NO_X EMISSION REDUCTION FROM GAS BURNER SYSTEM APPLYING WATER COOLING TECHNIQUE

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ABSTRACT

A gas burner system applying water cooling heat exchanger has been investigated using a 140 mm inside diameter combustor of 294 mm length. The combustor was placed vertically upwards. All tests were conducted using natural gas only. A fixed straight blade radial swirler with 76 mm outlet diameter was placed at the inlet plane of the combustor. An orifice plate of 59 mm was inserted at the exit plane of the swirler to enhance turbulence and help in mixing of the fuel and air. Fuel was injected at the back plate of the swirler using central fuel injector with eight fuel holes pointed radially outward. Tests were conducted at 2 mmW.G.

pressure loss. A reduction of about 20 per cent on NO_X emissions was achieved at equivalence ratio of near stoichiometric (0.9). Other emissions such as carbon monoxide were well under 100 ppm except below equivalence ratio of near 0.5 due to cooling effect. Unburned hydrocarbon emissions were well below 10 ppm except below equivalence ratio of 0.5.

1.0 INTRODUCTION

The need to protect the environment from combustion generated NO_x has led to considerable demand to improve burner design. Previously, attention has been given to gas turbine used in aero engines and for power generation in stationary plant. Intense research has been going on to develop gas turbine with very low NO_x emissions. In recent years considerable attention has been paid to reducing NO_x from process burners. However, in order to minimize NO_x emissions from gas turbine combustor it is necessary to increase the primary zone air flow and to improve air and fuel mixing. Many low-emission gas turbine combustor designs aim to pass as much air as possible through the combustor head and add the remaining air further downstream in a dilution zone with no air needed for film cooling [1]. The primary zone is a burner configuration [1]. Thus, many of the methods used to effectively minimize NO_x emissions from gas turbine can be adapted to burner configurations and conditions.

A major use of combustion is in heating appliances, both industrial and domestic. The operation of such devices may be classified in terms of the burner used, which in turns depends on the physical state of the fuel employed [2]:

a) Gas burners: gaseous fuels are burnt in pre-mixed or diffusion flames.
b) Liquid fuel burners: oil, the principal liquid fuel, is atomised and the droplets burnt in an atmosphere of air.

c) Dust burners: pulverised coal is dispersed in a gas stream and burnt in a similar way to liquid fuels although, since the mechanism of combustion differs, the design of the combustion chamber is not identical.

d) Fluidised-bed combustors: air is passed upwards through a bed of sand and coal particles that becomes fluidised. When the coal is ignited, a self-sustaining reaction takes place..

e) Fuel beds: larger particles of solids fuels are also burnt simply by placing them on a grid through which air passes from below.

Whole house heating usually utilises an appliance with a gas rate of the order of 11.7 to 22.0 kilowatt (40,000 Btu/h to 75,000 Btu/h) [3].

However, in the present work, the interest is put on the gas burners for domestic central heating system using natural gas. Therefore, the other burners using liquid fuels and solid fuels will not be discussed here.

Basically there are two types of gas burner: the diffusion burners and the premixed burners. All domestic and most industrial burners are simple variants of the laminar flame Bunsen burner and provide an aerated flame. The principal behind it is that the gas enters a narrow tube through a jet and draws into it sufficient primary air to render the flame non-luminous. The remaining secondary air that is required to complete the combustion will be admitted downstream. This is basically the principle of an air staged diffusion burner. A pure fuel jet diffusion flame from hydrocarbon fuel has a yellow colour as a result of radiation from carbon particles that are formed within the flame [4]. In premixed burner, the air and fuel are mixed before they are passed through a jet into the combustion zone and sufficient initial air is mixed to prevent soot formation. The velocity of the mixture through the burner jet is important: if the velocity is too low (below the burning velocity of the mixture) the flame can light back into the mixing region, whereas if the velocity is too high the flame can 'liftoff' from the burner to the extent where it can be extinguished by entrainment of additional air around the burner.

The use of natural gas for domestic and commercial central heating system is widespread. In the United Kingdom, domestic heating is dominated by gas, in contrast to European countries such as Germany and Switzerland where oil is the principal fuel [5]. This is reflected in Table 1 [6] where it can be seen that natural gas is the dominant fuel used domestically in the United Kingdom in 1980. From this energy delivered and used by the domestic sector, it is clearly shown that most of these fuels are used for space heating as envisaged in Table 2 [6].

Emissions of carbon monoxide, unburned hydrocarbon and also sulphur dioxide are very low from natural gas burner if sufficient excess air is used, the most difficult pollutant to control is oxides of nitrogen. Oxides of nitrogen emissions levels are usually of the order of 100 ppm per combustion unit and their combined effect results in a significant annual ground level concentration [7]. The continued trend in emissions regulations will require lower oxides of nitrogen emission levels than at present.

Table 1 Household Use of Energy by Fuel in 1980 (GJ/household year) [6].

	Electricity	Oil	Coal & Solid Fuel	Gas	Total
Delivered Energy	15.4	5.7	17.4	44.2	82.7
Useful Energy	13.9	4.1	7.3	28.5	53.8
Overall Efficiency	90%	71%	42%	64%	65%

A laminar flame bar burner or bladed burner used for domestic water-heating unit operating at a maximum thermal input of 11 kW was demonstrated to have NO_x emissions of 29.5 ppm by Dupont [8]. This burner is typical of the current burner designs used as water-heating boilers. The blades are 116 mm long, 50 mm deep and penetrate the burner body by 25 mm in height. This particular burner is similar to the Bray burner product group 59.

Table 2 Household Use of Energy by Purpose of End Use in 1980 (GJ/household year) [6]

	Appliances & Lighting	Cooking	Water Heating	Space Heating	Total
Delivered Energy	4.6	6.0	14.5	57.6	82.7
Useful Energy	(4.6)*	3.7	7.5	38.0	53.8
Efficiency	(100%)*	62%	52%	66%	65%

*Useful energy requirements of appliances are not well-defined because the uses are so varied, so the use of a nominal 100% efficiency here does not imply that there is no longer scope for reducing the amount of electricity needed to perform a particular task (e.g. by insulating fridges better and by using more micro-electronics in television).

Another Bray burner was studied by Foster et al. [9] that is used in domestic water-heating unit also. This larger burner could operate at a maximum heat input of 27 kW before the onset of resonance, but was typically used with a heat input of 10-25 kW. NO_x emissions of between 20-120 ppm expressed on a dry O_2-free basis for 9.87-27.80 kW, respectively, were recorded without flue gas recirculation. Even with 13.5% flue gas recirculation, the corrected NO_x emissions were well over 50 ppm for the highest thermal input.

2.0 NO_X EMISSIONS REDUCTION TECHNIQUES

The effects of increased levels of NO_x (oxides of nitrogen) in the atmosphere are wide-reaching. In the atmosphere nitric oxide (NO) is rapidly oxidised to nitrogen dioxide (NO_2) and in this form plays an essential role in the formation of tropospheric ozone and photochemical smog. NO_2 is further oxidised to form nitric acid that may then be deposited as acid rain [10]. At ground level, increased concentrations (above 0.06 ppm) of NO_2 can cause respiratory problem [11].

NO_x emission limits legislation in many parts of the world has substantially complicated the burner design process. Attempts at lowering NO_x

emissions by reducing the flame temperature will lead to reduced flame stability or increased carbon monoxide (CO) emissions. The lowest NO_x emission obtained in a given configuration is always limited by unacceptable stability problems or CO emissions. Therefore, the process of burner design has become a trial-and-error, multi-parameter optimisation process [12].

Generally 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.

The former approach has been widely applied. The main aim is to reduce the flame temperature thus reducing the formation of NO_x. There are several ways of accomplishing this.

One method is the modification of combustion processes, i.e. either to burn fuel-rich or fuel-lean. The operation of lean burning is to introduce additional air in order to reduce the flame temperature. This would generally cause a significant decrease in the production of NO_x. A reduction of the primary-zone flame temperature, on the other hand may increase the emissions of carbon monoxide and unburned hydrocarbon (UHC). The problem with fuel-rich combustion is the formation of soot and CO, even though the stability margin is widen.

Another method, i.e. steam or water injection has been shown to be very effective in accomplishing the above goal [13]. A typical NO_x reduction curve as a function of rate of water injection is shown in Figure 1 [14]. These data were obtained in an aeroderivative inductive gas turbine at full power. The same effect was also demonstrated by Fox and Schlein [15] testing the FT8 gas turbine combustor in their final test run. The FT8 engine is an industrial/marine gas turbine engine which is a derivative of the widely used JT8D aircraft jet engine. However, to avoid detrimental effects on turbine durability the water has to be purified to a maximum of 2-5 ppm of dissolved solids [14 & 16]. Furthermore, there are other complications such as incorporating the water injection system to the combustor design. Another disadvantage of water injection is the undesirable side effects of quenching CO burnout. These drawbacks caused water injection method to be unattractive for smaller gas turbines or where availability of sizeable water supply is difficult. However, it is a feasible technique for burner NO_x control in water heater or steam generator.

Other methods of NO_x control involve staged combustion, variable geometry combustion, lean premixed prevaporised combustion and catalytic combustion.

In the present work, a water cooling heat exchanger comprised of a 6 mm stainless steel tube coil, was used to reduce emissions by supplying constant amount of cooling water through it. The heat exchanger was placed at a particular distance from the flame.

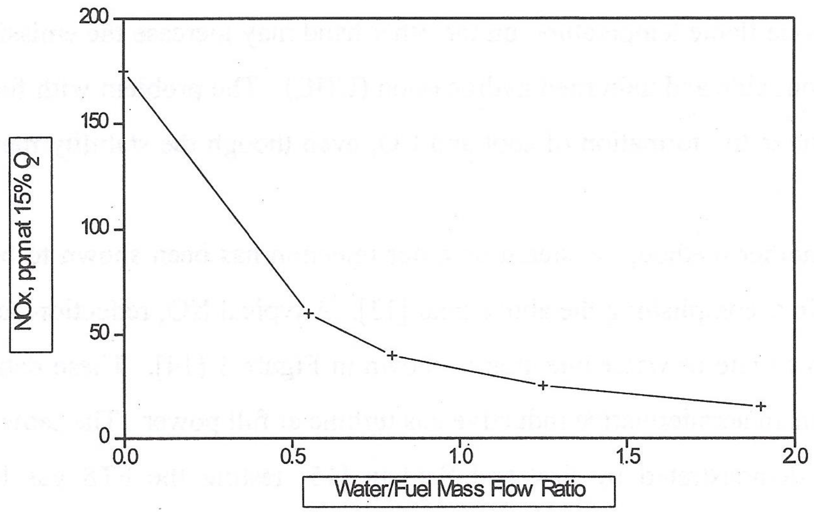


Figure 1 NO_x as a function of water addition in a gas-turbine combustor running on natural gas at a pressure ratio of 30 [14].

3.0 EXPERIMENTAL SET-UP

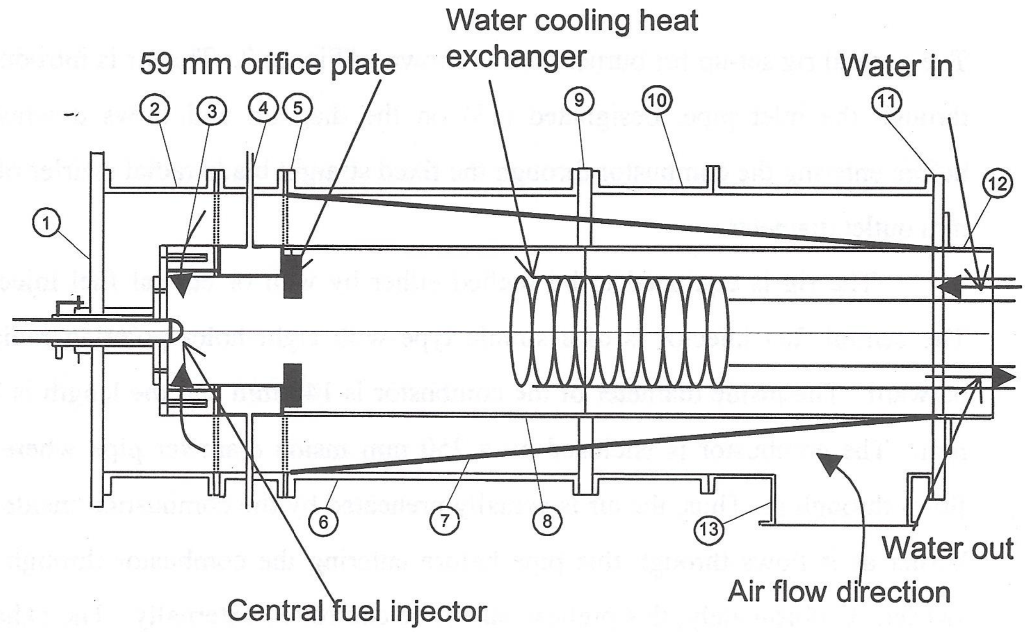
The general rig set-up for burner tests is shown in Figure 2. The air is introduced through the inlet pipe, designated (13) on the diagram and flows downward before entering the combustor through the fixed straight blade radial swirler of 76 mm outlet diameter.

The rig is equipped to be fuelled either by wall or central fuel injector. The central fuel injector is of a simple type with eight holes pointing radially outward. The inside diameter of the combustor is 140 mm and the length is 294 mm. The combustor is enclosed by a 250 mm inside diameter pipe where air flows through it. Thus, the air is actually preheated by the combustion inside the burner as it flows through this pipe before entering the combustor through the swirler. Unfortunately, this preheat cannot be controlled externally. The exhaust sampling probe is mounted at the top of the end pipe to measure all the major exhaust gases such as oxides of nitrogen (NO and NO_2), carbon monoxide and unburned hydrocarbon.

4.0 TEST CONDITIONS

Tests were carried out at around 400 K inlet temperature simulating domestic central heating unit. However, to maintain the inlet temperature at this value is almost impossible since the air preheat temperature cannot be controlled externally. At some point in these tests, the air inlet temperature exceeded 400 K. Natural gas was used as fuel throughout the entire investigation.

Domestic central heating boilers operate with a fan air supply at below 0.5 percent (5 mb, 50 mmH_2O) pressure loss compared to 2-5 percent pressure loss for gas turbine combustion. These pressure losses of 2-5 percent when converted to burner pressure loss are equivalent to 200-500 mmH_2O.



1. End flange for fuel supply & swirler support.
2. Swirler case pipe.
3. Variable swirler.
4. Wall fuel injector.
5. Flange.
6. 250 mm dia. case pipe.
7. Impingement tube.
8. 140 mm dia. case pipe.
9. Joint flange.
10. Reverse air passage pipe.
11. Recess flange.
12. End pipe.
13. Reverse air inlet pipe inlet. Inlet P & T tappings

Figure 2 Burner Test Rig Using Cooling Coil Heat Exchanger

Tests were conducted using an orifice plate of 59 mm diameter that was inserted at the exit plane of the outlet of the wall injector section (refer to Fig. 2). This is to enhance flame stabilisation and to provide better mixing of air and fuel prior to ignition. The orifice plate also helps to prevent fuel from entraining into the corner recirculation zone that will create local rich zone hence resulting in higher local NO_x emissions which in the end contributes to high total NO_x emissions.

5.0 RESULTS AND DISCUSSIONS

Figures 3-6 show the effect of applying water cooling heat exchanger on reducing mean exhaust emissions from gas burner system using straight blade radial swirler.

Figure 3 shows the reduction in NO_x emissions when using water cooling heat exchanger. A reduction of 20 per cent was achieved at equivalence ratio of 0.9, while a reduction of about 13.2 per cent was achieved at equivalence ratio of 0.45 when compared to the tests where heat exchanger were not applied. These demonstrate that water cooling method is quite effective in reducing NO_x emissions in gas burner system especially near stoichiometric condition or fuel rich condition.

Figure 4 shows the combustion inefficiency plotted against equivalence ratio for both test conditions. It could be seen that the combustion inefficiency was increased when using water cooling. Combustion efficiency reduces drastically near lean condition to higher than 99 per cent from about 99.97 per cent at equivalence ratio of near 0.9. This could be due to flame quenching when using water cooling where the emission of carbon monoxide increases and causing the combustion efficiency to drop.

Figure 5 shows the carbon monoxide emissions plotted against equivalence ratio. It shows the same trend as combustion inefficiency curve which implies that the combustion efficiency was influenced by carbon monoxide oxidation. However, it could be seen that carbon monoxide emissions increase when applying water cooling heat exchanger. The carbon monoxide curves show the same curvature for both conditions, that is a drastic increase in the lean region which is due to insufficient residence time for carbon monoxide burnout at the low flame temperatures of these lean mixtures. Whereas, drastic increase in carbon monoxide emissions on the rich side is due to high equilibrium carbon monoxide at these rich equivalence ratios.

Unburned hydrocarbon emissions of less than 10 ppm can be achieved for both conditions over a wide range of operating equivalence ratios and this is shown in Figure 6. However, water cooling actually increases unburned hydrocarbon emissions rather than reducing them.

6.0 CONCLUSIONS

It could be concluded that there was a significant amount of NO_x emission reduction when applying water cooling heat exchanger. A reduction of 20 per cent was achieved at equivalence ratio of 0.9. However, this was achieved at the expense of increased in other emissions such as carbon monoxide and unburned hydrocarbon.

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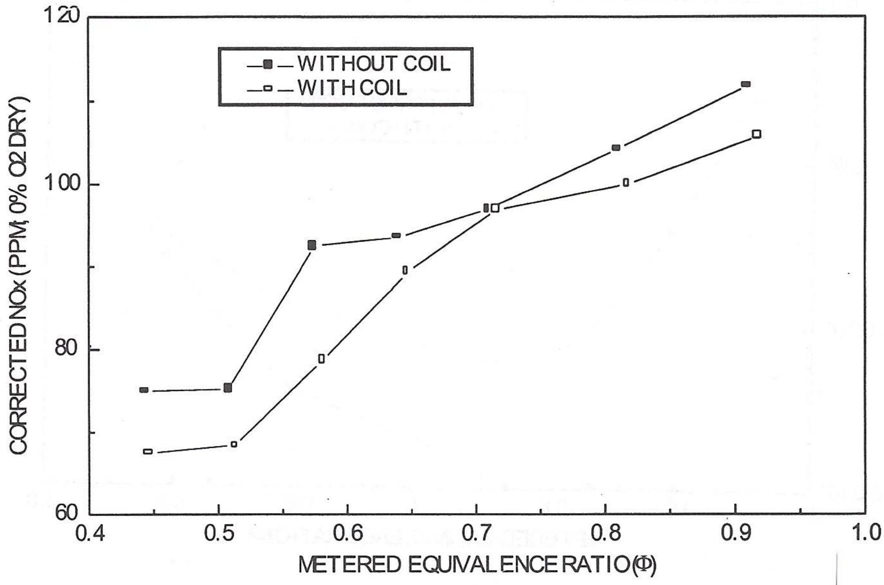

Figure 3 Corrected NO_x vs Metered Equivalence Ratio for 59 mm Orifice Plate $\Delta P = 2\ mmH_2O$, Fixed Straight Swirler

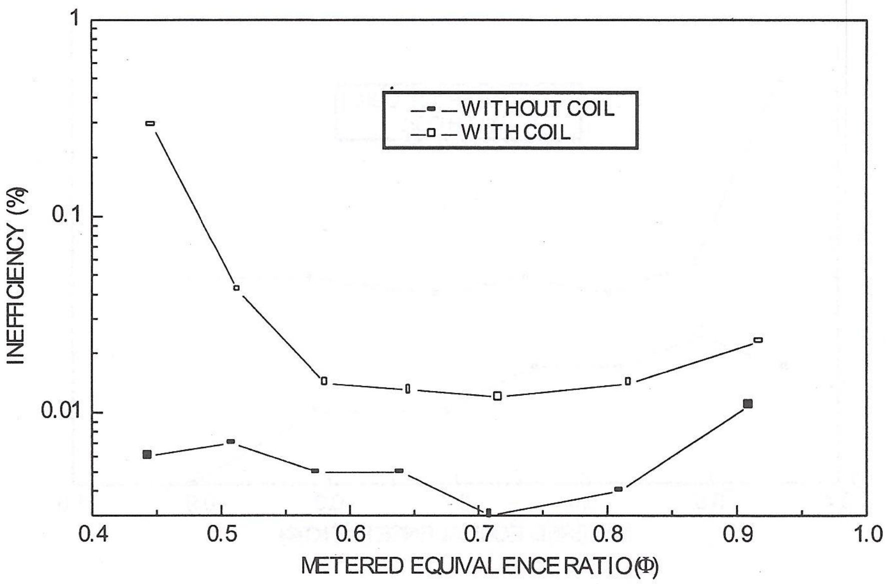

Figure 4 Inefficiency vs Metered Equivalence Ratio for 59 mm Orifice Plate $\Delta P = 2\ mmH_2O$, Fixed Straight Swirler

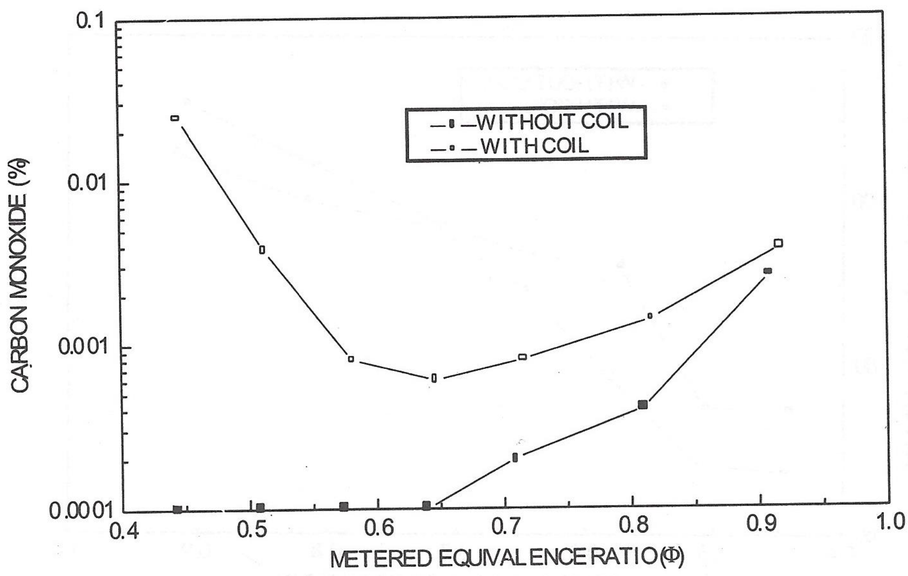

Figure 5 Carbon Monoxide vs Metered Equivalence Ratio for 59 mm Orifice Plate $\Delta P = 2$ mmH_2O, Fixed Straight Swirler

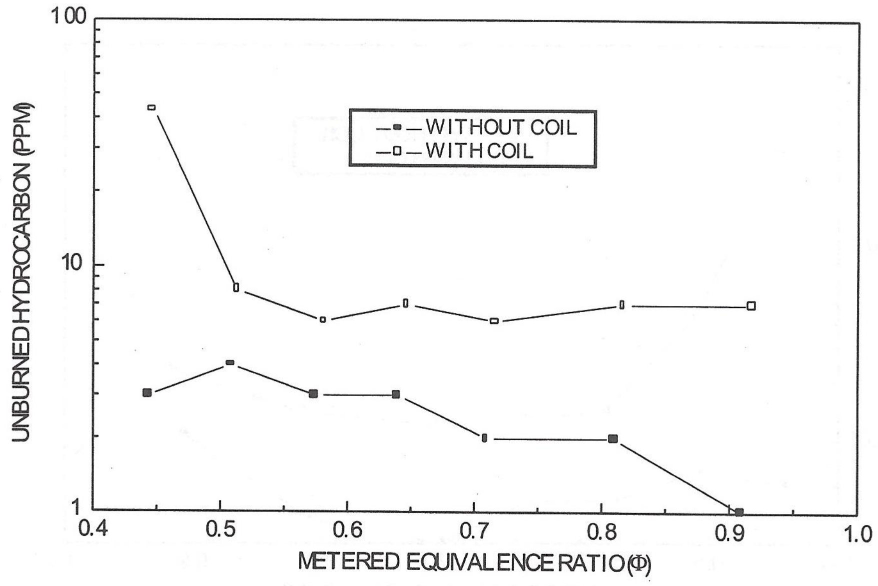

Figure 6 Unburned Hydrocarbon vs Metered Equivalence Ratio for 59 mm Orifice Plate $\Delta P = 2$ mmH_2O, Fixed Straight Swirler