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# A Study of Low NOx Combustion in Medium-Btu Fueled 1300℃-class Gas Turbine Combustor in IGCC

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## ABSTRACT

The development of integrated coal gasification combined cycle (IGCC) systems ensures cost-effective and environmentally sound options for supplying future coal utilizing power generation needs. The Japanese government and the electric power industries in Japan promoted research and development of an IGCC system using an air-blown entrained-flow coal gasifier. We worked on developing a low-Btu fueled gas turbine combustor to improve the thermal efficiency of the IGCC by raising the inlet-gas temperature of gas turbine.

On the other hand, Europe and the United States are now developing the oxygen-blown IGCC demonstration plants. Coal gasified fuel produced in an oxygen-blown entrained-flow coal gasifier, has a calorific value of 8.6MJ/m³ which is one fifth that of natural gas. However, the adiabatic flame temperature of oxygen-blown medium-Btu coal gaseous fuel is higher than that of natural gas and so NOx production from nitrogen fixation is expected to increase significantly. In the oxygen-blown IGCC system, a surplus nitrogen in quantity is produced in the oxygen-production unit. When nitrogen premixed with coal gasified fuel is injected into the combustor, the power to compress nitrogen increases. A low NOx combustion technology which is capable of decreasing the power to compress nitrogen is a significant advance in gas turbine development with an oxygen-blown IGCC system. We have started to develop a low NOx combustion technology using medium-Btu coal gasified fuel produced in the oxygen-blown IGCC process.

In this paper, the effect of nitrogen injected directly into the combustor on the thermal efficiency of the plant is discussed. A 1300℃-class gas turbine combustor with a swirling nitrogen injection function designed with a stable and low NOx combustion technology was constructed and the performance of this combustor was evaluated under atmospheric pressure conditions. Analyses confirmed that the thermal efficiency of the plant improved by 0.2 percent (absolute), compared with a case where nitrogen is premixed with coal gasified fuel before injection into the combustor. Moreover, this new technique which injects nitrogen directly into the high temperature region in the combustor results in a significant reduction in NOx production from nitrogen fixation. We estimate that CO emission concentration decreases to a significant level under high pressure conditions, while CO emission concentration in contrast to NOx emission rises sharply with increases in quantity of nitrogen injected into the combustor.

## NOMENCLATURE

| | | |
|---|---|---|
| HHV | :higher heating value of the fuel | MJ/m³ |
| LHV | :lower heating value of the fuel | MJ/m³ |
| Lb | :loading rate in the combustor | W/(m³·Pa) |
| N2/Fuel | :nitrogen by the fuel supply ratio | kg/kg |
| P | :pressure in the combustor | MPa |
| Tadia | :adiabatic flame temperature | ℃ |
| Tair | :air inlet temperature | ℃ |
| Tfuel | :fuel inlet temperature | ℃ |
| Tex | :combustor exit gas temperature | ℃ |
| $T_{N2}$ | :nitrogen inlet temperature | ℃ |
| $\phi$ | :equivalence ratio ( inverse way of air/fuel ratio ) | |
| $\phi$ ex | :equivalence ratio at combustor exit | |
| $\phi$ p | :equivalence ratio in the primary combustion zone | |

## INTRODUCTION

IGCC is considered one of the most important systems for future coal utilization technology in power generation systems, and is being promoted by Japan, the United States and Europe. In Japan, the government and electric power companies have been undertaking experimental research for a 200T/D pilot plant program (Ichikawa, 1996), from 1986 to 1996. The Central Research Institute of Electric

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Power Industry (CRIEPI) has developed an air-blown pressurized two-stage entrained-flow coal gasifier (Kurimura et al, 1995), a hot gas cleaning system (Nakayama et al, 1990), 150MW, 1500 ℃-class gas turbine combustor technology (Hasegawa et al., 1997). Furthermore, the government and electric power companies have started feasibility studies at a demonstration IGCC plant from 1996.

Other studies concerning the IGCC system and gas turbine combustor using the oxygen-blown coal gasified fuel include: The Cool Water Coal Gasification Project (Savelli and Touchton, 1985), the flagship demonstration plant of IGCC; the SHELL process (SGCP) (Bush et al., 1991) in Buggenum as the first commercial plant, which started test operation in 1994 with commercial operation was expected from 1998; the Wabash River coal gasification repowering plant (Roll, 1995) in the United States, in operations since 1995; and the TEXACO process at the Tampa power station (Jenkins, 1995), in commercial operation since 1996. Furthermore, the diversification of fuels used for the electric power industry, such as biomass, poor quality coal and residual oil, are also the most significant issues for gas turbine development in IGCC: The development of biomass-fueled gasification received considerable attention in the United States in the early 1980s (Kelleher, 1985) and the prospects for commercialization technology appear considerably improved at present (Consonni et al., 1997); Our research institute started researching the gasification technology of orimulsion fuel (Ashizawa et al., 1996). All of the systems which used oxygen as an oxydizer were assumed to adopt the wet type syngas cleaning system. Moreover, in almost all systems, NOx emission is controlled and gas turbine output is increased by premixing the surplus nitrogen, produced from the oxygen production unit, with a gaseous fuel. From the view point of both expensive running costs and initial costs of removing the NOx in exhaust gas derived from the gas turbine system, the electric power industry aimes for low-NOx combustion technology that promises higher efficiency and environmentally sound options.

According to the research into low-NOx combustion technology using medium-Btu gaseous fuel, other studies include: White et al. (1983) studies on the rich-lean combustor for low and medium-Btu gaseous fuels; Döbbeling et al. (1994) studies on low NOx combustion technology which quickly mixed fuel with air using the ABB double cone burner (called EV burner). (Since the flame speed of medium-Btu fuel is about 6 times greater than conventional natural gas, a lean premixed combustion for low NOx emissions has so far been difficult to adopt); Döbbeling et al. (1996) studies on the premixed combustion technology of medium-Btu gaseous fuel in a small burner for low NOx emission; Cook et al. (1994) studies on the effective method of returning nitrogen to the cycle, where nitrogen is injected from the head end of the combustor for NOx control; Zanello and Tasselli (1996) studies on the effects of steam content in the medium-Btu gaseous fuel on combustion characteristics. There are no studies on low-NOx combustion technology using surplus nitrogen actively.

This paper will propose a low-NOx combustion technology for medium-Btu gaseous fuel and provide useful engineering guidelines for the research and development of a gas turbine combustor using nitrogen injection technology at the swirler of the combustor for NOx control.

## DESCRIPTION OF THE OXYGEN-BLOWN IGCC SYSTEM

### Characteristics of the Oxygen-blown IGCC System

In the oxygen-blown IGCC system, nitrogen in large quantities is produced in the air separation unit. In almost all of the systems, coal gasified fuels, premixed with the rest of the nitrogen not used to feed coal into the gasifier, etc., are injected into the combustor to increase electric power and to decrease NOx emissions from the gas turbine.

In this paper, we propose a system, shown in figure 1, in which nitrogen is directly injected into the gas turbine combustor. This system is based on the practical gasification of coal using an oxygen-blown entrained-flow gasifier at elevated pressure and temperature; the use of aqueous scrubbing to clean a fuel gas; a reheated double high pressure steam system; and, the development of 1300 ℃-class (combustor-outlet gas temperature is about 1400 ℃) gas turbines, which have the potential to significantly increase the efficiency of the IGCC system in excess of 43 percent (higher heating value basis). Furthermore, it is necessary to return a large quantity of nitrogen produced from the air-separation unit (as much as the fuel flow rate), to the cycle from the standpoint of recovering power for oxygen production. Basically, the flow rate of the surplus nitrogen produced in the air-separation unit is almost proportional to the fuel flow rate at any gas turbine load, and all surplus nitrogen should be effectively injected into a gas turbine combustor prior to a turbine, while surplus nitrogen fluctuates little in proportion to changes in the gas turbine load. When the nitrogen flow rate over fuel flow rate ($N_2$/fuel) increases, the thermal efficiency and the output of IGCC plant slightly increase. For example, the output of IGCC increases about 1.3 percent, and thermal efficiency rises relatively by about 0.2 percent when the $N_2$/fuel ratio increases from 1.1 to 1.22.

### Characteristics of Oxygen-blown Gasified Fuel

The typical compositions of medium-Btu gaseous fuel produced in oxygen-blown gasifiers are shown in Table 1. Each gaseous fuel produced some raw materials with CO and $H_2$ as the main combustible components, and a small amount of $CH_4$. Calorific values

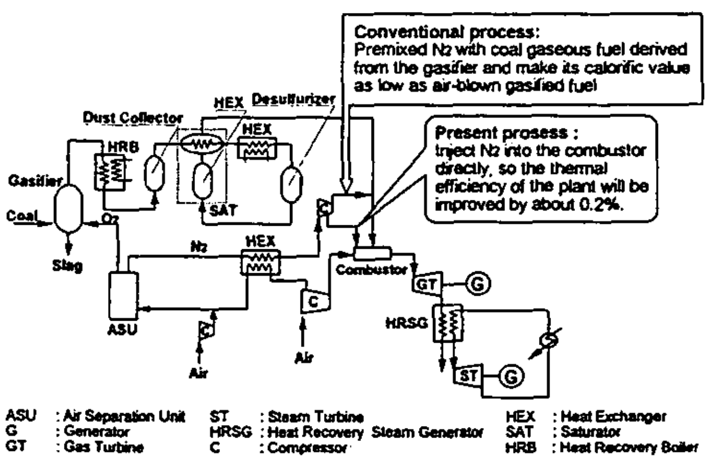


**Fig.1 Schematic diagram of oxygen-blown IGCC system**

Table 1 Typical compositions delivered from the oxygen-blown gasifiers

| Fuel | Coal | | | | Biomass | Heavy residue | Orimulsion™ |
|---|---|---|---|---|---|---|---|
| Gasifier type | Fixed | Entrained | | | Entrained | Entrained | Entrained |
| Fuel feed | Dry | Dry | | Slurry | | | |
| Developer | BGL | Shell | Hycol | Texaco | | Texaco | |
| Composition | | | | | | | |
| CO | 56.4% | 65.2–67.8% | 55.2–59.4% | 49.0% | 21.9–23.1% | 51.7% | 43.5% |
| $H_2$ | 25.6% | 28.8–31.0% | 31.1–33.7% | 34.0% | 12.5–22.4% | 43.1% | 42.2% |
| $CH_4$ | 6.6% | 0.01% | 1.0– 2.0% | 0.2% | 2.2% | 0.2% | 0.4% |
| $CO_2$ | 2.8% | 1.0– 2.8% | 7.6–10.4% | 9.7% | 20.7–18.6% | 3.2% | 11.8% |
| $H_2O$ | - | (Dry base) | - | - | 40.9–31.5% | - | (Dry base) |
| $NH_3$ | unknown | 100– 600ppm | unknown | unknown | 0– 200ppm | unknown | unknown |
| $H_2S$+COS | 20ppm | - | - | - | - | 1.6% | 1.35% |
| Others($N_2$) | 8.6% | - | - | 6.1% | - | 0.2% | - |
| $CO/H_2$ mole ratio | 2.2 | 2.2– 2.3 | 1.8 | 1.4 | 1.0–1.8 | 1.2 | 1.0 |
| HHV[MJ/m³] | 13.0 | 12.2–12.3 | 11.3–12.6 | 10.6 | 5.2–6.6 | 12.1 | 11.0 |

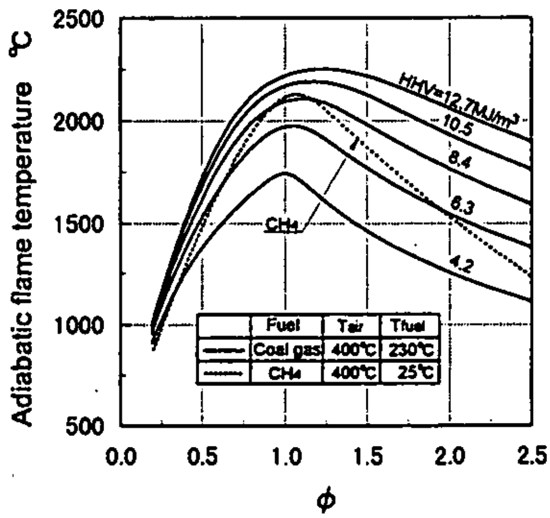


Fig.2 Relationship between equivalence ratio and adiabatic flame temperature for coal gas and $CH_4$

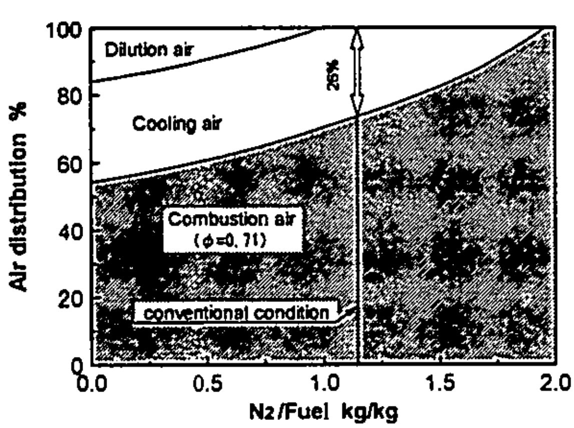


Fig.3 Air distribution design of a medium–Btu gaseous fueled gas turbine combustor with nitrogen injection.

varied widely (5.2-13.0MJ/m³), from about one-eighth to one-third of natural gas, with raw materials and gasifier types. For example, a gaseous fuel derived from biomass contained 30-40 percent steam in the gaseous fuel.

Figure 2 shows the adiabatic flame temperature of fuels which were: 1) medium-Btu fuels, with fuel calorific values (HHV) of 12.7, 10.5, 8.4, 6.3MJ/m³ without nitrogen; 2) medium-Btu fuel blended with surplus nitrogen, or low-Btu fuel of 4.2MJ/m³ (HHV); and 3) natural gas. Calculations of flame temperature were done with a CO-$H_2$ mixture (CO/$H_2$ molar ratio of 1.4:1) under any condition, and the fuel calorific value was adjusted with nitrogen. When the fuel calorific value was 8.4MJ/m³ or higher, the maximum flame temperature of the medium-Btu fuel without nitrogen was about 400K higher than that of the nitrogen-blended fuel. That is, the flame temperature of medium-Btu gasified fuel, produced in an oxygen-blown gasifier, was higher than high-calorie gases such as methane, while the medium-Btu fuel had a calorific value as low as one fifth of methane. NOx emission was expected to increase more when burning medium-Btu fuel than with nitrogen-blended fuel, or low-Btu fuel (Hasegawa and Sato, 1996). We intended to inject surplus nitrogen directly into higher temperature regions of the burner and to effectively decrease both NOx emissions produced from these regions, and decrease liner wall temperature, simultaneously.

## TESTING THE COMBUSTOR

### Subjects of the Medium-Btu Fueled Combustor

As mentioned above, if the surplus nitrogen was premixed with fuel produced from the oxygen-blown gasifier, the fuel calorific value decreased as low as a low-Btu fuel derived from an air-blown gasifier. Therefore, the combustion air of medium-Btu gaseous fuel for any combustion temperature decreased as the flow rate of nitrogen injection into the combustor increased. Figure 3 shows the relation between the nitrogen injection flow rate into the gas turbine combustor and air distribution using medium-Btu coal gaseous fuel. The average gas temperature of the combustor outlet was set at 1400 ℃. To calculate air distribution, the overall amount of air was assumed

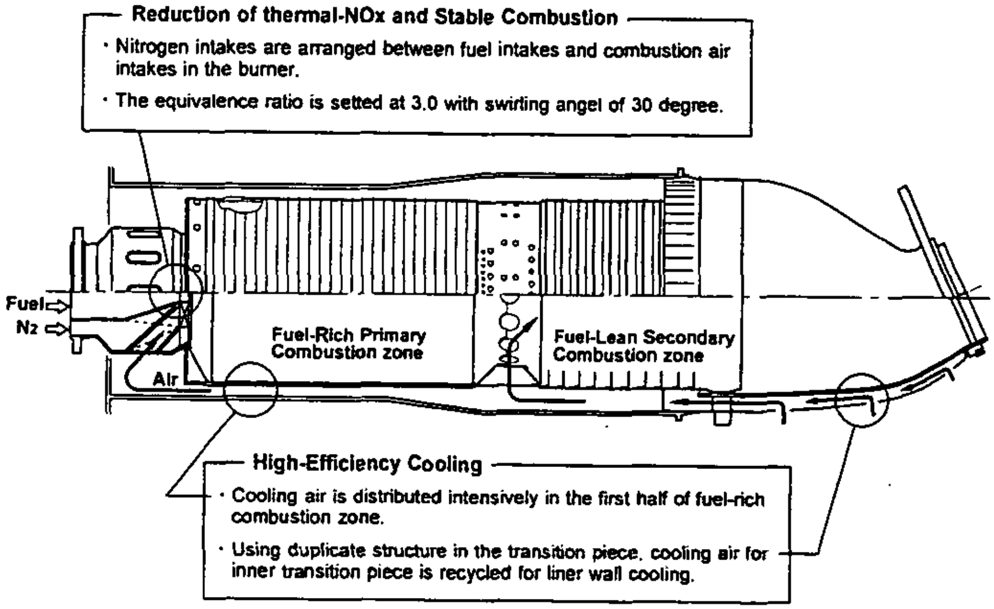


**Fig.4 Design concept of a medium-Btu fueled gas turbine combustion**

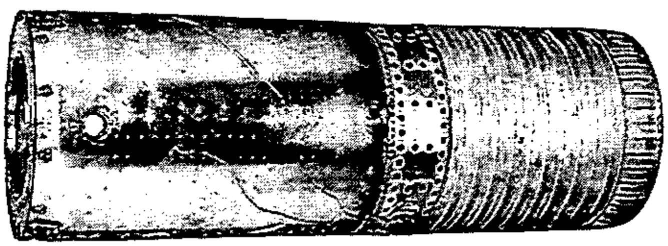

**Fig.5 Tested 1300 ℃ -class combustion liner**

to be 100 percent. The amount of air for combustion was first calculated at 1.4 times theoretical air ( $\phi$ = 0.71). 30 percent of the total air was considered cooling air for the combustion liner wall, and the remainder as diluting air. According to figure 3, as the ratio of nitrogen flow over fuel flow ($N_2$/fuel) increases to as much as 1.0, the ratio of cooling air and diluting air decreases significantly, and flexibility of combustor design is minimized. Since the $N_2$/fuel was 1.15 in the system considered in the present study, the ratio of cooling and diluting air to total air was as low as 26 percent. To summarize these characteristics, it can be said that the design of a gas turbine combustor, utilizing direct injection of nitrogen into a combustor, should consider the following issues for an oxygen-blown IGCC:

(1)Low NOx-emission technology. It is necessary to restrain NOx production from nitrogen fixation using nitrogen injection into the combustor, not by blending nitrogen with gaseous fuel.

(2)Combustion stability. It is necessary to stabilize the flame of medium-Btu fuel with nitrogen injection.

(3)Cooling technology. It is necessary to maintain the combustor wall under a heat resistant temperature with less amount of air.

Figure 4 presents characteristics of the designed, medium-Btu fueled 1300 ℃ -class combustor based on the above considerations. Figure 5 illustrates the external view of the combustion liner. The main design concept for the tested combustor in the present study was to secure stable combustion of medium-Btu fuel with nitrogen injection in a wide range of turn-down operations, low NOx emissions and enough cooling-air for the combustion liner. The tested burner was developed to achieve low-NOx emissions from nitrogen fixation and stable combustion. A combustion liner, which had been developed to achieve a stable combustion for a low-Btu fueled 1500 ℃ -class gas turbine combustor through proper design the combustor (Nakata et al., 1994), was used for the following tests. We believe further improvement will be needed to design a 1300 ℃ -class gas turbine combustor for medium-Btu fuel. The overall length of the combustor is 1093mm and the inside diameter is 356mm.

## Reduction of NOx and CO Emission, and Stable Combustion

To restrict thermal NOx production originating from nitrogen fixation and CO emissions, we designed the burner with nitrogen injection, based on combustion tests previously conducted using a small diffusion burner (Hasegawa et al., 1996).

Figure 6 presents an example of the test results which indicate the influence of the primary-equivalence ratio on NOx emission characteristics in two-staged combustion for comparing three cases: 1) a fuel calorific value (HHV) of 12.7MJ/m$^3$, without nitrogen injection; 2) a fuel calorific value of 12.7MJ/m$^3$, where nitrogen is blended with the combustion air from the burner; 3) a fuel blended with nitrogen of the same quantity as case 2), or low-Btu fuel of 4.2MJ/m$^3$. From figure 6, we know that nitrogen blended with fuel or air injected from the burner has a great influence over decreasing NOx emissions from nitrogen fixation. That is, in the case of nitrogen-injection into the combustor, the only combustion method which is fuel-rich in the primary-combustion zone and fuel-lean in the secondary-combustion zone effectively decreases NOx emissions, while the rich-lean combustion and the lean-lean combustion were effective in decreasing NOx emissions in the case of no nitrogen-injection. Furthermore, the effect of nitrogen blended with fuel is slight-

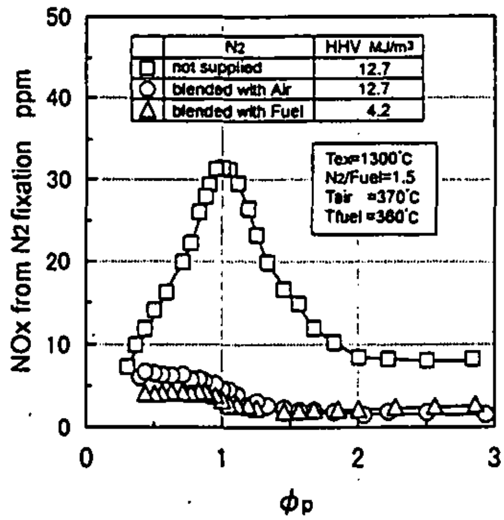


**Fig.6 Effect of nitrogen injection on NOx emission characteristics in two-staged combustion, using a small diffusion burner**

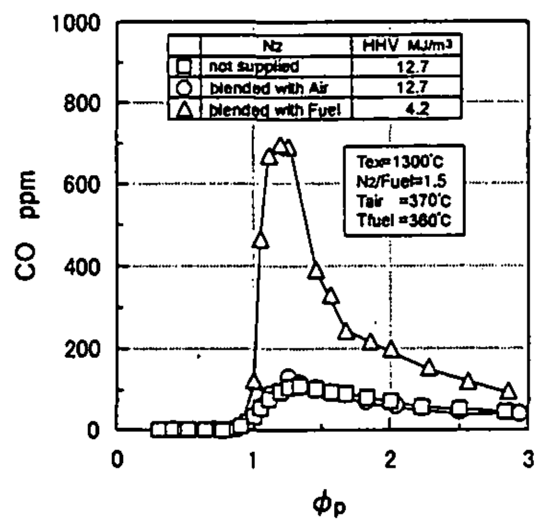


**Fig.7 Effect of nitrogen injection on CO emission characteristics in two-staged combustion, using a small diffusion burner**

ly higher than that of nitrogen blended with air. As a result, we found that NOx emissions significantly decreased in the medium-Btu fuel, both by injecting nitrogen from the burner and by setting the primary-equivalence ratio over 1.4 under rated load condtions.

Figure 7 shows the CO emission characteristics under the same conditions as figure 6. According to CO emissions, we know that CO emission concentration reached a maximum value at the primary-equivalence ratio of 1.2 and decreased as the primary-equivalence ratio increased, in every case. Furthermore, in the case where nitrogen-blended with air was injected into the combustor, CO emissions decreased as low as medium-Btu gaseous fuel not blended with nitrogen, while CO emissions significantly increased when fuel was blended with nitrogen. That is, all of the surplus nitrogen should be injected into the combustor from the burner to reduce the high temperature region and should not be blended with fuel for stable

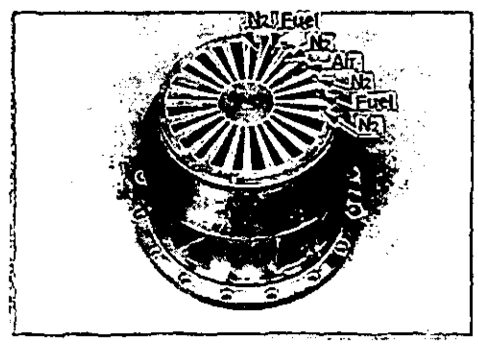


**Fig.8 Tested burner for the 1300 ℃ -class combustor**

combustion in a wide range of turn-down operations.

Figure 8 illustrates the external view of the designed burner. Based on these results, we arranged the nitrogen injection intakes between fuel and air intakes to surround each air intake with nitrogen intakes. The nitrogen injected directly into a combustor has the effect of decreasing power to compress nitrogen higher than the pressure of fuel or air, which is needed for even blending. In this way, from the system analysis, the thermal efficiency of the plant improved by 0.2 percent (absolute), compared with a case where nitrogen was premixed with coal gasified fuel before injection into the combustor. Furthermore, it is possible to controll the mixing of fuel, air, and nitrogen positively by way of nitrogen being injected separately into the combustor. The fuel, the combustion air from the burner and the nitrogen are separately injected into the combustor through a swirler (which has a 30-degree swirl angle and a 15-degree introvert angle), to collide medium-Btu fuel with air in an atmosphere where nitrogen is superior in amount to both fuel and air. This new technique causes a decrease in flame temperature in the primary combustion zone which produces NOx from nitrogen fixation, NOx production and liner wall temperature near the burner are controlled, just as in the case of fuel blended with nitrogen. The primary-equivalence ratio was set at 1.4 for a low NOx combustion based on the results of combustion tests using a diffusion burner. By setting the equivalence ratio of the burner outlet at 3.0 under the rated-load condition, a stable flame can be maintained in a wide range of turn-down operations. As indicated in the following section, the burner is especially effective for maintaining flame under partial load conditions.

## Cooling of the Combustion Liner Wall

In order to compensate for a declined cooling-air ratio associated with a surplus nitrogen injection into a gas turbine combustor, the tested combustor is equipped with a dual-structure transition piece so that the cooling air in the transition piece can be recycled to cool down the combustor liner wall. The cooling air flowing into the transition piece from the exterior wall cools the interior wall by an impingement method, and moves to the combustion liner on the upstream side. For the primary-combustion zone where temperatures are expected to be especially high, the layer-built cooling structure combining impingement cooling and film-cooling, was employed. For the secondary-combustion zone, the film-cooling method was used.

Table 2 shows the standard properties of the tested fuel. As for tests, the higher heating value of the tested fuel was set at 8.6MJ/m $^3$, a $(CO+CH_4)/H_2$ molar ratio at 1.2. The flow rate of surplus nitrogen produced from the air-separation unit was about 1.2 times the fuel flow. Rated load conditions in the combustion tests are summarized in Table 3. The loading rate in the combustor at the design point is $3.6 \times 10^2$ W/(m $^3$ ·Pa).

## TEST FACILITIES AND TEST METHOD

### Test Facilities

The schematic diagram of the test facilities is shown in figure 9. The raw fuel obtained by mixing $CO_2$ and steam with gaseous propane was decomposed to CO and $H_2$ inside the fuel reforming device. A hydrogen separation membrane was used to adjust the $CO/H_2$ molar ratio. $N_2$ was added to adjust the fuel calorific value to the given calorie, then coal derived simulated gases were produced.

**Table 2 Typical composition of the tested fuel**

| | | |
|---|---|---|
| Composition | CO | 32.6 % |
| | $H_2$ | 28.7 % |
| | $CH_4$ | 2.0 % |
| | $CO_2$ | 31.0 % |
| | $H_2O$ | - |
| | $N_2$ | 5.7 % |
| HHV | 8.6 MJ/m$^3$ (2050 kcal/m$^3$ ) | |
| LHV | 7.7 MJ/m$^3$ (1840 kcal/m$^3$ ) | |

**Table 3 Standard test condition**

| | | |
|---|---|---|
| Tair | : | 370 ℃ |
| Tfuel | : | 260 ℃ |
| $T_{N_2}$ | : | 60 ℃ |
| $N_2$/Fuel | : | 1.15 kg/kg |
| Tex | : | 1400 ℃ |
| P | : | 0.1 MPa |
| $\phi$ ex | : | 0.50 |
| Lb | : | $3.6\times10^2$ W/(m$^3$·Pa) |

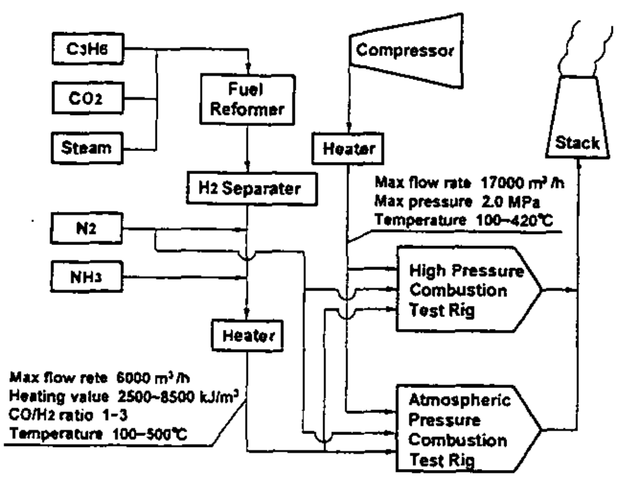


**Fig.9 Schematic diagram of experimental facility**

This facility had one more nitrogen supply line, by which nitrogen was directly injected into the combustor. Air provided to the combustor was pressurized to 2.0MPa by using a four-stage centrifugal compressor. Both fuel and air were supplied to the gas turbine combustor after being heated separately with a preheater to a given temperature.

The combustion testing area had two test rigs, each of which were capable of performing full-scale atmospheric pressure combustion tests as well as half-scale high-pressure combustion tests for a 150MW-class multican type combustor. Figure 10 shows a cross-sectional view of the combustor test rig under atmospheric pressure conditions. After passing through the transition piece, the exhaust gas from the combustor was introduced into the measuring section where gas components and temperatures were measured. The components of the combustion gas were analyzed by an automatic gas analyzer. After that, the gas temperature was lowered through a quenching pot, using a water spray injection system.

Combustion tests were conducted on a full-scale combustor under atmospheric pressure condition.

### Measurement System

Sample gases were taken from the exit of the combustor through a water-cooled stainless steel probe located on the center-line of a height-wise cross section of the measuring duct. The sample lines of stainless steel were thermally insulated with heat tape to maintain the sampling system above the dew point of the exhaust gas. The gas samples taken were from an average of three points on the center-line of the measuring duct and continuously introduced into an emission console which measured: CO and $CO_2$ by infrared analyses, NO and NOx by chemiluminescence analyses, $O_2$ by paramagnetic analysis, and hydrocarbons by flame ionization. The medium-Btu simulated fuel were sampled from the fuel gas supply line at the inlet of combustor and CO, $H_2$, $CH_4$, $H_2O$, $CO_2$ and $N_2$ were determined by gas chromatography. Heating values of the simu-

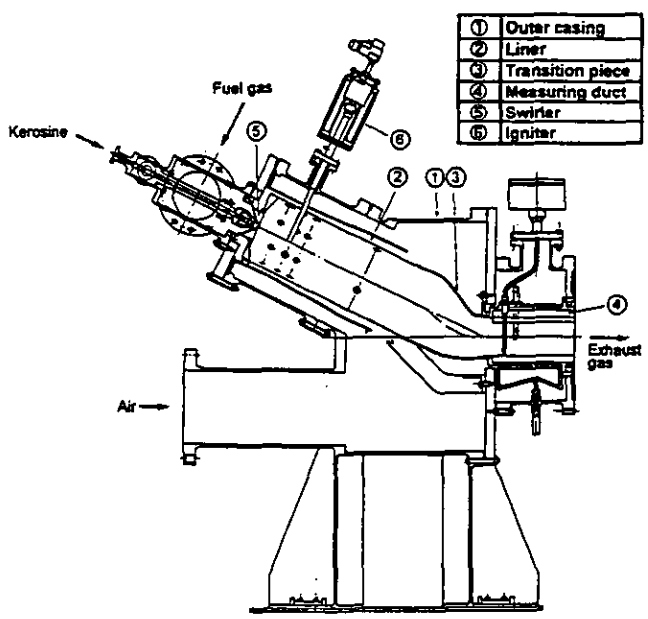


**Fig.10 Atmospheferic pressure combustion test rig**

lated gaseous fuel were monitored by a calorimeter and calculated from analytical data of gas components obtained from gas chromatography.

The temperatures of the combustor liner wall were measured by 100 sheathed type-K thermocouples with a diameter of 1mm welded on the liner wall. The temperature distributions of the combustor exit gas were measured by traversing an array of five type-R thermocouples.

## TEST RESULTS AND DISCUSSION

### Thermal Characteristics of the Combustor Liner Wall

Figure 11 shows the temperature distributions of the combustor liner wall, when the outlet gas temperature of the combustor changes. In the combustion tests for changing the combustor-outlet gas temperature, the air flow rate was maintained at a constant value of 1.09m$^3$/s and the fuel flow rate was changed. At any outlet gas temperature, the overall liner-wall temperature remained almost always under 850 ℃, the allowable heat resistant temperature. Furthermore, the liner wall temperature in the primary combustion zone significantly decreased by way of directly injected nitrogen compared with nitrogen-blended fuel (not indicated in figure 11). This reveals that, nitrogen injection effectively lowers the flame temperature in the primary combustion zone, where a high temperature region is expected, while the amount of air for the convection cooling on the combustor wall decreases with the increase of nitrogen injection. It is thus believed that nitrogen injection from the burner has the effect equal to converting cooling air partially for the combustor liner wall.

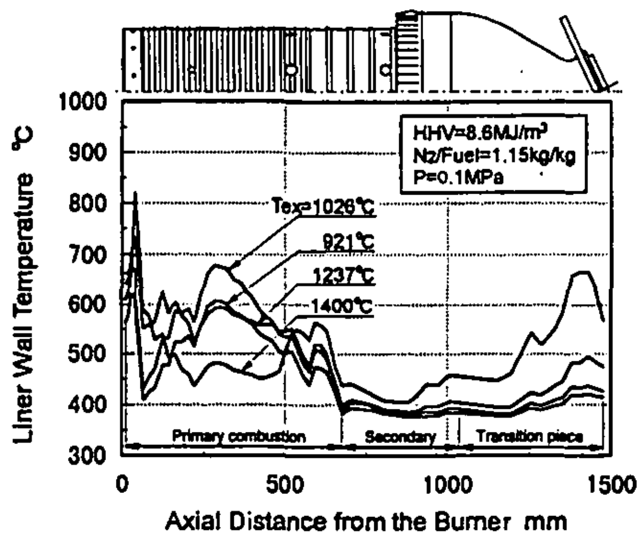


Fig.11 Combustor wall temperature distribution using the average gas temperature of combustor outlet as a parameter

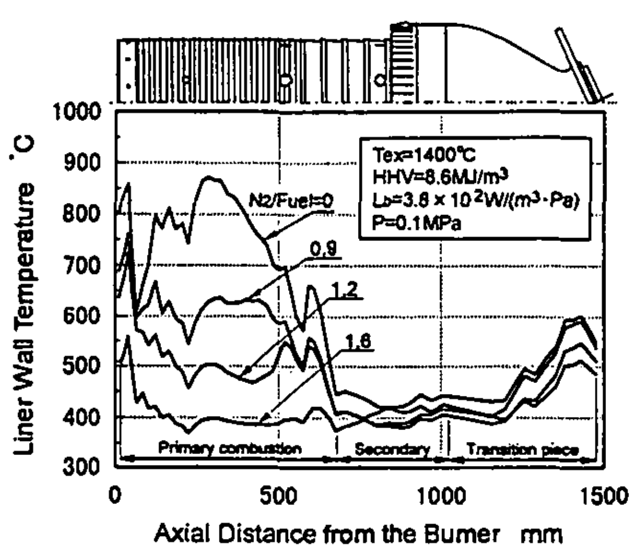


Fig.12 Effect of the nitrogen injection flow rate from the burner on the combustor wall temperature distribution.

Figure 12 shows the effects of nitrogen injection into the primary combustion zone. Based on the overall temperature, we can see that the combustion gas temperature in the primary combustion zone is sharply decreased when nitrogen, injected between fuel and air from the swirler into the combustor, is increased. From the test results, this technique which injects nitrogen directly into the high temperature region in the primary combustor results in a significant reduction in liner wall temperatures near the burner and in NOx production from nitrogen fixation. That is, in the case of nitrogen injection (which is 1.2 times the fuel, from the burner into the high temperature region), the overall temperature in the primary-combustion zone declines by 350 ℃. The adiabatic flame temperature of the nitrogen-blended fuel (HHV=4.2MJ/m$^3$) is about 500 ℃ lower than that of the medium-Btu fuel (HHV=8.4MJ/m$^3$) in the rich-side of the stoichiometric. Because both a decrease of the flame temperature and a lowering of the emissibity of the combustion gas is produced through nitrogen-injection in the primary-combustion zone, the liner wall temperature in the primary-combustiom zone sharply decreases.

The next stages of development for the prototype combustor, under current progress, will address the issue of increasing flame temperature in the primary combustion zone to levels adequate for maintaining a stable combustion in the partial loads. As a target, the

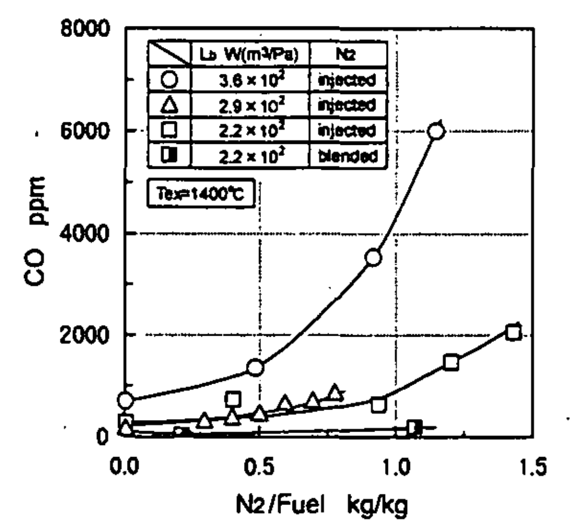


Fig.13 Effect of the loading rate in the combustor on CO emission characteristics

temperature of the primary combustion zone should maintain an adequate level between 600 and 850 ℃ for any load, through better utilization of nitrogen injection from the burner, introducing the remaining nitrogen (not used in the burner) as cooling air for the combustion liner, and as cooling air for the primary combustor.

## CO Emission Characteristics

Nitrogen injection into the combustor resulted in the rise of the equivalence ratio by which the outlet gas temperature of the combustor was adjusted to 1400 ℃. We observed the emission characteristics of CO when the nitrogen injection flow rate to the combustor changed. Figure 13 shows the effect of the loading rate on the relationship between nitrogen injection and CO emission characteristics, where nitrogen is directly injected into the primary combustion zone from the burner and where the combustor-outlet gas temperature is 1400 ℃. To compare the technique of nitrogen direct injection with conventional techniques, this figure also shows the case where nitrogen premixed with fuel is injected under a condition of 60 percent rated load condition. In every case, CO emissions increased sharply as the flow rate of nitrogen injected into the combustor increased. This is because the equivalence ratio, which adjusted the combustor-outlet gas temperature to 1400 ℃, was raised as nitrogen injection increased. Under the 60 percent rated load condition, or a loading rate of $2.2 \times 10^2$ W/($m^3$·Pa), CO emission was higher where nitrogen was directly injected than where nitrogen was blended with the fuel. As a result of injecting nitrogen between the fuel intakes and air intakes at the burner, the oxidation reaction of CO was restrained while simultaneously decreasing the flame temperature in the primary combustion zone. This was also realized from results which indicated the effect of nitrogen injection on liner wall temperature distribution, shown in figure10. Furthermore, for any nitrogen injection flow rate, CO emissions increased sharply as the loading rate in the combustor increased, or when CO emissions reached 6000ppm under the rated load condition. For the next stage of development of the prototype combustor, we will address the issue of decreasing CO emissions to an adequate level, by better utilizing air distribution in the combustor.

When the equivalence ratio rises, CO emissions tend to increase and combustion efficiency declines. Figure 14 shows the effect of equivalence ratio at the combustor-outlet on CO emission characteristics, both in the case where nitrogen with a $N_2$/fuel ratio of 1.15 is directly injected into the primary combustion zone from the burner, and where nitrogen is not injected under atmospheric pressure conditions. To estimate CO emissions under high pressure conditions, this figure also shows the result of high pressure combustion test (Hasegawa et. al., 1997) using the low-Btu fuel, just as with the nitrogen-blended fuel, under the loading rate of 2.2W/($m^3$·Pa). In tests, the higher heating value of fuel is set at a constant 8.6MJ/$m^3$. In each case, CO emission characteristics showed the same tendencies; CO emission concentration increased sharply as the equivalence ratio of the combustor-outlet rose. This was because nitrogen injected into the primary combustion zone increased the equivalence ratio which adjusted the combustor-outlet gas temperature of 1400 ℃ and increased CO emissions. That is, we were concerned that CO emission characteristics were dominated by the equivalence ratio of combustor-outlet regardless of nitrogen injection.

Furthermore, we know from high pressure combustion tests (Hasegawa et.al., 1997) that CO emission concentration tends to decline significantly (up to a few tenth of ppm at the operating pressure of the gas turbine). while CO emitted about 3000ppm at the atmospheric pressure. That is to say that CO emissions decreased up to one-hundredth that of the atmospheric pressure condition. Taking the above charteristics in consideration, the increase in CO emissions is attributable to a delayed decomposition of CO, due to excessive-fuel concentration in the combustor. However, under high-pressure conditions, combustion efficiency improve with an increase in pressure. Thus, under real application conditions of the

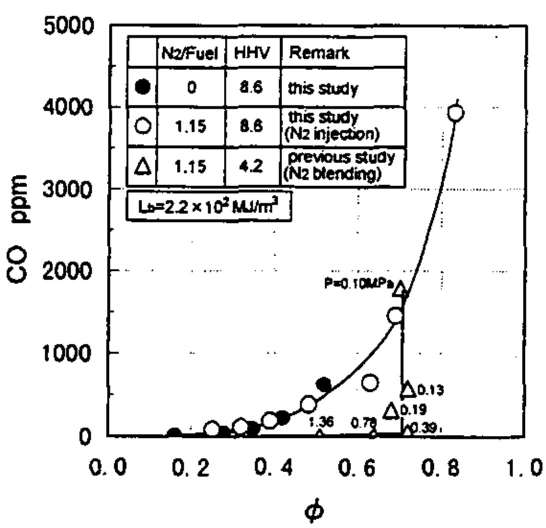


**Fig.14 Effect of the nitrogen injection on CO emission characteristics**

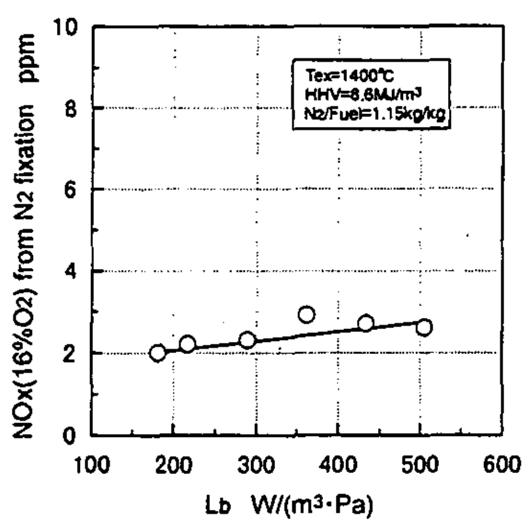


**Fig.15 Effect of the combustor loading rate on NOx emission concentration from nitrogen fixation**

medium-Btu fueled gas turbine, we can expect that CO emissions will be reduced to less than 100 ppm by the pressure effect.

## NOx Formation Characteristics

A more practical and efficient method of returning nitrogen to the cycle in oxygen-blown IGCC applications, other than reinjecting it into the syngas fuel stream or the combustion air stream, is to directly inject it into the combustor from the burner for NOx control. Figure 15 shows the effect of the combustor loading rate on NOx emission characteristics from a nitrogen fixation, where the condition where the combustor-outlet gas temperature was 1400 ℃. The vertical axis represents the NOx emissions which are produced from a nitrogen fixation with a correction of 16 percent $O_2$ when its coal gaseous fuel does not contain $NH_3$. NOx emission concentration slightly increased from 2.0ppm to 2.5ppm when the loading rate of the combustor increased from $2.0 \times 10^2$ W/(m$^3$·Pa) to $5.0 \times 10^2$ W/(m$^3$·Pa). We consider the loading rate as hardly affecting NOx emissions from the nitrogen fixation, while the loading rate has a great influence on CO emissions.

Figure 16 shows the relationship between the combustor-outlet gas temperature and NOx emission characteristics from a nitrogen fixation, using the $N_2$/fuel ratio as a parameter. This figure also shows the case where nitrogen blended with fuel is injected under a 60 percent rated load condition, or $L_b$=$2.2 \times 10^2$ W/(m$^3$·Pa). When the injected nitrogen flow rate is changed, the combustor-outlet gas temperature is set at a constant and the equivalence ratio at the combustor-outlet is adjusted. At any temperature of the combustor-outlet, NOx emissions decreased sharply as nitrogen increased. When the flow ratio of nitrogen injection over fuel ($N_2$/fuel) increased above 1.15, NOx emissions decreased as low as the case where nitrogen ($N_2$/fuel=1.15) was blended with fuel, or NOx emissions were under 4ppm at any combustor-outlet temperature. When the combustor-outlet temperature was about 1100 ℃ in the case of no injection of nitrogen, NOx emissions reached maximum value, because the equivalence ratio in the primary-combustion zone was stoichiometric. From the test results that NOx emission had no peak concentration at any partial load condition in each case of nitrogen injection. Nitrogen injection effectively lowered the flame temperature in the primary combustion zone in which a high temperature region was expected, as mentioned in Figure12.

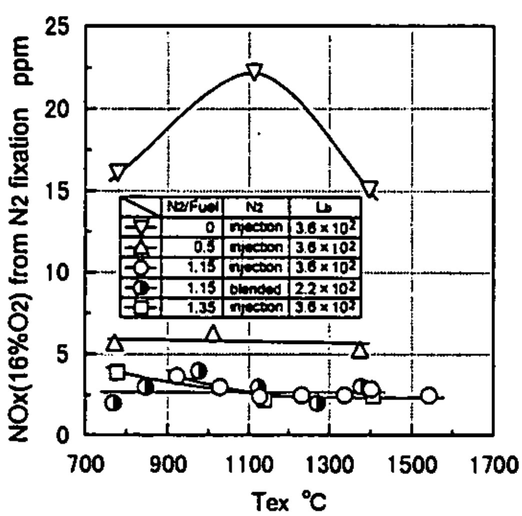


**Fig.16 Effect of the combustor outlet gas temperature on NOx emission concentration from nitrogen fixation**

Prompt NO formation does not necessarily need to be taken into account in our study, because the coal gaseous fuel contains only a small amount of $CH_4$. On the other hand, as we mentioned in Figure 2, the flame temperature of medium-Btu gaseous fuel is as high as that of hydrocarbon flame. That is, most of the NOx originated from the production process of Zeldovich NO. From the view point of NOx formation, NOx emission is affected by both the flame temperature and the super-equilibrium O-atom. By injecting nitrogen into the high temperature zone, the super equilibrium O-atom decreases significantly and NOx emission decreases.

From these results, it appears that direct injection of nitrogen will sufficiently allow low NOx levels and the IGCC system will be able to achieve the full performance benefits of this concept.

## CONCLUSION

Based on the combustion tests results using a small diffusion burner, we have designed a suitable nitrogen injection system for an oxygen-blown IGCC, constructed the burner, and tested its performance under atmospheric pressurized conditions. Effects and subjects of a nitrogen direct injection method are summarized as follows:

(1)Nitrogen injection has the effect of both decreasing flame temperature near the burner in which a higher temperature zone is expected, and decreasing liner wall temperature in the primary-combustion zone.

(2)The direct nitrogen injection technique has the effect of decreasing NOx emissions from nitrogen fixation, NOx emission decreases as low as cases where nitrogen is blended with fuel, or 4ppm or below (corrected at 16 percent $O_2$) for any load condition.

(3)CO emissions increased by an equivalence ratio of combustor-outlet, regardless of nitrogen injection, or reached 6000ppm at the rated load condition. The next stage of development of the prototype combustor, currently in progress, will address the issue of decreasing CO emissions to adequate levels, by better utilizing air distribution in the combustor.

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