Combustion of LCV Coal Derived Fuel Gas for High Temperature, Low Emissions Gas Turbines in the British Coal Topping Cycle

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ABSTRACT

Advanced coal based power generation systems such as the British Coal Topping Cycle offer the potential for high efficiency electricity generation with minimum environmental impact. An important component of the Topping Cycle programme is the development of a gas turbine combustion system to burn low calorific value (3.5 - 4.0 MJ/m³ wet gross) coal derived fuel gas, at a turbine inlet temperature of 1260°C, with minimum pollutant emissions.

The paper gives an overview of the British Coal approach to the provision of a gas turbine combustion system for the British Coal Topping Cycle, which includes both experimental and modelling aspects. The first phase of this programme is described, including the design and operation of a low-NOx turbine combustor, operating at an outlet temperature of 1360°C and burning a synthetic low calorific value (LCV) fuel gas, containing 0 to 1000 ppmv of ammonia. Test results up to a pressure of 8 bar are presented and the requirements for further combustor development outlined.

NOMENCLATURE

Conversion Rate

\[
\frac{[\text{Total NOx}] - [\text{Thermal NOx}]}{\text{Air + Fuel Flow}} \times \frac{\text{Fuel Flow}}{\text{Fuel Flow}}
\]

\[I = \text{Combustion Intensity MW/m}^3\text{bar}\]

OTDF = Overall Temperature Distribution Factor, where

\[
\frac{\text{Peak Temp} - \text{Tex}}{\text{Tex} - \text{Ti}}
\]

\[\text{Pi} = \text{Mean Air and Fuel Inlet Pressure (Pa)}\]

RTDF = Radial Temperature Distribution Factor, where

\[
\frac{\text{Peak Circumferential Temp} - \text{Tex}}{\text{Tex} - \text{Ti}}
\]

\[\text{Ti} = \text{Mean Air and Fuel Inlet Temperature (K)}\]

\[\text{Tex} = \text{Mean Exhaust Temperature (K)}\]

\[V = \text{Combustor Volume (m}^3\text{)}\]

INTRODUCTION

British Coal are currently developing an advanced coal burning combined cycle power generation system known as the British Coal Topping Cycle to provide economic, high efficiency, low emissions technology for coal based power generation (Figure 1). In this system, coal is partially gasified in air using a spouted fluidised bed gasifier operating at around 980°C and 18 bar pressure to produce a low calorific value (LCV) fuel gas and a char residue. The fuel gas is cleaned at high temperature and burned in a gas turbine to produce electrical power. The char is combusted in either a Circulating Fluidised Bed Combustor (CFBC) or a Pressurised Fluidised Bed Combustor (PFBC) to produce steam to drive a steam turbine for further electrical power.

A major assessment study commissioned by British Coal and carried out by Bechtel Limited (Dawes et al, 1989), in which a number of power generation technologies were compared at 200 - 400 MWe size, identified the British Coal Topping Cycle as the preferred option for development. For this cycle, the potential advantages identified were:

1. A cycle efficiency approaching 45% (higher heating value basis) corresponding to around a 20% reduction in cost of electricity compared with conventional pulverised fuel fired plant with flue gas desulphurisation (pf + FGD).

2. Low nitrogen oxides (NOx) and sulphur dioxide (SO₂) emissions.

3. 20% reduction in carbon dioxide (CO₂) per unit of electricity generated compared with pf + FGD, a potential factor in global warming.
To achieve the near term objective of having a demonstration Topping Cycle plant in operation by the year 2000, development work has concentrated on the CFBC Topping Cycle described by Minchener (1990), the variant with the least development requirements and lowest technical risk.

Gas Turbine Combustion System Requirements

The decision to develop the CFBC Topping Cycle has meant that conventional gas turbine combustion technology, albeit modified to accommodate the features of the high temperature coal derived LCV fuel gas can be adopted. However, the duty and demands for the combustion system operating at a COT of 1360°C and a fuel gas temperature in the region of 600°C are sufficiently different from conventional technology to require separate development as part of British Coal's Topping Cycle development strategy. The particular features of the fuel gas which the combustion system must accommodate are:-

1. The calorific value of the fuel gas is predicted to be 3.5 - 4.0 MJ/m³ wet gross (approx 100 Btu/ft³), about one tenth the heat content of natural gas.

2. This low calorific value means that to achieve a COT of 1360°C, low air:fuel mass ratios, typically 2:1 or less, will be required. Consequently the fuel flow will be a significant factor in determining the combustor flow pattern. Further, the low air:fuel ratio will restrict the amount of air available for combustor wall cooling, introducing a potentially significant design constraint.

3. The fuel gas has small amounts of nitrogen containing species present, principally ammonia (about 300 ppmv is predicted), so that fuel-NOx will be the dominant source of the combustor NOx emission.

Development of a Gas Turbine Combustion System

Combustor development leading to the provision of a combustion system for utility size Topping Cycle plant will take place in several phases (Table 1). Phase 1 of this development assesses the performance of a preliminary combustor design at a nominally 1MWth scale. The design, manufacture and testing of this Topping Combustor was carried out by Aero and Industrial Technology Ltd. (AIT) under contract to British Coal.

<table>
<thead>
<tr>
<th>Year</th>
<th>Phase 1</th>
<th>Phase 2</th>
<th>Phase 3</th>
<th>Mathematical Modelling</th>
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<tbody>
<tr>
<td>1988</td>
<td></td>
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COMBUSTOR DESIGN RATIONALE

The tubo-annular type of combustor was chosen for initial development. This was considered the most likely geometry to be adopted in utility size machines with 1260°C turbine entry temperatures: General Electric having developed the Frame 9 machine with tubo-annular combustors. The Frame 9 would provide the correct gas turbine output for the utility size Topping Cycle plant.

The main requirement of the combustor was to achieve high combustion efficiency whilst maintaining low pollutant emissions, especially NOx. To achieve low NOx emissions, the design approach adopted was to stage the combustion by adding the combustion air in stages along the tube surface, such that the primary zone would be fuel rich with the subsequent intermediate and dilution zones being fuel lean. Thus, conditions in the primary zone would be achieved which would promote the reduction of nitrogen containing species in the fuel to elemental nitrogen, rather than oxidation to fuel-NOx. Additionally, a beneficial reduction in peak flame temperature would be achieved, thus minimising thermal-NOx formation.
To achieve high combustion efficiency, a combustor was designed with sufficient volume in relation to the throughput of fuel and air, to accommodate the high volume of inert in the fuel gas and the relatively slow burning rate of carbon monoxide (CO). This is of particular significance as operation with staged combustion may augment this problem. Thus, a combustor with a relatively modest Combustor Loading (Q) to ensure high combustion efficiency, and a length to diameter ratio of 2:1 to provide sufficient residence time was adopted.

In addition, effective mixing within the combustor to promote efficient combustion and low NOx was achieved, in part, by the use of flow swirlers to impart radial motion to portions of both the air and fuel, and in part by the interaction of the secondary air and the fuel gas jets. (See sections on Air Distribution and Fuel Admission).

The particular features of the combustor design are now outlined.

**Combustor Size**

The constraint on the combustor size was that the design data produced should be suitable for scale-up of the combustor design to larger scale units, typically those associated with a generically similar gas turbine in the region of 40MW. This represents a probable gas turbine size for an early demonstration Topping Cycle plant and thus provides the facility to take the scale-up to full utility size. The minimum size for the test Topping Combustor was considered to be about the 1MWth level when the combustor was operated at conditions representative of the Topping Cycle.

After consideration of the combustor loading and heat release properties of typical small scale industrial gas turbines, the following nominal values were adopted for the initial combustor evaluation:

- Combustor Loading (Q) = 0.6 kg/s m² Pa
- Combustion Intensity (I) = 6.3 MW/m² bar
- Air Casing Mach Number = 0.013

These relatively modest values were adopted initially, to ensure high combustion efficiency and acceptable flame stability whilst burning LCV gas. In terms of the combustor size, these values, together with the minimum size constraint equate to a 127 mm (5") internal diameter combustor.

**Air Distribution**

For the initial phase of this programme, a conventional air distribution design was chosen with the primary air flow being admitted through a swirler and the secondary, intermediate and dilution flows being admitted through plain holes along the flame tube surface (Figure 2).

The primary air swirler and secondary air admission holes were sized in conjunction with the fuel injector to produce a fuel rich, stable primary toroidal recirculation zone. The intermediate air admission was arranged to ensure complete combustion of any CO remaining. The theoretical adiabatic flame temperature as a function of the air:fuel ratio is shown in Figure 3.

**Fuel Admission**

The fuel was admitted through an injector positioned at the upstream end of the flame tube, inboard and concentric with the primary air swirler (Figure 2).

To promote rapid mixing of the fuel and air, approximately two thirds of the fuel was admitted through a swirler injecting in a contra direction to the air swirler. This technique had previously been successfully employed for low calorific value fuels by Beebe and Blanton (1985).

The remaining fuel was introduced directly into the primary zone through plain holes positioned inboard of the fuel swirler. The injection angle of these holes was arranged to support the primary zone toroidal vortex produced by the flow reversal from the air secondary holes.
The nett fuel swirl momentum and the momentum values of the directly injected fuel and secondary air were estimated, to ensure that the arrangement would not produce excessive residual swirl at the combustor outlet.

**Combustor Pressure Losses**

The low aerodynamic and heat release loadings adopted for the combustor permitted the use of relatively low pressure loss levels.

For the initial design, an air pressure loss of approximately 1% was used, with a fuel injection pressure loss of approximately twice this value. However, it should be noted that the cycle efficiency figures for the Topping Cycle, quoted in the introduction to this paper can be achieved with combustor air pressure loss of greater than 5%.

**Wall Cooling**

Serious consideration of techniques to achieve satisfactory combustor wall temperatures did not form part of the initial programme. To avoid premature failure of the flame tube, however, some 30% of the air was distributed through 4 splash cooling skirts along the flame tube wall and via a number of sweat cooling holes in the exhaust duct. The effects of wall cooling films is of secondary importance in determining the parameters of interest to the Phase I programme.

To design a combustor with a commercial life would require further work to optimise the efficient distribution of the limited amount of air available for effective protection of the relatively large flame tube surface. It may be necessary to incur extra pressure loss in either the air or the fuel streams for external convective or impingement cooling, or to introduce additional features to utilise the cooling potential of the fuel. However, until the crucial factor of optimum combustor loading can be determined through the development programme, an exercise in optimising liner cooling is not considered necessary.

**Test Equipment**

**Test Rig Description**

The layout of the pressure test rig is shown in Figure 4.

![Diagram of Combustor Test Rig](image)

Air is supplied from a compressor at up to 8 bar pressure. The air is metered and the temperature adjusted to 350°C by passing through a gas/air heat exchanger, which is heated by a separate liquid fuelled combustor.

The test Topping Combustor is designed with a discharge duct of circular to sector cross-sectional area, which is compatible with an existing exhaust plane traverse system. This consists of a water cooled exhaust drum and adaptor plate. Transverse probes are fitted to monitor either temperature or gas composition at the combustor outlet. The probes are fitted with four sampling points at different radii across the sector and are moved at intervals of 2° across a 36° sector. Platinum/platinum 13% rhodium Type B thermocouple are used in the thermocouple probes, with appropriate heat shields. Approximately 10 downstream of the exhaust drum, a crucifix sampler based upon EPA design rules is fitted. All gas samplers are cooled by water at 150°C and the sample transfer lines are electrically heated to 150°C.

**Fuel Supply**

A synthetic fuel gas, with similar calorific value and average molecular weight typical of an air blown gasifier fuel gas, is supplied in a high pressure tube trailer. The constituents and properties of this gas are given in Table 2 where they are compared with the composition of a typical fuel gas from an air blown gasifier. At this stage of the combustor development programme, the combustible component of the fuel gas is represented by carbon monoxide and hydrogen. Subsequent tests will include methane in the synthetic fuel gas to achieve improved similarity with air blown gasifier fuel gas. The methane content of the fuel gas has been shown to be important in determining the NH₃ to NOₓ conversion rate by Sato et al (1990). At this stage, for ease of preparation, the inert is simulated by using nitrogen only. When required, ammonia is injected into the gas supply stream close to the fuel injector. The synthetic fuel gas is supplied to the test Topping Combustor at temperatures in the range 20 - 220°C.

<table>
<thead>
<tr>
<th>CONSTITUENT</th>
<th>TYPICAL FUEL GAS</th>
<th>SYNTHETIC FUEL GAS</th>
</tr>
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<tbody>
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<td>CO % vol</td>
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<td>18.5</td>
</tr>
<tr>
<td>H₂ % vol</td>
<td>15.0</td>
<td>15.0</td>
</tr>
<tr>
<td>CH₄ % vol</td>
<td>1.5</td>
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<td>N₂ % vol</td>
<td>47.3</td>
<td>66.5</td>
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<td>H₂O % vol</td>
<td>11.5</td>
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</tr>
<tr>
<td>CO₂ % vol</td>
<td>10.0</td>
<td>0</td>
</tr>
<tr>
<td>Ammonia ppmv</td>
<td>300</td>
<td>0 - 1000</td>
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<td>Molecular</td>
<td>24.4</td>
<td>24.1</td>
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PHASE 1 TEST PROGRAMME

To assess the performance of the test Topping Combustor design, an initial three stage test programme (Phase 1) was undertaken. These stages were water flow visualisation, ignition and stability tests at atmospheric pressure, and combustion and emissions performance at pressures up to 8 bar.

Water Visualisation

Prior to combustion testing, the flow pattern within the combustor was assessed. This was done using a full scale replica of the combustor manufactured from clear acrylic material (Perspex) and installed in the Flow Visualisation Laboratory of CTEC.

In the test which simulated the design air:fuel ratio of 1.7:1, the initial design resulted in an asymmetric flow pattern within the combustion chamber. This was attributed to the flow from the fuel injector swirler. Modifications to improve the distribution inside the fuel injector body were successful and produced the symmetrical flow pattern shown in Figure 5. Further, at this air:fuel ratio of 1.7:1 the contra rotating swirler gave a balanced flow pattern with no residual swirl. Based on these observations and improvements to the original design, similar modifications were made to the test Topping combustor, prior to ignition and stability tests.

Ignition and Stability

The ignition and stability performance of the test Topping Combustor were determined using air and fuel inlet conditions at atmospheric pressure and ambient temperature (the measured ignition and stability limits are shown in Figure 6).

Ignition was obtained easily with the LCV value gas using a high energy surface discharge igniter fed from a 12 Joule ignition box. Thus, the liquid fuel torch igniter system, provisionally installed, was not required.

Stability was encouraging, with a stable flame achieved at a wide range of air:fuel mass ratios (up to 15:1 at the proposed elevated pressure and temperature nominal test conditions). Indeed, the effect of increasing pressure has been shown to increase the range of air:fuel ratios at which a stable flame is produced (Lefebvre, 1983), so it was expected that a stable flame would be produced at even higher air:fuel ratios in the elevated pressure tests.

Elevated Pressure Tests

The nominal operating conditions of this phase of the programme are given in Table 3. The results achieved during the elevated pressure tests are presented in the following section.

Table 3
Elevated Pressure Test Programme Nominal Operating Conditions

<table>
<thead>
<tr>
<th>Pressure</th>
<th>3, 5, 7, 8 bar</th>
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</thead>
<tbody>
<tr>
<td>Air Inlet Temperature</td>
<td>350°C</td>
</tr>
<tr>
<td>Fuel Inlet temperature</td>
<td>20, 120, 220°C</td>
</tr>
<tr>
<td>Air:Fuel Mass Ration</td>
<td>2:1 to 4:1</td>
</tr>
<tr>
<td>Ammonia doping</td>
<td>0, 330, 660, 990 ppmv</td>
</tr>
<tr>
<td>Air $\dot{M}_A$ = constant</td>
<td></td>
</tr>
</tbody>
</table>

RESULTS

Combustion Efficiency

Over the range of conditions tested, the combustion efficiency as determined by gas analysis was close to 100% (Figure 7). This high combustion efficiency was achieved with values of combustor loading ($\phi$ up to 1.5) considerably higher than the design value of $\phi$ = 0.6. These high values of combustor loading were achieved by reducing the operating pressure to 3 bar. Thus, there is scope to increase the combustor loading without compromise of high combustion efficiency. The advantage of operating with a high loading parameter would be to reduce the combustor size for a given heat output. Consequently the area of combustor liner to be cooled would be reduced.
FIGURE 7 EFFECT OF COMBUSTOR LOADING ON COMBUSTION EFFICIENCY

Exhaust Temperature Distribution

The overall temperature distribution factor (OTDF) varied between 4% and 10% across the range of conditions tested. The radial temperature distribution factor (RTDF) varied between 2% and 4%. This indicates that an acceptable temperature profile to the gas turbine first stage rotor blades may be achieved with the current design of combustor.

Pressure Loss

The air side pressure loss was determined to be 0.8% compared with a design value of 1%. This relatively low pressure loss is compatible with the low combustor loadings adopted in the initial design. A higher pressure drop of approximately 4% would be required for an engine combustor scaled from the test Topping Combustor and operating at the same COT of 90. Since the overall Topping Cycle efficiency of 45% can be achieved with a combustor pressure loss of greater than 5%, there is still some pressure loss available for improved wall cooling.

Metal Temperatures

In assessing the metal temperature using thermal indicator paint, it must be noted that no attempt was made, at this stage, to optimise the cooling. When operating at the design COT of 1360°C, a considerable area of the flame tube wall was above 800°C as shown in Figure B. However, as indicated above there is scope to reduce the combustor metal temperatures by more efficient utilisation of the available cooling air.

NOx Emissions

Thermal-NOx

For all test conditions studied, the NOx emission in the absence of ammonia addition to the fuel gas i.e. thermal NOx, was below 5 ppmv (dry, 15% O2) with COT in the range 970 - 1370°C. Although thermal-NOx increased slightly with pressure between 3 and 8 bar (Figure 9), the effect was negligible and certainly within the accuracy of measurement. Thus, thermal-NOx at 14 bar pressure, required for the Topping Cycle application may be similarly low.

Fuel-NOx

When ammonia was added to the fuel gas, higher levels of NOx were produced due to the formation of NOx from the oxidation of ammonia species (fuel-NOx). The importance of this fuel-NOx component increased as the level of ammonia increased, such that at the 1000 ppmv of ammonia level, fuel-NOx represented around 90% of the total NOx emission (Figure 10). However, the efficiency at which the ammonia was converted to NOx decreased as the ammonia level increased, so that the relationship between fuel-NOx and ammonia is non-linear. At the 300 ppmv level of ammonia (that predicted for an air-blown gasifier in the Topping Cycle application), the overall NOx emission at 7 bar with a COT of 1300°C was about 20 ppmv (dry, 15% O2). This corresponds to an ammonia to NOx conversion rate of about 42%

There is some indication that when the combustor was operated at a higher COT of 1370°C, achieved by increasing the fuel inlet temperature from 20 to 220°C, the ammonia to NOx conversion rate and hence fuel-NOx decreased, as shown in Table 4 and Figure 11. The decrease in fuel-NOx at the higher COT was greater than the corresponding increase in thermal-NOx, producing a decrease in the overall NOx emission. Thus, with 300 ppmv of ammonia, the overall NOx emission was about 15 ppmv (dry, 15% O2) at 7 bar and COT of 1370°C. This corresponds to ammonia to NOx conversion rate of about 32%.
This result is of potential importance in any combustor design modifications as it indicates that whilst operation with a fuel rich primary zone is beneficial for minimising fuel-NOx, an optimum value for primary zone air:fuel ratio must exist which also maintains a sufficiently high temperature to promote the reactions which reduce ammonia.

**Table 4**
Effect of Fuel Gas Temperature on NOx Emission and Conversion Rate

<table>
<thead>
<tr>
<th>Fuel Gas Temperature = 30°C</th>
<th>COT = 1300°C</th>
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</thead>
<tbody>
<tr>
<td>Ammonia (ppmv)</td>
<td>0 360 660 1120</td>
</tr>
<tr>
<td>NOx (dry)</td>
<td>8 52 101 115</td>
</tr>
<tr>
<td>NOx (dry, 15% O₂)</td>
<td>3 21 40 45</td>
</tr>
<tr>
<td>Conversion Rate %</td>
<td>- 32 40 27</td>
</tr>
</tbody>
</table>

**Fuel Gas Temperature = 130°C**

| Ammonia (ppmv) | 0 | 360 | 660 | 1050 |
| NOx (dry)      | 11| 51  | 83  | 104  |
| NOx (dry, 15% O₂) | 4 | 20  | 33  | 42   |
| Conversion Rate % | - | 31  | 30  | 24   |

Nitrous Oxide (N₂O)

In addition to measurement of thermal and fuel NOx, measurements were also made of N₂O. The concentration of N₂O was less than 1 ppmv for all test conditions studied. This indicates that although the adiabatic flame temperature is lower when burning a LCV fuel gas compared with conventional fuels, it is still above that at which significant levels of N₂O are produced.

**CONCLUSIONS**

A gas turbine combustor for application in the British Coal Topping Cycle, based on conventional gas turbine combustion technology has been designed at the 1MWth scale. Preliminary testing of the combustor has been encouraging, with a high (>99.9%) combustion efficiency achieved whilst maintaining low OTDF and RTDF (less than 10% and 4% respectively) and achieving low NOx emissions. The NOx emission achieved was typically 15 - 20 ppmv (15% O₂ dry) at the predicted ammonia in fuel gas (300 ppmv) level for an an air blown gasifier in the Topping Cycle application. Thus a gas turbine combustor has been designed which merits further development work.

**FURTHER WORK**

The remaining tests within Phase 1 of the combustor development programme will assess the potential effects of methane on combustor performance, particularly NOx. Methane has been shown previously to affect the NH₃ to NOx conversion rate (Sato et al, 1990). In addition, the combustor loading parameter, initially set at 0.6 will be increased without compromise of high combustion efficiency.

Phase 2 of the combustor development programme scheduled to commence in April 1991, will assess the effects on combustor performance of increasing the fuel gas temperature to 600°C. This fuel gas temperature corresponds to the intermediate
temperature gas cleaning option being developed for the Topping Cycle. In addition, the combustor design will be modified to include an improved combustor cooling system to achieve acceptable combustor metal temperatures.

ACKNOWLEDGEMENTS

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Any views expressed are those of the authors and not necessarily those of the supporting organisations.

REFERENCES


