THE APPLICATION OF FUEL STAGING ON REDUCING NO$_X$ EMISSIONS FROM GAS BURNER

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ABSTRACT

This paper present the method of burning gaseous fuel at two different places, i.e. in the swirler vane passages and at the exit plane of the first combustor for a two stage combustion system. The present work only used natural gas. The effect of injecting the secondary fuel has a vast influence on reducing emissions of oxides of nitrogen. The reduction was found to be greater than 50% even at near stoichiometric conditions. The secondary fuel acted as a reducing element for the thoroughly formed oxides of nitrogen from the first stage combustion.

1.0 INTRODUCTION

The effects of increased levels of NO$_x$ in the atmosphere are wide-reaching. In the atmosphere NO is rapidly oxidised to NO$_2$ and this form plays an essential role in the formation of tropospheric ozone and photochemical smog, and is oxidised to form nitric acid that may then be deposited as acid rain (Harrison, 1990). At ground

level, increased concentrations (above 0.06 ppm) of NO_2 can cause respiratory problem (World Health Organisation, 1987).

The legislation of NO_x emission limits in many parts of the world has substantially complicated the process of burner design. Attempts at lowering NO_x emissions by reducing the flame temperature will lead to reduce flame stability or increased CO emissions. The lowest NO_x emission obtained in a given configuration is always limited by unacceptable stability problems or CO emissions. Thus the burner design has become a trial-and-error, multi-parameter optimisation process (Van Der Meij et al., 1994).

Basically there are two techniques of controlling NO_x: those which prevent the formation of nitric oxide (NO) and those which destroy NO from the products of combustion. In the present work the latter method is employed: lean combustion for low thermal NO_x followed by secondary fuel injection for combustion reaction in the second stage zone, which could destroy first stage NO_x through a reburn mechanism.

The methods that prevent the formation of NO involved modifications of the conventional burner designs or operating conditions, such as lean primary zone, rich primary zone, rich/lean, or reduced residence time, since the main factors governing formation of NO is temperature and oxygen availability. However, the rich/lean method tends to increase CO and UHC. Advanced combustor designs are needed for reducing all four major pollutants simultaneously over a range of thermal or engine power outputs. This gives rise to the use of variable geometry combustor and staged combustion to cope with the demands of burner turndown and power variations in gas turbines, when the overall A/F is increased as power is reduced. For ultra low NO_x emissions, lean premixed-prevaporised combustors and catalytic combustors are being developed.

In staged combustion, the combustion process is arranged to occur in a number of discrete stages. In theory, either circumferential, radial or axial

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staging may be employed. However, in practice circumferential fuel staging increases NO_x - instead of the fuel being distributed uniformly around the liner, it is injected at a small number of points, where it produces regions of high temperature (Lefebvre, 1975). The elaboration's for the above mentioned three types of fuel staging are as follows:

a) *Circumferential.* Usually this entails disconnecting alternately located nozzles from the fuel supply. It is ideally suited to tuboannular systems but on annular chambers its advantages are largely offset by the quenching effects of the surrounding cold air on the localised burning zones.

b) *Radial.* The simplest application of this technique is to double-banked annular combustors where, at low fuel flows, it is a relatively simple matter to inject all the fuel into the inner or outer combustion zone.

c) *Axial.* By designing the primary zone for optimum performance at low power settings, and then injecting the extra fuel needed at higher power levels at one or more locations downstream.

In the present work, axial fuel staging was employed that consisted of lean-lean staged combustion. The first combustor was operated very lean with all the air needed for combustion introduced in this zone and the operation was set close to the lean stability limit. Fuel, without any air, was then injected into the completely burnt products of this lean primary combustion zone to bring the burner to the desired overall excess air. Typically the lean zone may have an equivalent ratio of 0.6 with fuel injected in the secondary zone to bring the overall equivalent ratio to 0.9. Thus, it is a lean/lean staged system.

2.0 EXPERIMENTAL SET-UP

This system comprised of two different sizes combustors. The smaller one of 76 mm inside diameter was attached to the plenum chamber and acted as the first stage. The radial swirler of 40mm outlet diameter and 21.5 mm depth was used as a flame stabiliser. The first combustor was fuelled via the radial vane passage injection mode. The air and fuel were mixed thoroughly prior to ignition. At the exit plane of the first combustor a wall fuel injector of 76 mm diameter with eight fuel nozzles of 3.5 mm diameter was attached. This is the injector for the second stage secondary fuel. The mixtures of flue gas from the first combustor and the second fuel were allowed to expand freely into a larger combustor of 140 mm internal diameter. The wall injector and the second combustor were attached to the first combustor by the used of flanges. The schematic diagram of set-up of fuel staging test rig is shown in Fig. 1.
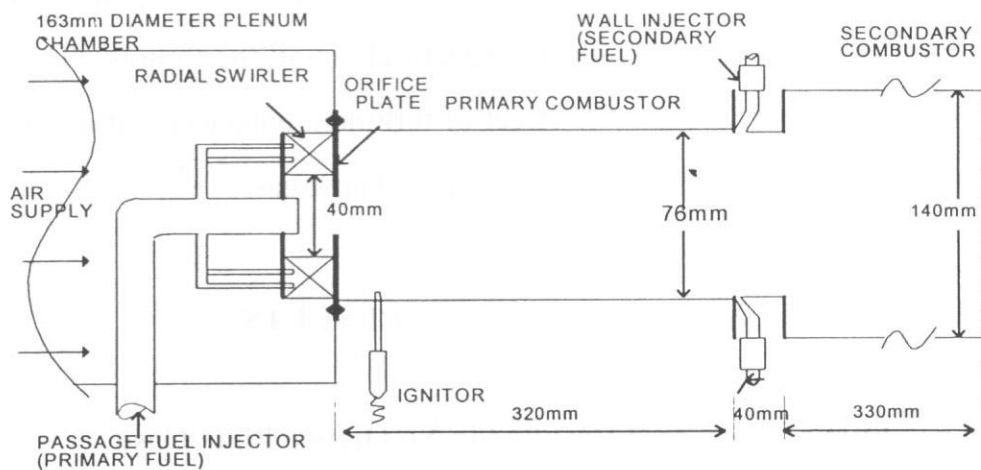

Fig. 1   Schematic Diagram of Fuel Staging Test Set-up Rig

The staged combustion tests were run at 40 mmWG pressure loss to simulate burner operating conditions. The radial swirler in the primary zone in Fig. 1 has a swirler outlet of 40 mm. This could be fitted with different orifice plates to reduce the swirler outlet areas. The role of the orifice plate was to enhance flame stabilisation and provide a better mixing of the air and fuel prior to ignition. It also created the pressure loss at the outlet rather than in the vane passage which generated maximum turbulence in the swirl shear layer. The orifice plate also helped to prevent fuel from entraining into the corner dumped expansion outer recirculation zone and thus create a rich local zone which would lead to higher NO_x emissions from this area. The orifice plate was mounted at the exit plane of the radial swirler as shown in Fig. 1.

All the tests were carried out at atmospheric pressure with an air inlet temperature of 400 K, with burner upstream pressure of 40 mmWG. The orifice plate sizes involved were 25.4 mm and without any orifice plate (40mm); designated as 25 mm and without (40 mm), respectively in all the figures. The 25.4 mm orifice plate was chosen to show the effect of turbulent enhancement due to area restriction which helps in reducing emissions from exhaust gas.

3.0   MEAN EXHAUST EMISSION RESULTS

3.1   Fuel Staging Test Results for No Orifice Plate Case

Figures 2-7 show the fuel staging test results as compared to the baseline results with all the fuel supplied to the primary swirler for no swirler outlet orifice plate. As can be seen generally, a marked reduction in NO_x emissions can be achieved. A reduction of greater than 50% was achieved even at near stoichiometric condition where the equivalent ratio was 0.913.

Figure 2 shows an operating equivalent ratio for fuel staged case of 0.655-0.913. The rich condition was limited by a drastic increase in carbon monoxide

(CO) emission to an unacceptable value. At an equivalent ratio of 0.913, a corrected NO_x emission to 0% oxygen on a dry basis of 29.3 ppm were formed representing a 50% reduction. Figure 3 shows that a combustion inefficiency of 0.01% was obtained for a wide range of equivalent ratios up to 0.815 when the combustion inefficiency started to rise. This means that a combustion efficiency of 99.99% was achieved for a wide range of equivalent ratios.

Figure 4 shows CO emissions of less than 10ppm were obtained over a wide range of equivalent ratios up to 0.745. Even at an equivalent ratio of 0.8 the CO emission were only 15 ppm. The lowest CO emission was 6ppm at an equivalent ratio of 0.65. However, fuel staging increases the CO to 100ppm for equivalent ratio of 0.9 but were acceptable at 15 ppm for an equivalent ratio of 0.78. This CO increases due to the lower residence time with fuel staging and the lower oxygen availability. Unburned hydrocarbon (UHC) emissions of less than 5 ppm were achieved for the entire range of operating equivalent ratios. Figure 6 shows the NO_x results as a function of the adiabatic flame temperature, which shows that even at high flame temperatures the increase in NO_x was not very steep for fuel staging. This was due to higher temperature which increased the rate of formation of NO, when single stage combustion occurred. Patry and Engel (1950) studied the reaction between nitric oxide and methane at temperatures up to 1100°C and found that 70 percent of the nitric oxide was converted in 3 seconds at this temperature, to form hydrogen cyanide, water and hydrogen via the following reaction:

$$CH_4 + NO \rightarrow HCN + H_2O + 1/2H_2 \qquad (1)$$

Thus, HCN can be an undesirable emission generated at low temperature in the exhaust.

3.2   Fuel Staging Test Results for 25mm Orifice Plate Case

Figures 8-13 show the fuel staging test results as compared to the baseline results for 25 mm swirler outlet orifice plate. It can be seen that thee marked reduction in NO_x emissions can be achieved. The reduction of corrected NO_x is even greater than the case of no orifice plate condition (Fig. 8). A reduction of greater than 70% was achieved at an equivalent ratio of 0.875 (near stoichiometric condition).

Figure 8 shows an operating equivalent ratio for fuel staging case of 0.65-0.88. The rich condition was once again limited by a drastic increase in carbon monoxide (CO) emission to an unacceptable value. At an equivalent ratio of 0.875, a corrected NO_x emission to 0% oxygen on a dry basis of 15ppm were formed representing a 72% reduction. Figure 9 shows that a combustion inefficiency of 0.01% was obtained for a wide range of equivalent ratios up to 0.83 when the combustion inefficiency started to rise. This means that a combustion efficiency of 99.99% was achieved for wide range of equivalent ratios.
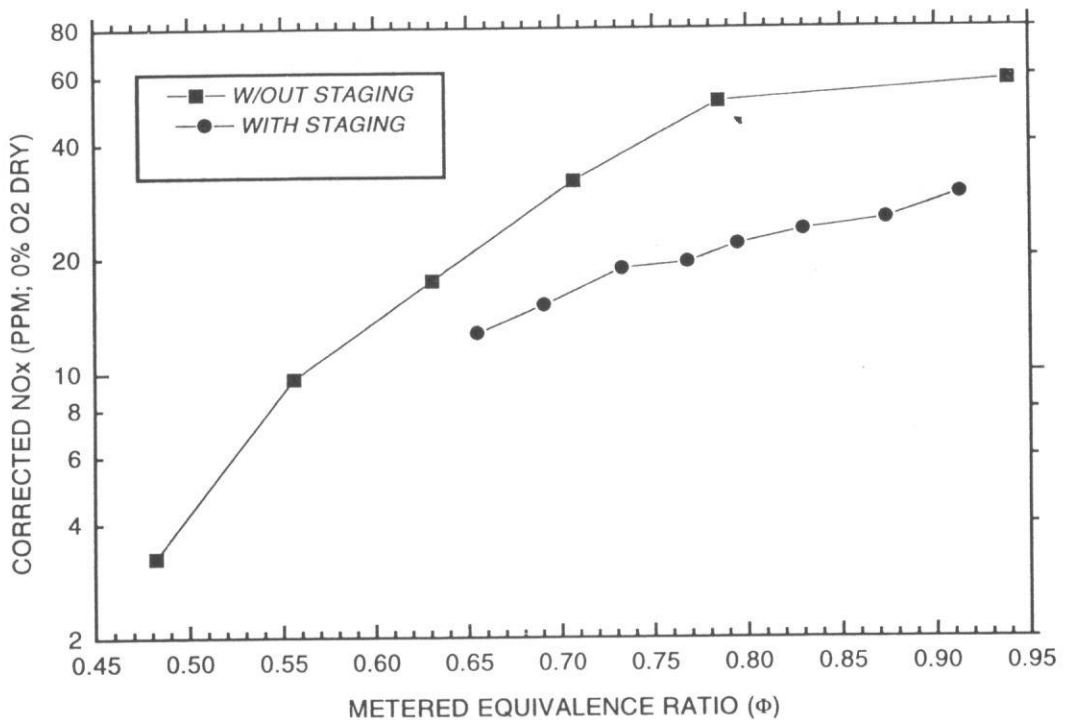
Fig. 2   Corrected NO_x vs Metered Equivalence Ratio for

No Orifice Plate @ $\Delta P = 40$ mmH$_2$O; $T_{in} = 400$ K; $\Phi_{primary} = 0.625$
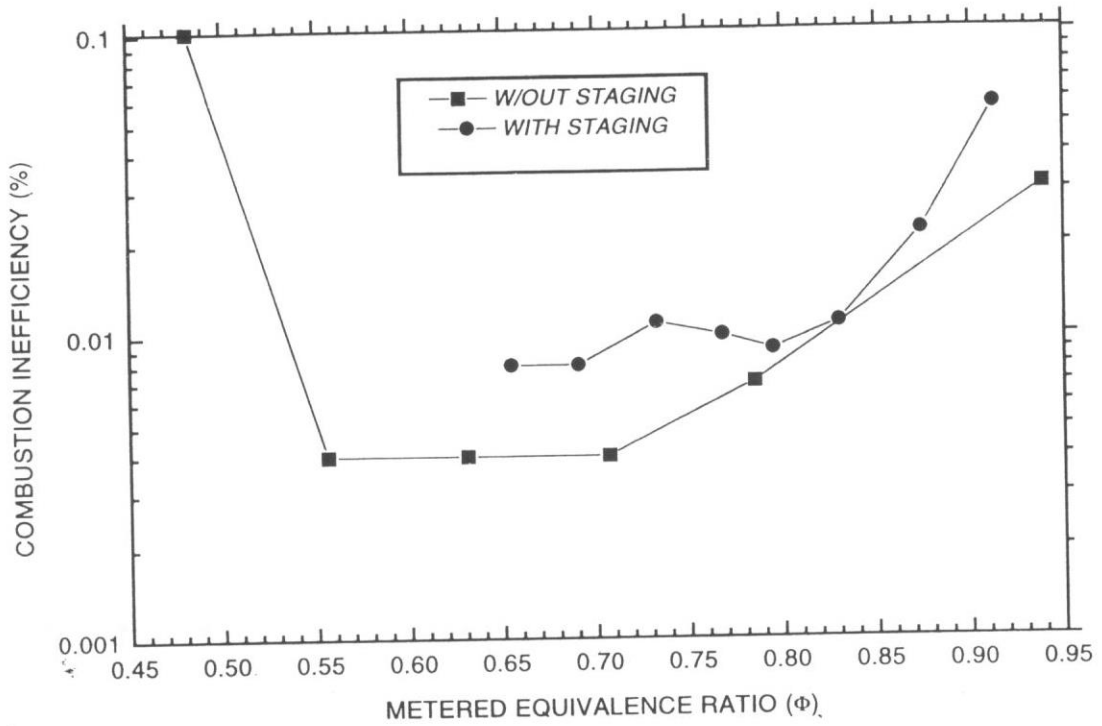
Fig. 3 Combustion Inefficiency vs Metered Equivalence Ratio for
No Orifice Plate @ $\Delta P = 40$ mmH$_2$O; T$_{in}$ = 400 K; $\Phi_{primary}$ = 0.625

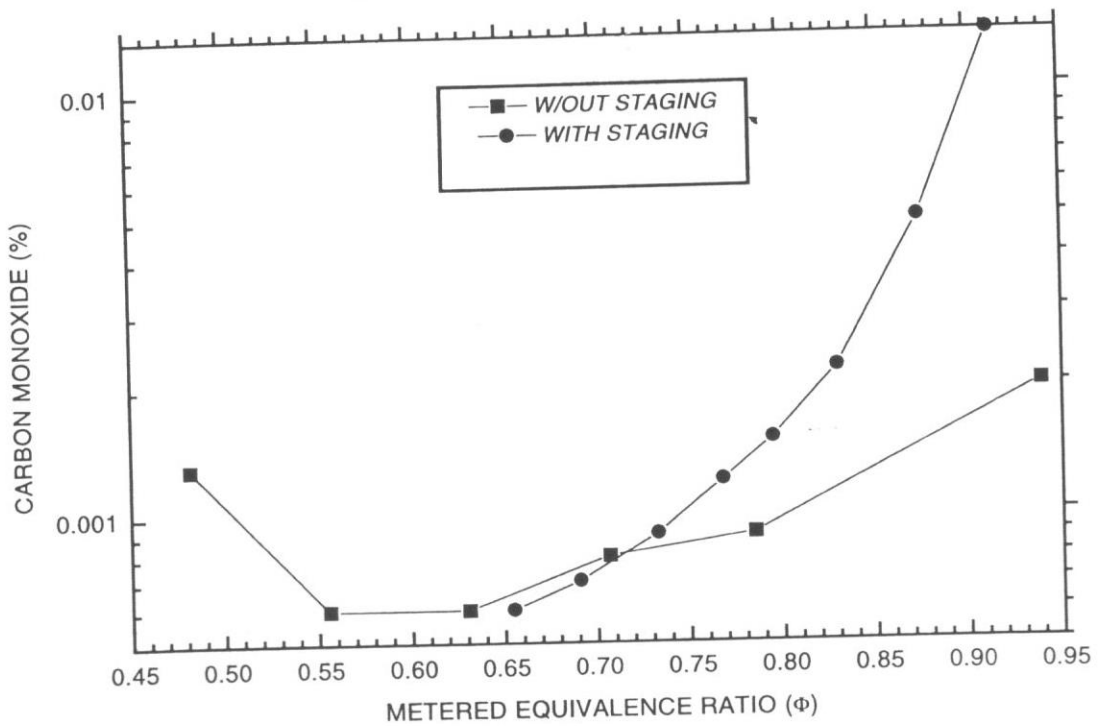
Fig. 4 Carbon Monoxide vs Metered Equivalence Ratio for
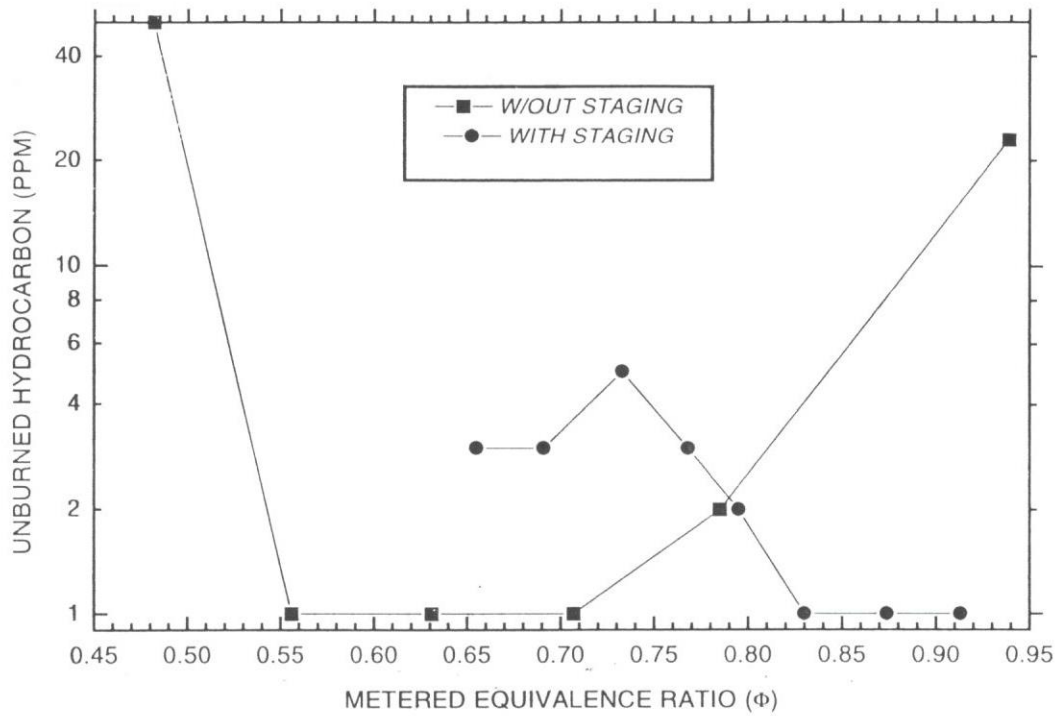No Orifice Plate @ $\Delta P = 40$ mmH$_2$O; T$_{in}$ = 400 K; $\Phi_{primary}$ = 0.625

Fig. 5   Unburned Hydrocarbon vs Metered Equivalence Ratio for
No Orifice Plate @ ΔP = 40 mmH$_2$O; T$_{in}$ = 400 K; $\Phi_{primary}$ = 0.625
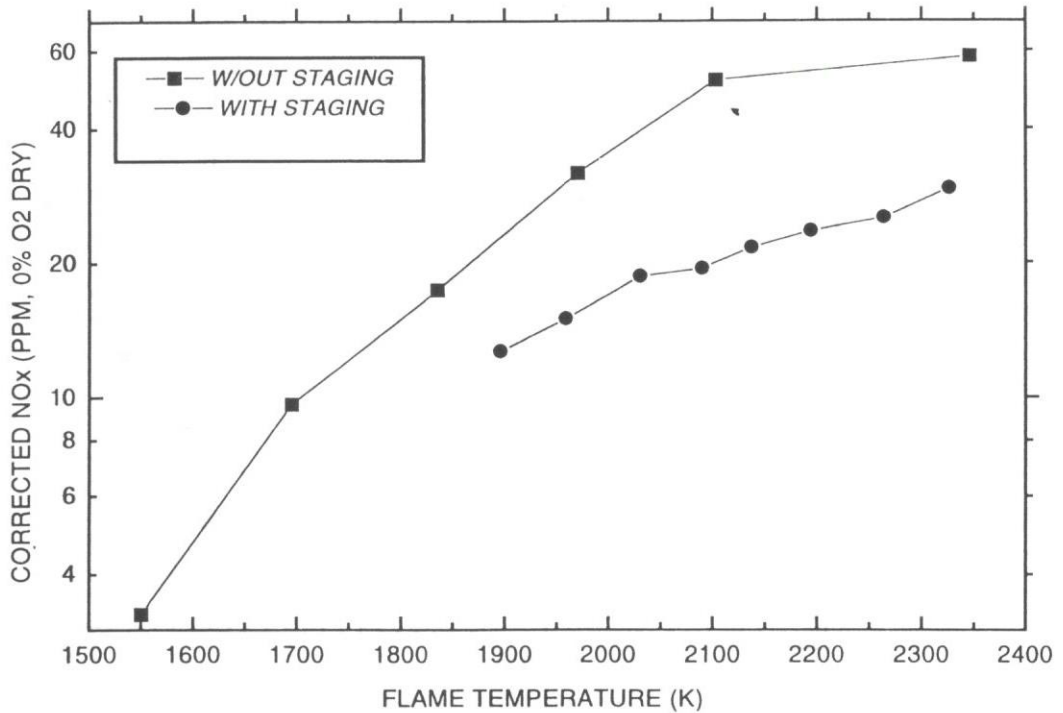
Fig. 6   Corrected NO$_x$ vs Flame Temperature Ratio for
No Orifice Plate @ ΔP = 40 mmH$_2$O; T$_{in}$ = 400 K; $\Phi_{primary}$ = 0.625
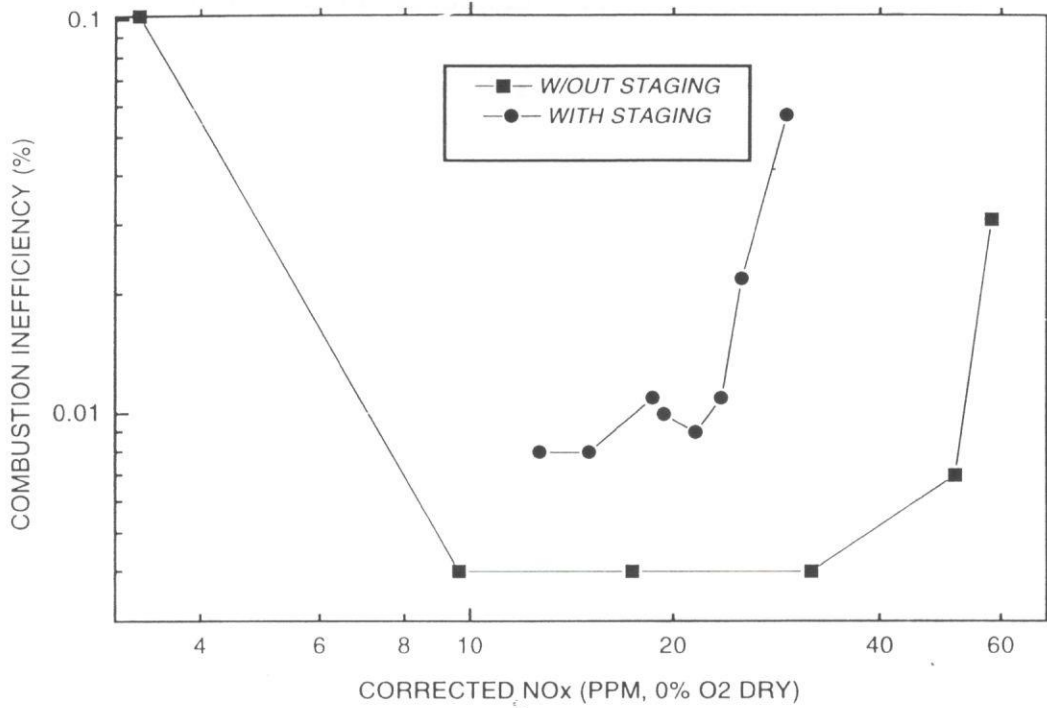
Fig. 7 Combustion Inefficiency vs Corrected NO_x for

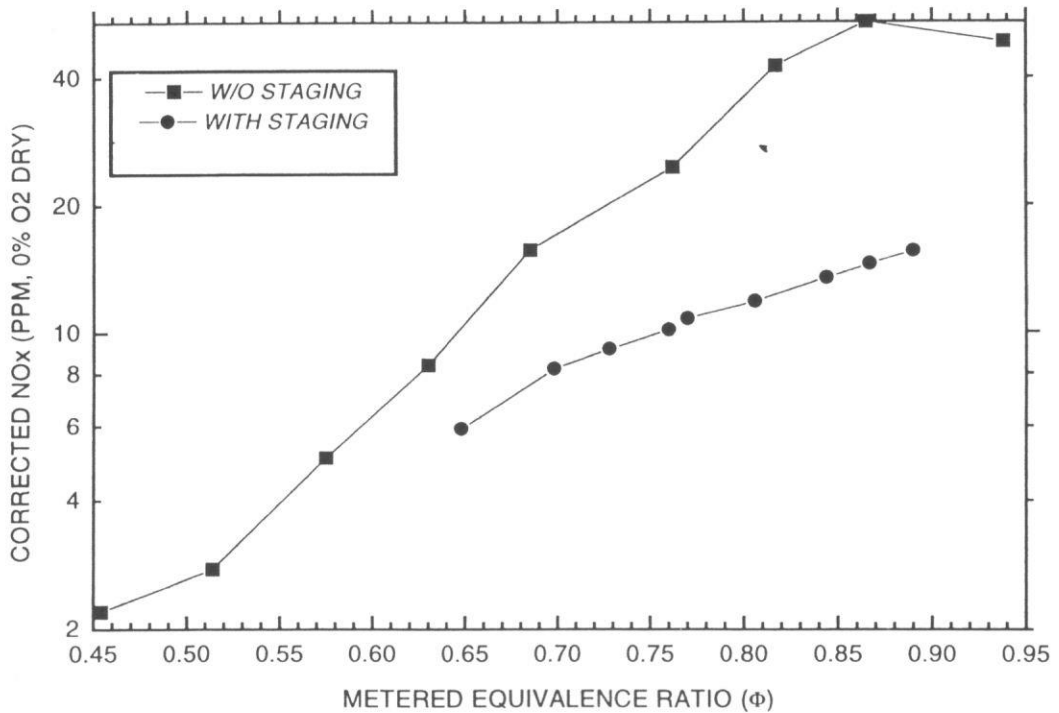No Orifice Plate @ $\Delta P = 40$ mmH$_2$O; $T_{in} = 400$ K; $\Phi_{primary} = 0.625$

Fig. 8 Corrected NO_x vs Metered Equivalence Ratio for

25 mm Orifice Plate @ $\Delta P = 40$ mmH$_2$O; $T_{in} = 400$ K; $\Phi_{primary} = 0.582$

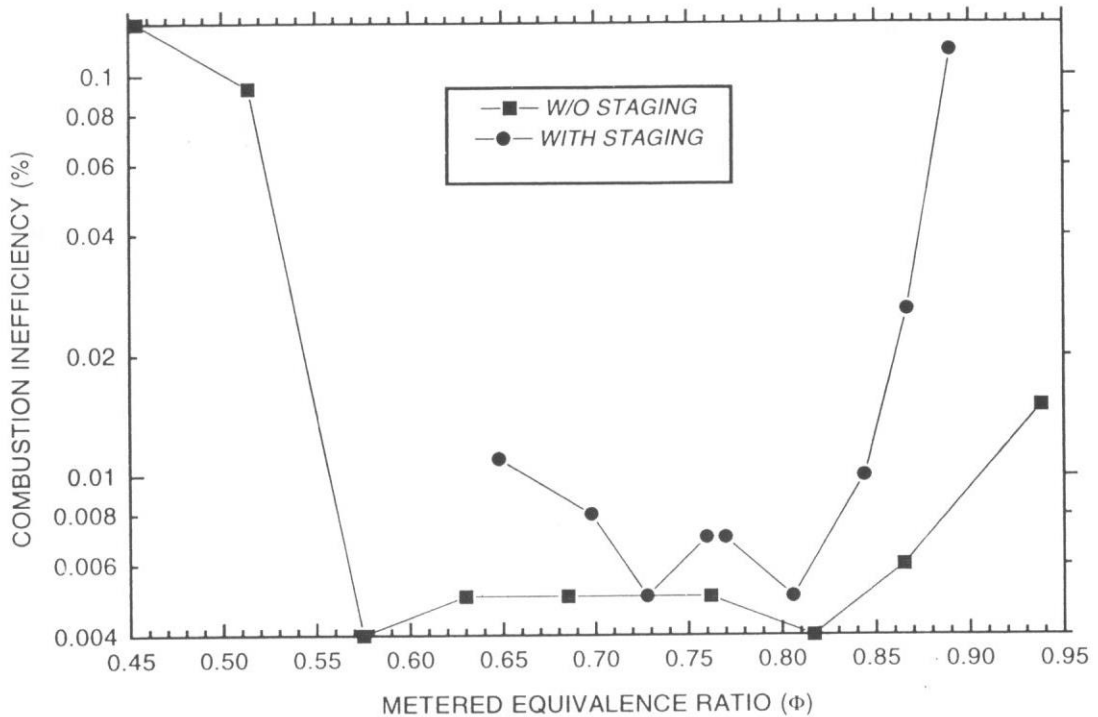Fig. 9 Combustion Inefficiency vs Metered Equivalence Ratio for
25 mm Orifice Plate @ ΔP = 40 mmH$_2$O; T$_{in}$ = 400 K; $\Phi_{primary}$ = 0.582

Figure 10 shows CO emissions of less than 10ppm were obtained over a wide range of equivalence ratios up to 0.81. This is also a better result than the no orifice plate case. Even at an equivalent ratio of 0.84 the CO emission were only 20ppm. The lowest CO emission was 7ppm at an equivalent ratio of 0.65. However, fuel staging increases the CO to more than 100 ppm for equivalent ratio greater than 0.86. This CO increases was due to the lower residence time with fuel staging and the lower oxygen availability. Unburned hydrocarbon (UHC) emissions of less than 5 ppm were achieved for most of the range of operating equivalent ratios as shown in Fig. 11. Figure 12 shows the NO$_x$ results as a function of the adiabatic flame temperature, which shows that even at high flame temperatures the increase in NO$_x$ was not very steep for fuel staging. Figure 13 shows that lowest NO$_x$ emission of 6ppm was obtained at an combustion inefficiency of 0.01%.

4.0   CONCLUSION

A NO_x reduction of at least 50% could be achieved for both orifice plate sizes using fuel staging method with a very lean primary zone. The lower NO_x at an equivalent ratio of 0.78 was obtained for the 25mm orifice plate. NO_x with staged fuel injection had a much lower dependence on flame temperature than a single stage combustion system. Single digit NO_x corrected to 0% O dry could be obtained at equivalent ratio of 0.78 (30% excess air) for the 25mm orifice plate case with the lowest value of 8.75 ppm obtained with the 25 mm orifice plate.
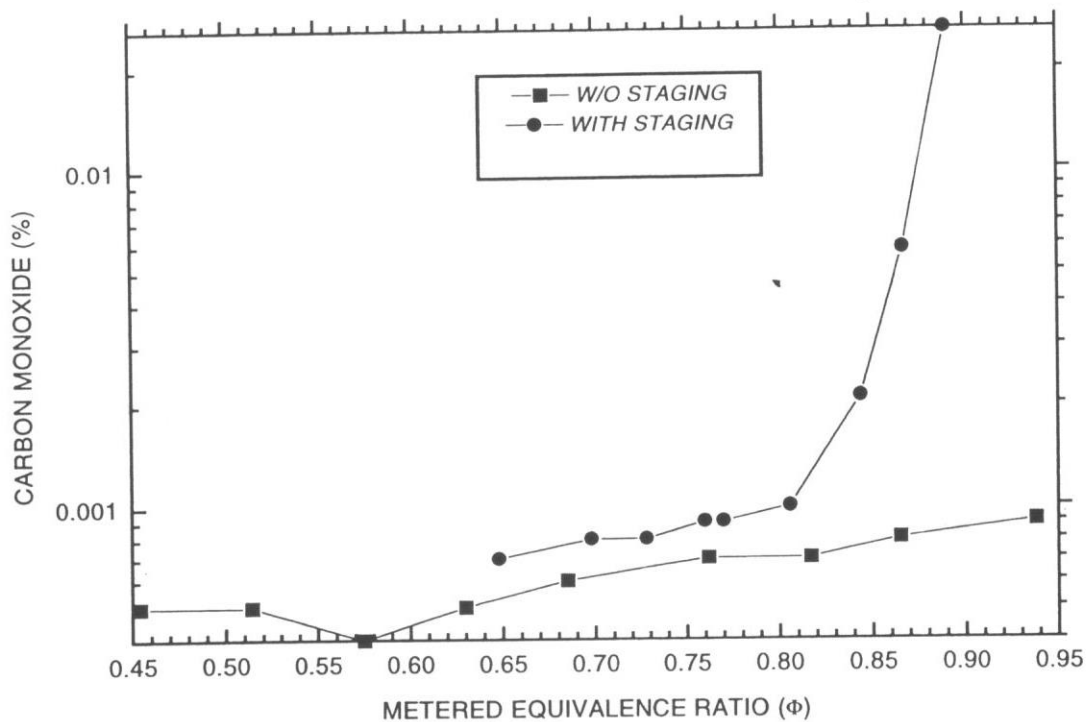

Fig. 10   Carbon Monoxide vs Metered Equivalence Ratio for
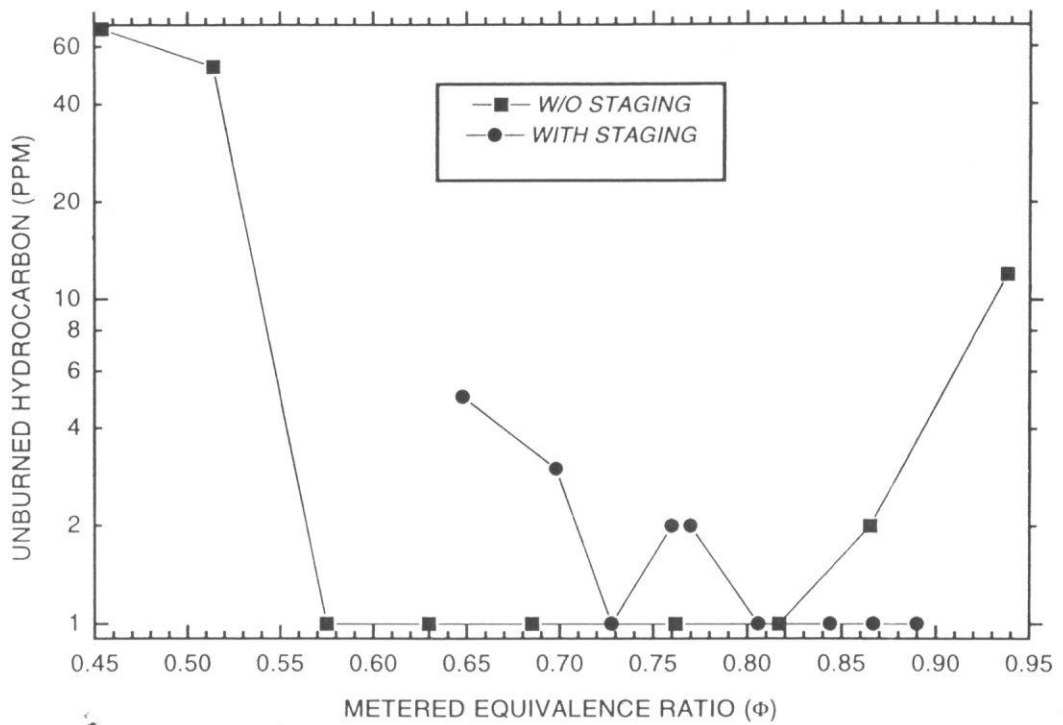25 mm Orifice Plate @ $\Delta P = 40$ mmH$_2$O; $T_{in} = 400$ K; $\Phi_{primary} = 0.582$

Fig. 11 Unburned Hydrocarbon vs Metered Equivalence Ratio for
25 mm Orifice Plate @ $\Delta P = 40$ mmH$_2$O; T$_{in}$ = 400 K; $\Phi_{primary}$ = 0.582
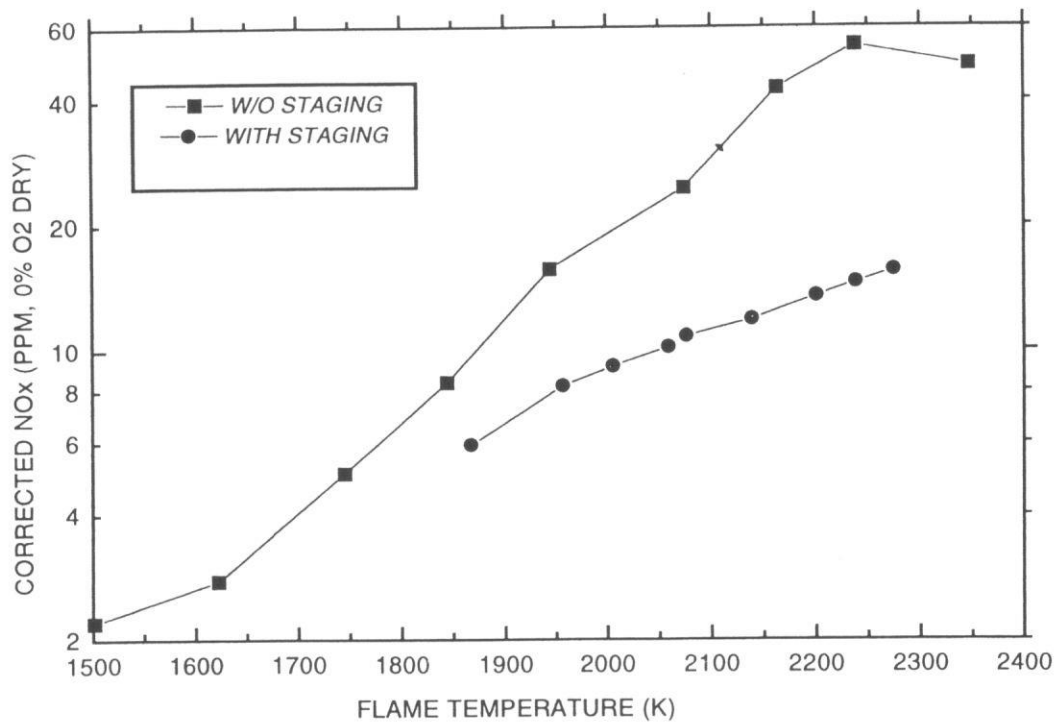

Fig. 12 Corrected NO$_x$ vs Flame temperature for
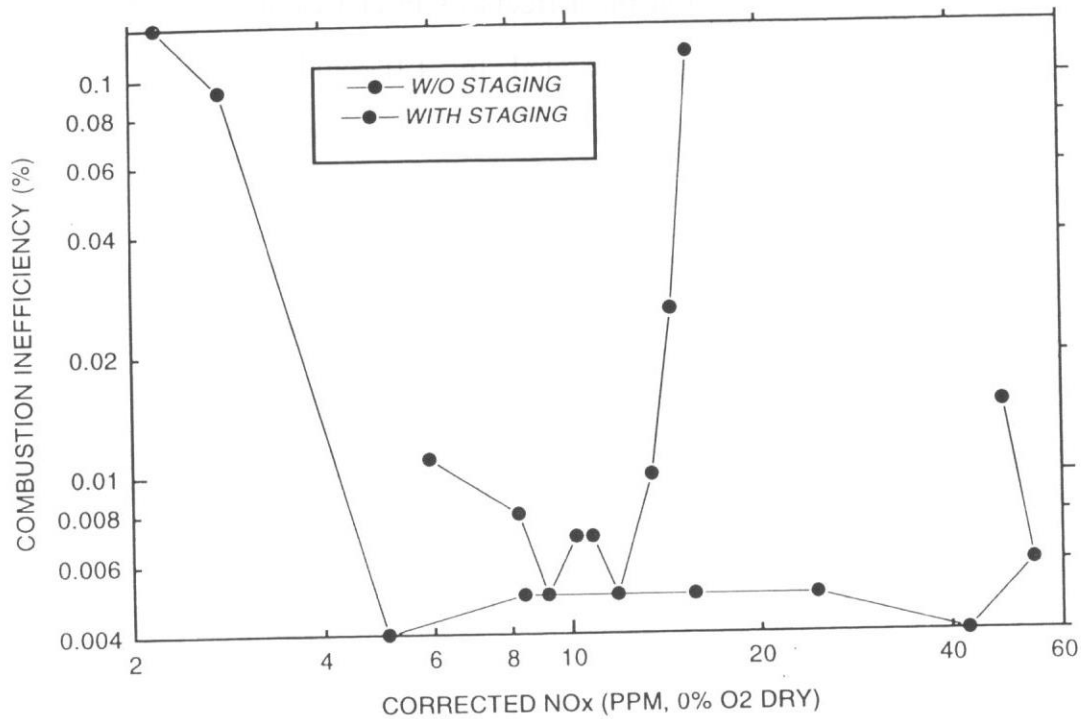25 mm Orifice Plate @ $\Delta P = 40$ mmH$_2$O; T$_{in}$ = 400 K; $\Phi_{primary}$ = 0.582

Fig. 13 Combustion Inefficiency vs Corrected NO_x for

25 mm Orifice Plate @ $\Delta P = 40$ mmH$_2$O; $T_{in} = 400$ K; $\Phi_{primary} = 0.582$

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