine hardware tests were conducted using a high pressure facility capable of testing a single gas turbine fuel nozzle at full compressor discharge temperature and pressure. The LPP liquid vaporizer was used to supply the liquid fuels in gaseous

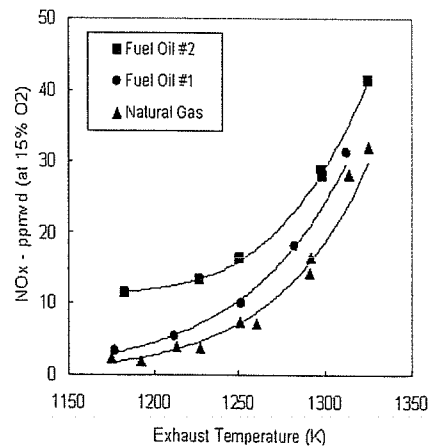


Fig. 10 Comparison of  $\text{NO}_x$  emission measurements for fuel oil 2, fuel oil 1, and natural gas as a function of measured exhaust gas temperature for a single fuel nozzle at Centaur 50 full load conditions (100%). The combustion air temperature was 613 K, the combustor pressure was 1 atm, and the fuel dilution was 6:1 (molar basis).

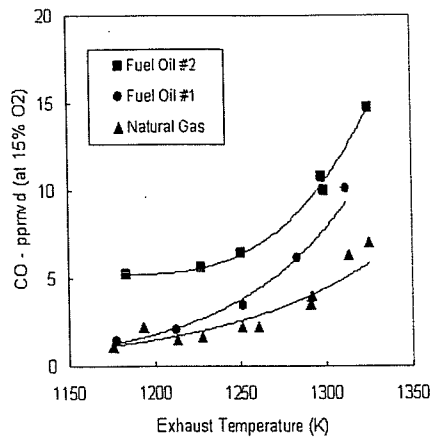


Fig. 11 Comparison of CO emission measurements for fuel oil 2, fuel oil 1, and natural gas as a function of measured exhaust gas temperature for a single fuel nozzle at Centaur 50 full load conditions (100%). The combustion air temperature was 613 K, the combustor pressure was 1 atm, and the fuel dilution was 6:1 (molar basis).

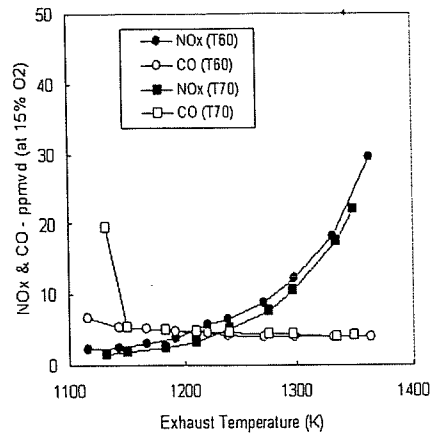


Fig. 13 Comparison of NO<sub>x</sub> and CO emission measurements for fuel oil 2 as a function of measured exhaust gas temperature for a single fuel nozzle at Taurus 60 (T60) and Taurus 70 (T70) full load conditions (100%). The combustion air temperatures were 648 K (T60) and 706 K (T70), the combustor pressures were 12.6 atm (T60) and 16.2 atm (T70), and the fuel dilution was 5:1 (molar basis).

form. The same fuel nozzle used for natural gas testing was also used for liquid fuel testing on LPP gas without any modifications. Figure 12 shows NO<sub>x</sub> and CO emissions at full load conditions for both natural gas and fuel oil 2.

During the testing, emission and dynamics data were taken over a range of lean equivalence ratios from approximately 0.75 to the LBO limit. However, the emission data are plotted against the measured exhaust gas temperature in order to provide a common temperature reference. The lowest temperature data points shown in Fig. 12 reflect the experimentally observed LBO limit. Figure 12 shows that fuel oil 2 LPP gas has an extended LBO limit compared to natural gas and can thus achieve NO<sub>x</sub> emissions nearly as low as natural gas despite the fuel-bound nitrogen.

Figure 12 also shows that the crossover point between NO<sub>x</sub> and CO emissions extends to lower temperatures (and therefore lower equivalence ratios) for fuel oil 2 LPP gas as compared to natural gas. As can be seen from the figure, fuel oil 2 LPP gas showed increased flame stability and an extended LBO limit at lower temperatures (equivalence ratio) compared to natural gas.

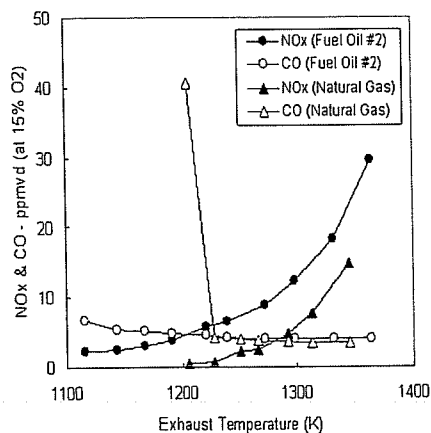


Fig. 12 Comparison of NO<sub>x</sub> and CO emission measurements for fuel oil 2 and natural gas as a function of measured exhaust gas temperature for a single fuel nozzle at Taurus 60 full load conditions (100%). The combustion air temperature was 648 K, the combustor pressure was 12.6 atm, and the fuel dilution was 5:1 (molar basis).

Figure 13 shows comparable NO<sub>x</sub> and CO emissions for both Taurus 60 and Taurus 70 single nozzle full load conditions. The data indicate that similar emissions are achieved, even though the Taurus 70 full load conditions are at higher temperature and pressure than the Taurus 60 operating conditions. Finally, as was observed in the atmospheric pressure tests, these high pressure tests also demonstrate that a stable burner operation was easily maintained when switching fuels from natural gas to LPP gas and back again.

The significance of the data shown in Fig. 12 is that liquid fuels, such as fuel oil 2 LPP gas, are able to achieve low NO<sub>x</sub> emission levels similar to natural gas. For an exhaust temperature (firing temperature) of 1318 K, Fig. 12 shows NO<sub>x</sub> and CO emissions for natural gas to be 9 ppm and 3.5 ppm, respectively. The comparable fuel oil 2 LPP gas emissions at the same exhaust temperature are 16 ppm for NO<sub>x</sub> and 4.0 ppm for CO. Because the LPP gas fuel characteristics are similar to those of natural gas, fuel oil 2 LPP gas is capable of being used in modern DLE gas turbine combustion systems without changes to the burner hardware while achieving much lower NO<sub>x</sub> and CO emissions than fuel oils burned in conventional spray flames with water addition.

## Conclusions

This paper described the development of and results from a patented low NO<sub>x</sub> LPP combustion system for liquid fuels. In the LPP combustion system, liquid fuels were vaporized into gaseous form in an inert environment using nitrogen as diluent. The effect of nitrogen diluent on ignition delay time was measured in a laboratory-scale flow reactor. The experimental results show that diluent nitrogen increased the ignition delay time at typical air/fuel premixing conditions in gas turbines. Also, high pressure, swirl-stabilized burner experiments were performed to study NO<sub>x</sub> formation using the LPP combustion system. The fuel oils in gaseous form were premixed with air and burned to achieve very low NO<sub>x</sub> emissions comparable to those of natural gas in a lean, premixed combustion system. The difference in NO<sub>x</sub> between methane and the fuel oils was attributed to the conversion of fuel-bound nitrogen into NO<sub>x</sub>.

Finally, the test program demonstrated that the LPP combustion system described in this paper was able to produce low NO<sub>x</sub> and low CO emissions without autoignition and flashback. These results were achieved at compressor discharge temperatures and pressures using actual full-scale turbine hardware designed for

natural gas operation without any modifications to the combustor hardware. The test data show that at equivalent exhaust (firing) temperatures,  $\text{NO}_x$  emissions for natural gas and fuel oil 2 LPP gas were 9 ppm and 16 ppm, respectively. CO emissions were 3.5 ppm and 4.0 ppm, respectively. Because the LPP gas fuel characteristics are similar to natural gas, fuel oil 2 LPP gas is capable of being used in modern DLE gas turbine combustion systems without changes to the burner hardware while achieving single digit  $\text{NO}_x$  and CO emissions levels.

### Acknowledgment

The authors would like to acknowledge the LPP Combustion design team for the work described in this paper. Specific thanks go to Maclain Holton, Diwakar Vashishat, Robert Schmidt, Joseph Currano, and David Sykes for their help during the design, building and testing phases of the project. The authors would also like to acknowledge the assistance of Solar Turbines, especially Dr. Ken Smith, Leonel Arellano, and Vu Phi. This work was funded by LPP Combustion, LLC.

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