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The Influence of Pressure Losses on Reducing NO_x 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 combustor of 140 mm inside diameter and 290 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 different sizes, i.e. 25 mm and 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 40 and 20 mm W.G. (water gauge) pressure losses for the 25 mm orifice plate, and 5 mm and 2mm W.G. for the 59 mm orifice plate to demonstrate the influence of modulating the burner by varying the pressure losses. A reduction of about 63.6 per cent was achieved at equivalence ratio of near stoichiometric (0.98) for the 40 mm W.G. pressure loss condition and a reduction of 40.5 per cent was achieved for the 20 mm W.G. pressure loss condition at equivalence ratio of 0.85. However, the 59 mm orifice plate tests demonstrate smaller reduction when using water-cooling.

Introduction

The effects of increased levels of NO_x in the atmosphere are wide reaching. In the atmosphere NO is rapidly oxidized to NO_2 and this form plays an essential role in the formation of troposphere ozone and photochemical smog, and is oxidized to form nitric acid that may then be deposited as acid rain [1]. At ground level, increased concentrations (above 0.06 ppm) of NO_2 can cause respiratory problem [2].

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 optimization process [3].

Basically there are two techniques of controlling NO_x: those, which prevent or reduce the formation of nitric oxide (NO) and those, which destroy NO from the products of combustion. The first 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 reducing the flame temperature. One method of modifying the combustion processes is to burn either 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, however, a reduction of the primary-zone flame temperature may increase the emissions of carbon monoxide and unburned hydrocarbon. On the other hand, the problem with fuel-rich combustion is the formation of soot and CO even though the stability margin is widened. Water or steam injection has been shown to be another effective technique to accomplish the above goal [4]. Other methods of NO_x control involve staged combustion, variable geometry combustion; lean premixed pre-vaporized combustion and catalytic combustion.

In the present work, a water-cooling heat exchanger, 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 distance from the flame. The effect of pressure losses was also studied for two different configurations, i.e. with 25 and 59mm orifice plate inserted at the exit plane of the swirler.

Review of Applications of Water Injection Technique

A typical NO_x reduction curve as a function of rate of water injection is shown in Fig. 1 [5]. These data were obtained in an aero derivative inductive gas turbine at full power. The same effect was also demonstrated by Fox and Schlein [6] who used the FT8 gas turbine combustor in their 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 [5, 7]. 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.

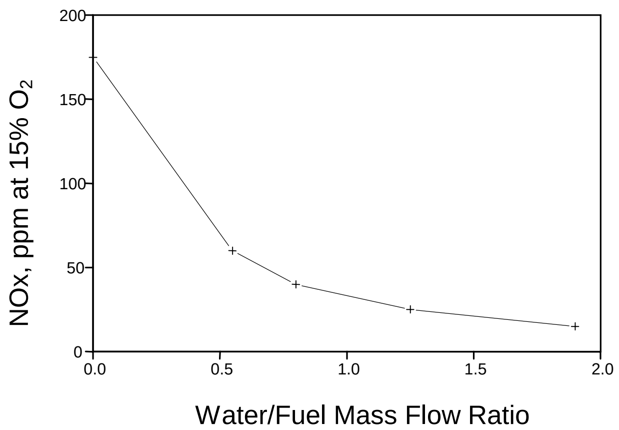
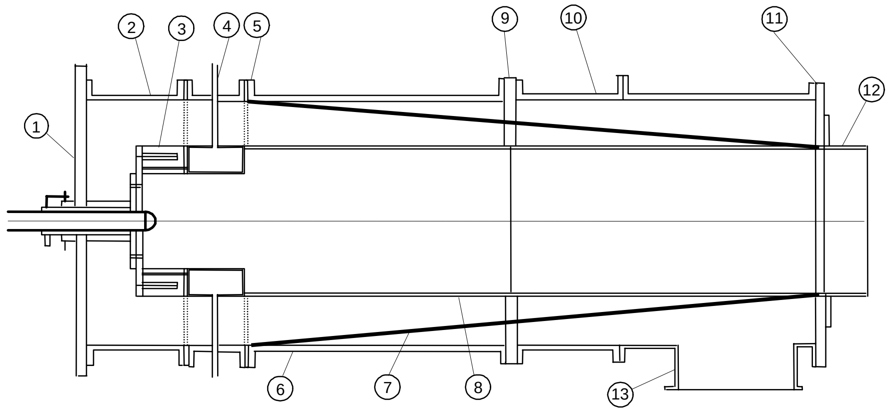


Fig. 1 NO_x as a function of water injection for a gas-turbine running on natural gas at a pressure ratio of 30

Experimental Set-Up

The general rig set-up for burner tests as used in the present work is shown in Fig. 2. The air introduced through the inlet pipe, designated (13) on the diagram, flows downward before entering the combustor through the fixed straight blade radial swirler of 76mm 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 preheating cannot be controlled externally. The exhaust sampling probe is mounted at the top of the end pipe.



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

Fig. 2 Burner test rig using cooling coil

There are two systems of supplying air for different air requirements: one of which used an air blower and the other used the laboratory compressed air supply. The higher volume requirements of air, usually needed for gas turbine conditions, used the fan blower; whereas the lower volume requirements of air, usually needed for domestic central heating burner systems, used the compressed air supply from the laboratory. In the present work the latter supply system was adopted.

The compressed air was passed through a variable area flow meter. A pressure tapping was used to monitor the inlet static pressure to the rotameter. The inlet temperature was monitored by a chrome-alumel type-K thermocouple fixed between the rotameter and the control valve. The airflow rates were then calculated, corrected for inlet temperature and pressure deviation from the rotameter calibration values of 15^oC and 760 mm Mercury barometric pressure.

The exhaust gas was sampled continuously by the gas analyzers. These analyzers were capable of a complete on-the-spot analysis of CO_2, UHC (unburned hydrocarbon), CO, NO, NO_x and O_2. The sampling system was designed and constructed to ensure that a representative samples were delivered to the various instruments.

An ‘X’ configuration gas sampling probe with 40 holes on centre of equal areas was used for mean exit sampling. It consisted of two stainless steel tubes welded eccentrically with water circulating in between the tubes to cool the exhaust gases and the probes. The sample was transported to the gas analyzers along a 7.6m heated Teflon sample line to be analyzed for CO_2, UHC, CO, NO, NO_x and O_2.

The sample gas from the combustor was pumped into the oven to draw samples to the analyzers. The sample gas was filtered in the oven and kept at 150^{o}C. The sample gas was then fed to the analyzers to be measured. The gas analyzers used and their detection ranges are summarized in Table 1.

Table 1 Particulars of gas analysis instruments

Gas Component	Analyzer or Detector	Reading Ranges	Manufacturer
CO_2	Non-Dispersive Infra-Red	4 ranges 0 - 0.1% 0 - 15%	Analysis Development Co
CO	Non-Dispersive Infra-Red	4 ranges 0 - 100ppm 0 - 1000ppm 0 - 1% 0 - 10%	Analysis Development Co
UHC	Flame Ion Detection	7 ranges 0 - 100ppm 0 - 10%	Signal Instrument
NO/NO_x	Chemiluminescent (Carbon Converter)	7 ranges 0 - 4ppm 0 - 400ppm	B.O.C. Luminox
O_2	Paramagnetic	0 - 100%	Taylor Instrument Servomex

The gas was analyzed for CO and CO_2 using an Analysis Development Co. (ADC) non-dispersive infra red (NDIR), analyzers. The CO and CO_2 infrared analyzers had two cells and two electrical amplification ranges. The CO analyzer had four ranges between 0-100ppm and 0-10%, the CO_2 analyzer also had four ranges but these were between 0-0.1% and 0-15%.

Flame ionization detectors (FID) were used to measure hydrocarbon (UHC) concentration in the sample gas. Fuel for the flame in the FID was a cocktail of 60 percent hydrogen and 40 percent helium burning in zero grade air, which was used to reduce the sensitivity of the FID responses to the sample oxygen level. In FID analyzers the electrical resistance of an H_2-He flame is monitored and when a sample gas that contains compound with C-H bonds is introduced to the flame ionization occurs with a subsequent reduction in the resistance of the flame. The resistance reduction is directly proportional to the concentration of UHC molecules present in the sample gas that is introduced to the flame. To prevent condensation of heavier hydrocarbon in the sample gas the sample gas lines to the FID analyzer are kept heated at 150^{o}C.

The oxygen concentration in the sample gas was measured by a Servomex paramagnetic analyzer with a single range of 0-100%. It was powered by self-contained batteries. Oxygen has the property of being paramagnetic due to the molecules unpaired electrons. The analyzer measures the paramagnetic susceptibility of the sample gas with oxygen content by means of a proven magneto-dynamic type-measuring cell.

The NO_x emissions in the sample gas were measured using a B.O.C. chemiluminescent analyzer (Luminox). The analysis of NOx in the sample gas is based on the emission of light radiation resulting from the reaction between NO and O_3. The intensity of the light emitted is measured by a photon detector and is proportional to NO concentration. A carbon-molybdenum converter was used to convert NO_2 from the sample gas to NO ahead of the reaction chamber and its temperature was controlled at 400^{o}C. The ozone was generated from zero grade air from the air purification system (APS). The air purifier was manufactured by Signal Instruments Control. Prior to any data being taken, all the analyzers had to be calibrated. The analyzers were calibrated using high accuracy calibration gases. A typical composition of the calibration gas was manufactured by Bedfont Scientific Ltd.

The raw data recorded was then fed to a computer programme to calculate the parameters required such as fuel mass flow, gas analysis based equivalence ratios, air humidity, adiabatic flame temperature, combustion inefficiency, and corrected NO_x emissions.

Test Conditions

Tests were carried out at around 400^oK 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 the tests the air inlet temperature exceeded 400^oK. Natural gas was used as fuel throughout the entire investigation. The parameter that was varied is the pressure losses.

Domestic central heating boilers operate with a fan air supply at below 0.5 percent (5 mb, 50 mm H_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.

Tests were conducted using two orifice plates of 25 mm and 59 mm diameter that were inserted at the exit plane of the outlet of the wall injector (refer to Fig. 2). This is to enhance flame stabilization and provide better mixing of air and fuel prior to ignition. The orifice plate also helps to prevent fuel from entraining into the corner re-circulation zone that will create local rich zone hence resulting in higher local NO_x emissions which in the end contribute to high total NO_x emissions. The other parameter that was varied is the pressure losses to modulate the burner performance. The influence of varying the pressure losses was investigated.

Results and Discussions

Figs. 3-6 show the effect of water-cooling on reducing mean exhaust emissions from gas burner system using straight blade radial swirler with the 25mm orifice plate inserted at the exit plane of the swirler outlet.

Fig. 3 shows the reduction in NO_x emissions when using water-cooling. A reduction of 40.5 per cent was achieved for the 20mm W.G. pressure loss condition at equivalence ratio of 0.85. This shows that water-cooling is quite effective in reducing NO_x emissions in gas burner system especially near stoichiometric condition or fuel rich condition. When the burner was modulated by increasing the pressure losses to 40mm W.G., a reduction of about 63.6 per cent was achieved at equivalence ratio of near stoichiometric (around 0.98). This demonstrates that burner modulation, i.e., by increasing the burner pressure losses tends to improve the combustion, hence increasing the NO_x emissions reduction. Even though the effect of varying the pressure losses was significant but there was no consistent influence with increase in pressure losses. The expected trend, if fuel and air mixing after the swirler exit is important, is a decrease in NO_x emissions with increase in pressure loss for lean mixtures. This is because good mixing, for near stiochiometric mixture, will result in more of the mixture burning at the maximum NO_x production rate at 0.9 equivalence ratio. If the mixing deteriorates then more mixture burns in richer or leaner regions, which have lower NO_x production and less burn at 0.9 equivalence ratios. Conversely, for lean mixtures the richer zone that burns locally for poor mixing always results in higher NO_x.

On the other hand, the major drawback for this water-cooling method is that the flame cannot be sustained when burning lean. This can be seen by looking at the lean flammability limits for both test conditions, i.e. 0.64 and 0.66 equivalence ratios for the 40 mm W.G. and 20 mm W.G. pressure losses condition respectively. These can be compared to the tests where water-cooling was not applied when the lean limit equivalence ratios are 0.42 and 0.5 for the 40 mmW.G. and 20 mm W.G. pressure losses condition respectively.

Fig. 4 shows the combustion inefficiency plotted against equivalence ratio for both tests conditions. It could be seen that there was some improvement in combustion efficiency for the 20mm W.G. pressure loss conditions when using water-cooling. However, near lean condition, combustion efficiency reduces drastically to near 99 per cent from about 99.97 per cent at equivalence ratio of near 0.8. This could be due to flame quenching when using water-cooling where the emissions of carbon monoxide increases and causing the combustion efficiency to drop. In the case of 40mm W.G. pressure loss, using water-cooling did not improve combustion efficiency at all and this could be seen for the whole range of operating equivalence ratios.

Fig. 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 for the 40mm W.G. pressure loss condition. There was some reduction in carbon monoxide emissions for the 20mm W.G. pressure loss condition, especially between equivalence ratios of 0.76 to 0.96. The carbon monoxide curves for all conditions show the same curvature, that is a drastic increase in the lean region. This is due to oxygen deficiency in the lean region.

Unburned hydrocarbon emissions of less than 100ppm can be achieved for both conditions over a wide range of operating equivalence ratios and this is shown in Fig. 6. However, water-cooling actually increases unburned hydrocarbon emissions rather than reducing them, except over a small range of equivalence ratios of 0.76 to 0.97 for the 20mm W.G. pressure loss condition. Even then, the reductions were quite small.

Figs. 7-10 show the effect of changing the orifice plate size to 59mm when applying water cooling heat exchanger on reducing mean exhaust emissions from gas burner system using straight blade radial swirler. Figs. 7-10 generally exhibit the same trend that was shown in Figs. 3-6 discussed previously.

Fig. 7 shows the reduction in NO_x emissions when using the 59mm orifice plate. 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. When the burner was modulated to 5mm W.G. pressure losses, the reduction achieved was 22.2 percent at equivalence ratio of 0.88 as compared to the case of 2mm W.G. pressure drop. This once again demonstrated the advantages of modulating the burner by increasing the pressure losses. This improvement is better than the lower pressure losses case since the reduction was achieved over the whole range of operating equivalence ratios. On the other hand, the lower pressure losses of 2mm W.G. demonstrated very small or no reduction at all near equivalence ratio of 0.64-0.70.

Fig. 8 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. The same trend of curves was obtained when increasing the pressure losses to 5mm W.G.

Fig. 9 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. The same trend of curves was obtained when increasing the pressure losses to 5mm W.G.

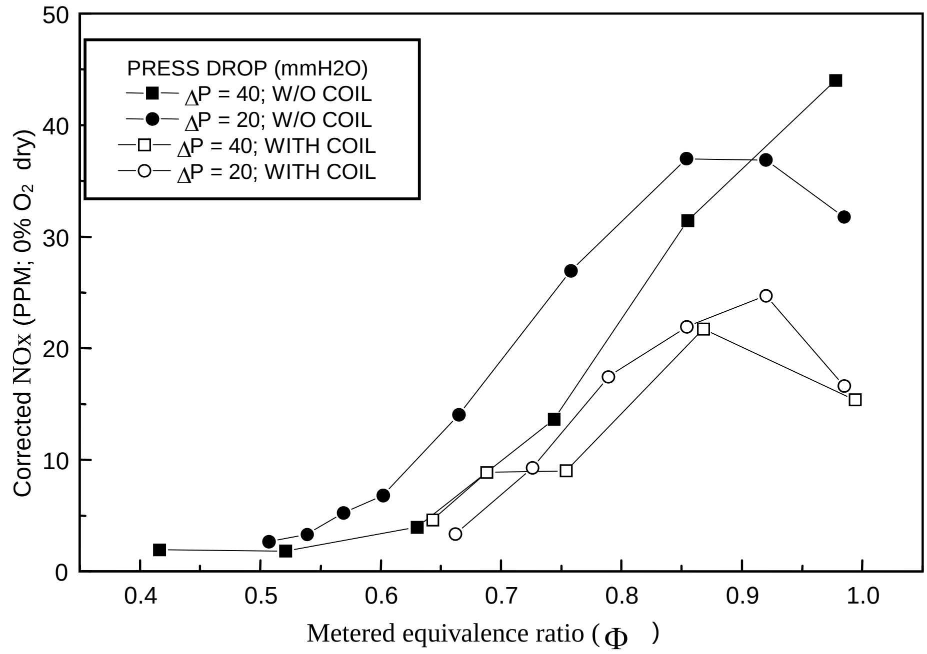


Fig. 3 Corrected NOx vs. metered equivalence ratio for 25mm orifice plate

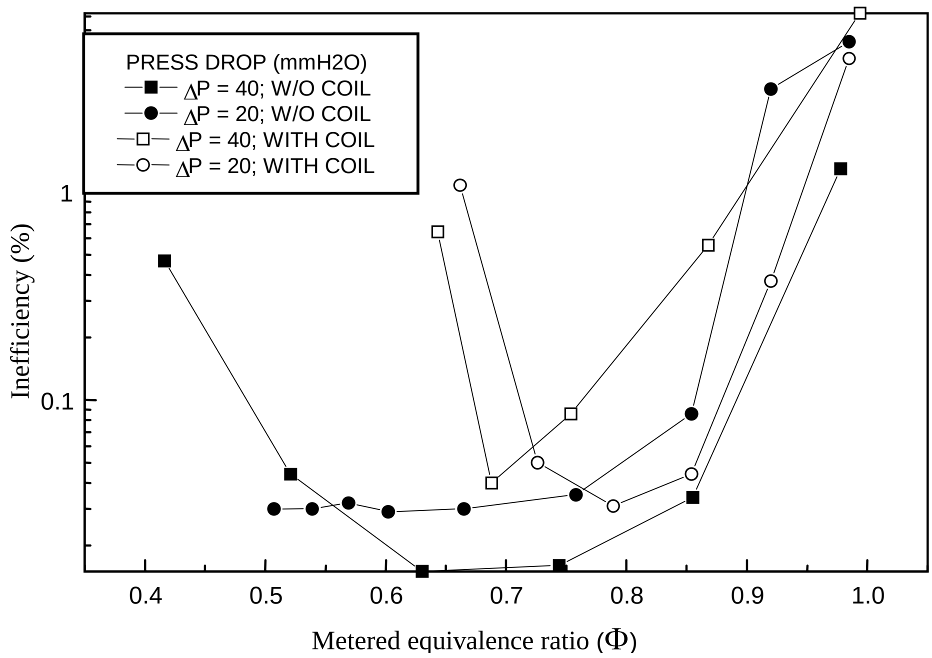


Fig. 4 Inefficiency vs. metered equivalence ratio for 25mm orifice plate

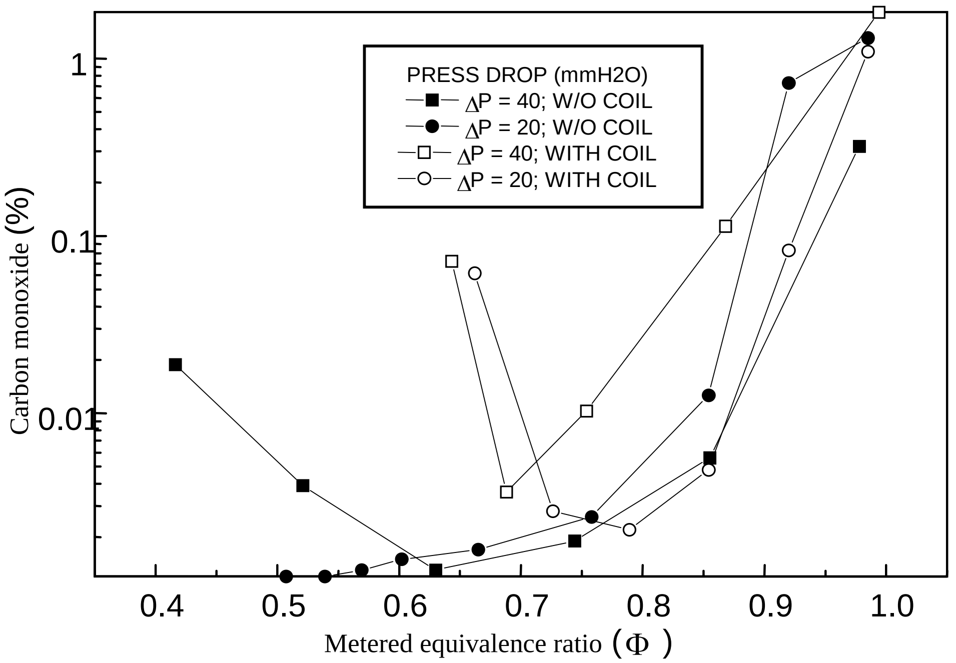


Fig. 5 Carbon monoxide vs. metered equivalence ratio for 25mm orifice plate.

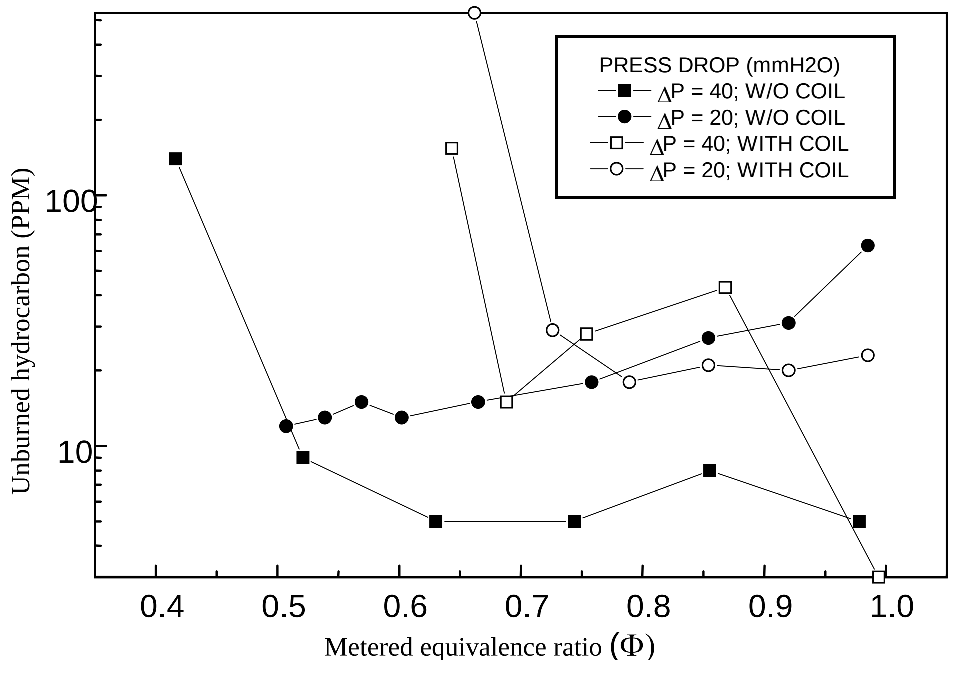


Fig. 6 Unburned hydrocarbon vs metered equivalence ratio for 25mm orifice

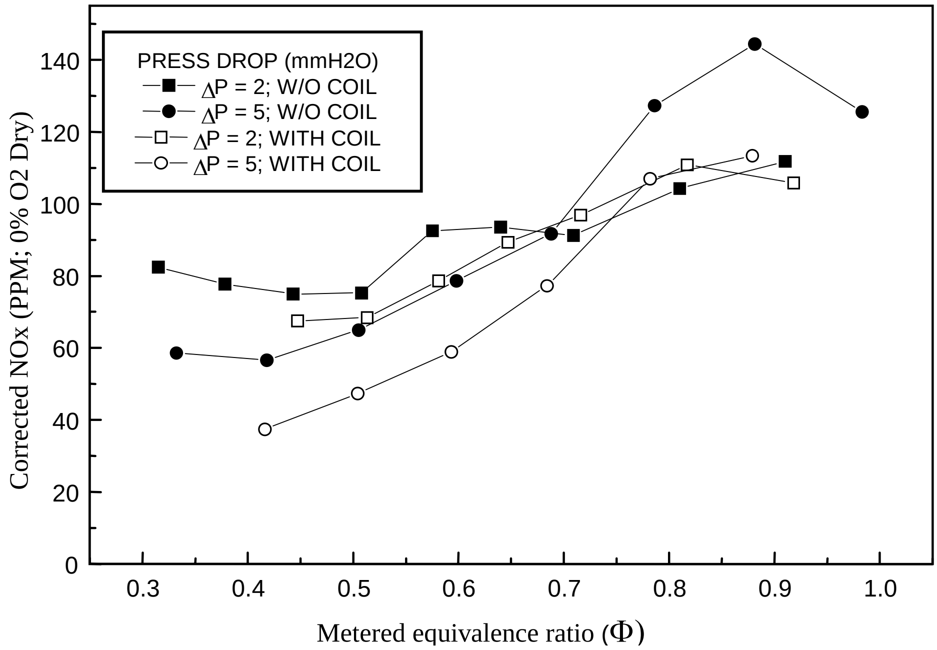


Fig. 7 Corrected NOx vs. metered equivalence ratio for 59mm orifice plate

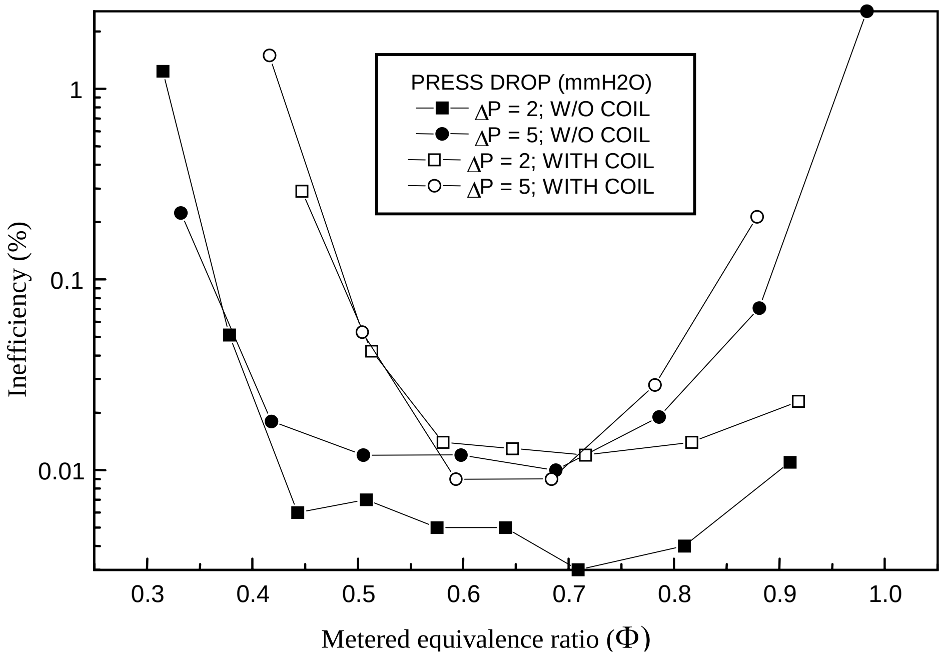


Fig. 8 Inefficiency vs. metered equivalence ratio for 59mm orifice plate

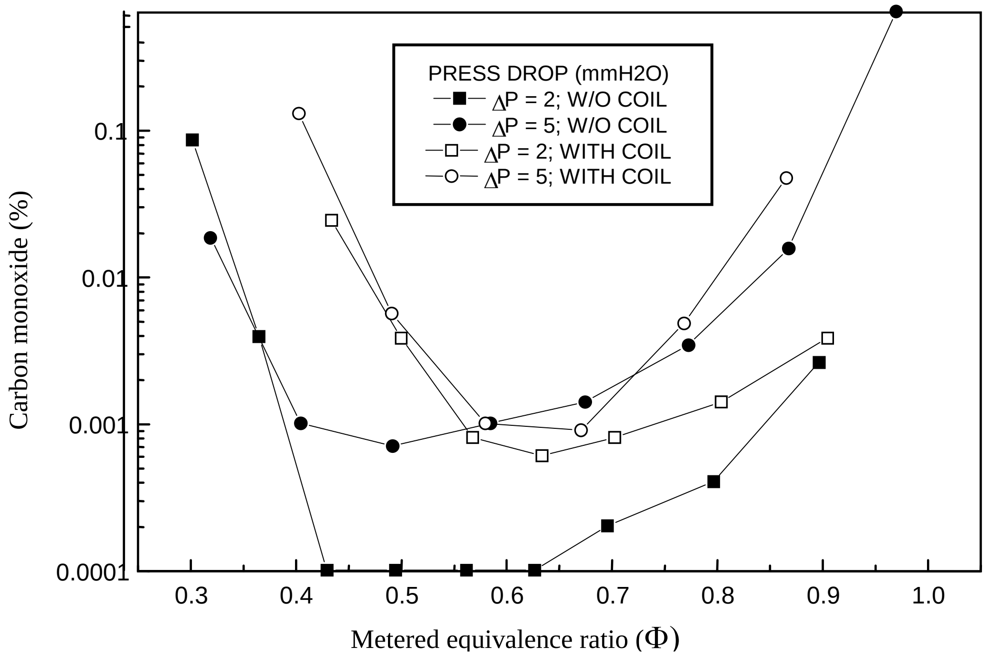


Fig. 9 Carbon monoxide vs. metered equivalence ratio for 59mm orifice plate

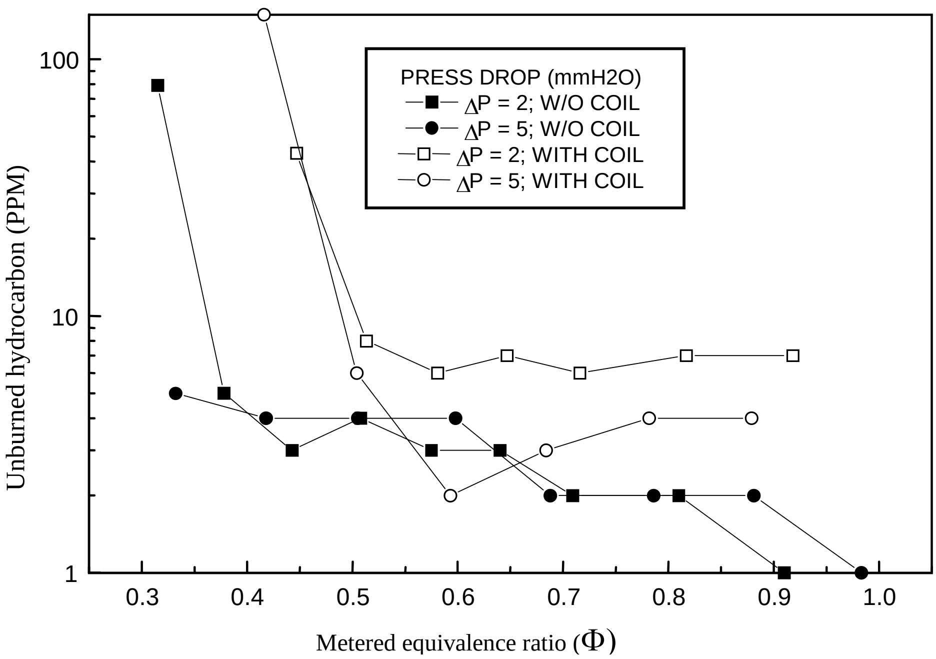


Fig. 10 Unburned hydrocarbon vs. metered equivalence ratio for 59mm orifice

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 Fig. 10. However, water-cooling actually increases unburned hydrocarbon emissions rather than reducing them. When increasing the pressure losses to 5mm W.G., a lower emission of UHC was obtained where UHC emissions of lower than 5 ppm were obtained over a wide range of equivalence ratios demonstrating better combustion due to better mixing.

Conclusions

Application of water cooling technique increased emissions of carbon monoxide and unburned hydrocarbon. Even the flammability limits were narrowed down especially near fuel lean conditions. Nevertheless, it could be concluded that there were a significant amount of reduction in NO_x emissions for all test conditions. A reduction of about 63.6 per cent was achieved at equivalence ratio of near stoichiometric for the 40mm W.G. pressure loss condition and a reduction of 40.5 per cent was achieved for the 20mm W.G. pressure loss condition at equivalence ratio of 0.85. This demonstrated the improvement achieved by increasing the burner pressure losses.

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