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Development of Gas Turbine Combustor for Each Gasified Fuel and prospect of high-efficiency generation of various resources

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ABSTRACT

Japan depends on imports for most energy resources. To obtain stable supplies of energy and protect the global environment, not only high-efficiency use of existing fossil-based power generation but unused resources' reexamination, waste material utilization, and highly effective use of such resources will be important. Discoverable reserves of bituminous raw materials are several times larger than that of crude (figure 1). In Japan, power-generation infrastructures are equipped in incineration of around 60% of the waste and the thermal-efficiency is only around 10% on the average. When introducing the high-efficiency technologies into waste incinerators, its electricity corresponds to around 4% power demand of the electric power industry or one second of hydroelectric generation. Developments of integrated gasification combined cycle (IGCC) continue worldwide, and such technologies enable high-efficiency generation from various quality resources. This paper reviews the trends of IGCCs' developments worldwide and outlines combustion technologies of the high temperature gas turbine for IGCC in Japan.

INTRODUCTION

In response to recent changes in energy-intensive and global environmental conditions, it is urgent and crucial concern to develop the energy saving technologies and high-efficiency technologies of unused resources and wastes for human recycling-oriented society.

Figure 2 shows time series of the world net electricity generation by region from year1980 to 2004 (Official Energy Statistics from the U.S. Government, World Electricity Data, 2006). In the intervening quarter-century, the amount of electricity generation in Asia/Oceania increased significantly or about four times, while world electricity generation was double. Figure 3 shows electricity transition in Asia/Oceania with respect to each power generation method (Official Energy Statistics from the U.S. Government, World Electricity Data, 2006). Of those, the conventional thermal electricity generation covers 78 percent of the demand increase, the hydroelectricity and the nuclear about 11% with each other. Our lives are founded on the amount of fossil fuels. Just like as Japan, we depend on imported resources for most of its primary energy needs or are greatly influenced.

When citing Japan as a typical example, the primary energy resources of coal, liquefied natural gas (LNG) and oil for thermal power plants those supply 60 percent of all public power demand in fiscal 2004. In response to the oil crises of the 1970s, the Japanese government and electric companies shift from crude oil unevenly distributed in the Middle East to resources of coal and LNG which is globally abundant and expected to be secured stable supply during the future shown in figure 4. In recent years, the developing world grows demand for energy resources of coal, oil and LNG, and

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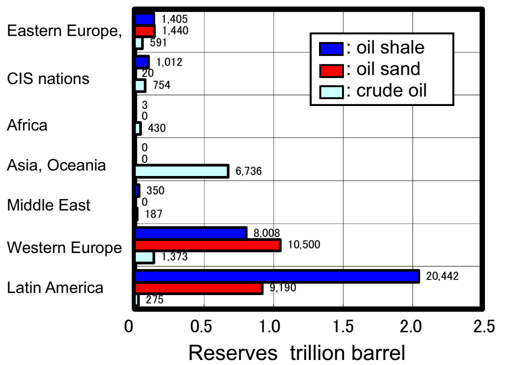


Fig.1 Discoverable reserves of crude oil, oil sand and oil shale (Petroleum Association of Japan, Home Page)

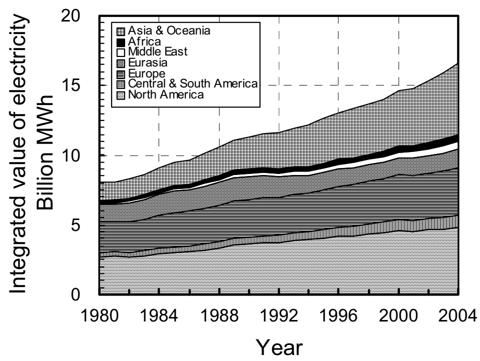


Fig.2 Time series of world net electricity generation by region

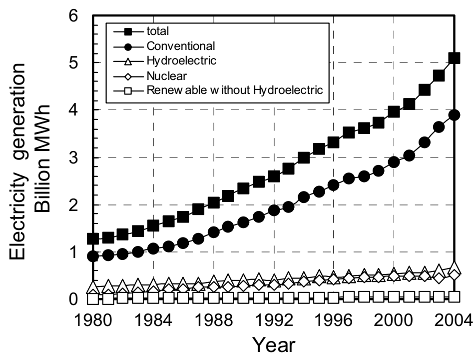
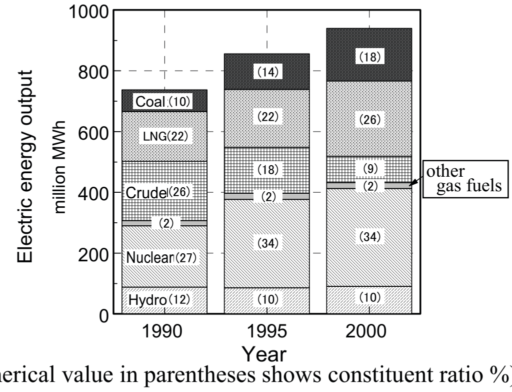


Fig.3 Time series of electricity generation with respect to each generation method in Asia and Oceania



(Numerical value in parentheses shows constituent ratio %)

Fig.4 Transition of composition ratio for electric power generation in Japan

international competition for development of energy sources of oil field and gas field in the world is ever intensified. From the view points of secure primary energy, reclamation of new energy resources and development of high-efficiency utilization technology become increasingly important in the world. From these reasons, each industry of energy resource, power generation and global environmental protection deal with power generation technologies of gasification combined cycle for each raw material (IGCC : Integrated gasified combined cycle) and gasification melting furnace, as part of each technological development, the high temperature gas turbine technology have been developed. Gasification technologies enable highly effective usages of unused resources of low-rank coal, tar crude, and oil shale which are equivalent to several times of the oil proved reserve. Moreover unused resources such as biomass fuel of the greenhouse gas free energy and wastes could be expected to use in IGCCs.

The typical compositions of gaseous fuels produced in air-blown and oxygen-blown gasifiers, and in blast furnaces are shown in Table 1 (Ashizawa et al., 1996),(Bush et al., 1991),(Consonni et al., 1997) ,(Cook et al., 1994),(Hasegawa, 2006a and 2006b),(Ichikawa and Araki, 1996),(Kalsall et al., 1994),(Modern Power Systems Review, 1993),(Modern Power Systems, 1994),(Ueda et al., 1995). Each type of gaseous fuel contained CO and H_2 as the main combustible components, and small amounts of CH_4, and its property is chiefly characterized by the gasification method and raw material. Fuel calorific values varied widely (4.2-13.0MJ/m^3), from about one tenth to one third of that of natural gas, depending upon the raw material, the gasification methods and so on. It is necessary to adopt the suitable combustion technology for each gaseous fuel.

In this paper, I examined influence of fuel composition on reducing combustion characteristics of fuel-bound nitrogen, called 'fuel-N' for short, in carbon monoxide (CO), hydrogen (H_2) and methane (CH_4) mixture fuel such as each gasified fuel or blast furnace gas (BFG) fuel for the high-efficiency gas turbine, and clarified its mechanism and the optimization technologies.

NOMENCLATURE

CO/H_2 : molar ratio of carbon monoxide to hydrogen in the fuel
C.R. : conversion rate from ammonia to NOx %
HHV : higher heating value of the fuel at 273K basis MJ/m^3
LHV : lower heating value of the fuel at 273K basis MJ/m^3
Ic : combustion intensity at 273K basis W/(m^3 ·Pa)
N_2/Fuel: nitrogen over fuel supply ratio kg/kg
NOx(16%O_2) : NOx emissions corrected at 16% oxygen in the exhaust gas ppm
NOx_{th} : thermal NOx emissions
T_{ad} : adiabatic flame temperature K
T_{air} : temperature of supplied air K
T_{ex} : average temperature of combustor exit gas K
T_{fuel} : temperature of supplied fuel K
T_{N_2} : temperature of supplied nitrogen K
Ur : mean velocity of cross-sectional flow of air at 273K basis m/s
ϕ_{ex} : equivalence ratio at combustor exit
ϕ_p : equivalence ratio in the primary combustion zone
$\Delta P/q$: total pressure loss coefficient (characteristics section is combustor-exit)

BACKGROUND OF IGCC DEVELOPMENT IN THE WORLD

Figure 5 examples the outline of the typical oxygen-blown IGCC system. In this system, the raw materials such as coal and crude are fed into the gasifier by slurry feed or dry feed with nitrogen; the synthetic gas is cleaned by usages of dust removing and desulfurizing process; the clean synthetic gas is fed into the high-efficiency gas turbine of topping cycle; and the steam cycle is equipped to recover heat from the gas turbine exhaust. The above IGCC system is approximately similar to the LNG fired gas turbine combined generation except for including a gasification and a syn-

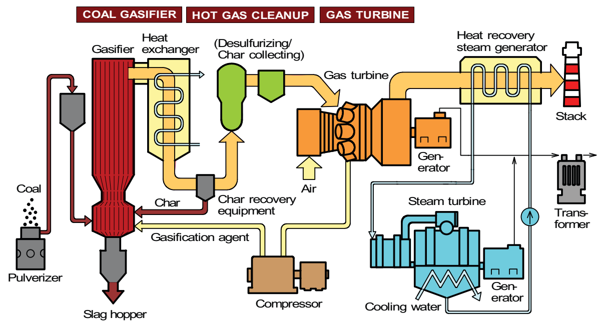
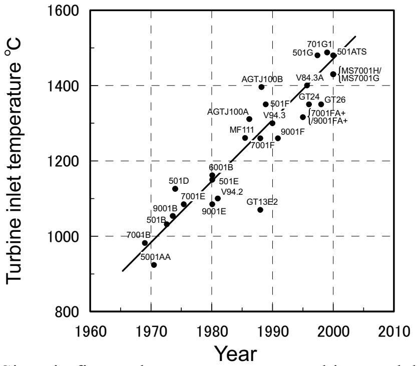


Fig.5 Schematic diagram of typical IGCC system

Table 1 Typical compositions of derived gases from the gasifiers and furnaces

Fuel	BFG	COG	Gasified fuel											
Resource			Waste*	RDF	Coal						Biomass		Heavy residue	Orimulsion™
Gasifier type			Fluidized		Fixed	Fluidized	Entrained				Entrained		Entrained	Entrained
Coal supply					Dry	Dry	Dry			Slurry				
Developer					BGL	BC	IGC	Shell	HYCOL	Texaco	Tampella		Texaco	CRIEPI
Oxydizer			Air	Air+O_2	O_2	Air	Air	O_2	O_2	O_2	Air	O_2	O_2	O_2
Composition														
CO[%]	20	6	6	30	56.4	7.9-14.7	25.9-27.6	65.2-69.5	55.2-59.4	40.9	8.0-15.0	21.9-23.1	51.7	43.5
H_2 [%]	3	56	1.6	22	25.6	13.2-15.0	10.9- 9.4	28.8-31.0	31.1-33.7	29.9	8.0-12.0	12.5-22.4	43.1	42.2
CH_4[%]	-	30	0.9	0.4	6.6	1.5-2.8	1.4- 0.5	0.01-0.03	1.0- 2.0	0.1	4.0- 8.0	2.2	0.2	0.4
CO_2[%]	20	-	12.4	4.1	2.8	10.0-12.0	6.7- 5.4	1.0-2.8	7.6-10.4	9.5	13.0-18.0	20.7-18.6	3.2	11.8
H_2O [%]	-	-	23.4	5.9	-(a)	11.5-18.4	-(a)	-(c)	-(a)	12.3	7.0-15.0	40.9-31.5	-(c)	-(c)
NH_3 [ppm]	-	-	-	-	-(a)	500-1000	1000(b)	100-600	-(a)	-(a)	-(a)	0-200	-(a)	-(a)
H_2S+COS[%]	-	-	-	-	20ppm	-(a)	404-714ppm	0.14-1.1	-(a)	-(a)	-(a)	0.285-1.132	1.6	1.35
Others[%]	N_2	C_2H_2etc.	N_2	C_2H_4etc.	8.6	45.9-47.3	54.2-56.1	-(a)	-(a)	7.3	-(a)	1.800-1.048	0.2	0.75
CO/H_2 mole ratio	7	0.1	3.8	1.4	2.2	-(a)	2.4- 3.0	2.1- 2.4	1.6-1.9(b)	1.4	-(a)	1.0- 1.8	1.2	1.0
HHV[MJ/m^3]	2.9	20	1.8	6.8	13.0	3.6- 4.1	4.9- 5.2	12.2-12.5	12.0(b)	9.0	4.0-7.0	5.2- 6.6	12.1	11.0

BFG:Blast furnace gas, COG:Coke-oven gas, *:Municipal solid waste, RDF:Refuse derived fuel, (a):No description, (b):Estimated values, (c):Dry Base, HHV:Higher heating value



(Signs in figure show names of gas turbine models)

Fig.6 Transition of turbine inlet temperature of gas turbine combustor for electric power generation

thetic gas cleanup process. IGCC requires a little higher station service power than the case of LNG gas turbine power generation.

Concerning innovation of gas turbine combined cycle power generation, the improvements of the thermal efficiency of the plants have been conducted by enhancing turbine inlet temperature so far. It is attempted that the turbine inlet temperature has been rose at the rate of about 20℃(20K) per year in these 30 years shown in figure 6. As a result, the thermal efficiency of the combined cycle power generation with the gas turbine will rise by about 15 percents in 30 years. Nowadays the 1450℃(1723K) class natural gas combustion gas turbine combined cycle power plant was introduced into Tohoku Electric Power Co., Inc. in our country, and its thermal efficiency exceeds 50 percents (on the HHV basis).

The development of gas turbine combustor for IGCC power generation received considerable attention in the 1970s. Other developments concerning the IGCC system and gas turbine combustor using the oxygen-blown gasified coal 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 from 1998; the Wabash River coal gasification repowering plant (Roll, 1995) in the United States, in operation since 1995; the Texaco process at the Tampa power station (Jenkins, 1995), in commercial operation since 1996; a HYCOL gasification process for the purpose of hydrogen production, which was developed in Japan (Ueda et al., 1995); and IGFC (Integrated coal gasification fuel cell combined cycle) pilot plant which consists of gasifier, fuel cell generating unit and gas turbine, in test operation from 2002 by Electric Power Development Co., Ltd. in Japan. Every plants adopt oxygen-blown gasification method.

On the other hand, the Japanese government and electric power industries undertook experimental research project of the air-blown gasification combined cycle system using a 200 tons per day pilot plant (Ichikawa, 1996) from 1986 to 1996. Of late, the government and electric power companies are promoting a demonstration IGCC project. In the future commercializing stage, the transmission-end thermal efficiency of the air-blown IGCC which adopting the 1773K (1500℃)-class (average combustor-outlet gas temperature is about 1773K) gas turbine is expected in excess of 48 percents (on HHV basis), while the thermal efficiency of the demonstration plant using a 1473K (1200℃)-class gas turbine is only 40.5 percents. IGCC technologies improve in the thermal efficiency of 5 points higher than in the case of a pulverized coal firing steam-power generation. The Central Research Institute of Electric Power Industry, called 'CRIEPI' for short, developed an air-blown pressurized two-stage entrained-flow coal gasifier (Kurimura et al., 1995), a hot/dry synthetic gas cleanup system (Nakayama, et al., 1990), 150MW, 1573K-class (Nakata et al., 1993) and 1773K (1500℃)-class gas turbine combustor technologies for low-Btu fuel (Hasegawa et al., 1998a). And in order to accept the developed various IGCC systems, 1773K-class gas turbine combustors of medium-Btu fuel by wet-type or hot/dry-type synthetic gas cleaning method have been proceeded to study (Hasegawa et al., 1997, 1998b, 1999, 2002, 2003, 2005).

Furthermore, the energy resources situation and a geographical condition in each country, and 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 as has been previously described: 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 (Consonni et al., 1997) appear considerably improved at present; The Central Research Institute of Electric Power Industry has started researching into the gasification technology of orimulsion^{TM} (emulsion of Orinoco tar) fuel (Ashizawa et al., 1996). Except for Japan, which has a national research and development project into air-blown entrained-flow IGCC system using a pilot plant with a capacity of 200 tons per day, all of the systems of using the oxygen-blown gasification, are the final stage for commercial operation overseas (Regenbogen, 1995).

The calorific value of gasified coal fuel differs according to the type of gasification agent used in the gasifier. If the gasification agent is air, then gasified coal fuel forms a low calorific fuel of about 4MJ/m^3, but if the agent is oxygen, then the fuel becomes an medium calorific fuel between approximately 9-13MJ/m^3. To increase the thermal efficiency of IGCC, it is necessary to use a hot/dry type gas cleaning system. However the fuel-bound nitrogen of the ammonia (NH_3) that is derived from nitrogenous compounds in coal in the gasifier is not removed in the hot/dry type gas cleaning. This NH_3 is then fed into the gas turbine where it forms fuel-NOx in the combustion process. For this reason, technology to suppress fuel-NOx is important. From the viewpoint of both high operating costs and initial costs of removing the NOx in exhaust gas derived from the gas turbine system, the electric power industry aims for low-NOx combustion technology that promises higher thermal efficiency and environmentally-sound options.

The research into the basic combustion characteristics of gasified coal fuel includes research into the flammability limit of mixed gas consisting of CH_4 or H_2 diluted with N_2, Ar or He (Ishizuka and Tsuji, 1981); the impact of N_2 on burning velocity (Morgan and Kane, 1962); the effect of N_2 and CO_2 on flammability limits (Coward and Jones, 1971), (Ishibasi et al., 1978); the combustion characteristics of low calorific fuel (Folsom et al., 1980), (Drake et al., 1984); White et al. (1983) studies on the rich-lean combustor for low and medium-Btu gaseous fuels; and the Central Research Institute of Electric Power Industry research into the fuel-NOx emissions characteristics of low-calorific fuel including the NH_3 through experiments using a small diffusion burner and analyses based on the reaction kinetics (Sato et al., 1990), (Nakata et al., 1991), (Yamauchi et al., 1991), (Nakata et al., 1998). It is widely accepted that the two-stage combustion as typified by the rich-lean combustion is effective to reduce fuel-NOx emissions (Martin and Dederick, 1977), (Yamagishi et al., 1974).

On the other hand, with respect to the combustion emission characteristics of oxygen-blown medium calorific fuel, Hasegawa et al. (1997) investigated the NOx reduction technology using a small burner; Dobbeling et al. (1996) studied on the premixed combustion characteristics of medium-Btu gaseous fuel in a fundamental small burner for low NOx emissions. Because the burning velocity of medium-Btu fuel was about 6 times greater than conventional natural gas, a premixed combustion for low NOx emissions was so far difficult to adopt. Pillsbury et al. (1976) and Clark et al. (1982) investigated the low-NOx combustion technologies using model combustors. Based on the fundamental knowledge acquired from the above researches, with respect to the research into low-NOx combustion technology using oxygen-blown medium calorific fuel, other studies include: in early phase of 1970s, the medium-Btu fuel was attempted into the gas turbine fuel (Battista and Farrell, 1979), (Beebe et al., 1982); Döbbeling et al. (1994) studied on low NOx combustion technology which quickly mixed fuel with air using the ABB double cone burner, which called EV burner; Cook et al. (1994) studied 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) studied on the effects of steam content in the medium-Btu gaseous fuel on combustion characteristics. In almost all systems, the surplus nitrogen is produced from the oxygen production unit and premixed with a gasified medium-Btu fuel (Becker and Schetter, 1992) from the standpoints of recovering the power oxygen production and of suppressing NOx emissions. Since the power to premix the surplus nitrogen with the medium-Btu fuel is large, Hasegawa et al. studied on low-NOx combustion technology using surplus nitrogen injected from the burner (1998b, 1999) and with lean combustion of instantaneous mixing (2003). Furthermore, Hasegawa et al. have been developing the low-NOx combustion technology for reducing both fuel-NOx and thermal-NOx emissions, in the case of employing the hot/dry synthetic gas cleanup to oxygen-blown IGCC (2002, 2005).

This paper will propose the two-stage combustion or the surplus nitrogen direct injection from the burner to achieve the low-NOx and stable combustion for the following gasification methods:

(1) Air-blown gasifier + Hot/Dry type synthetic gas cleanup method.
(2) Oxygen-blown gasifier + Wet type synthetic gas cleanup method.
(3) Oxygen-blown gasifier + Hot/Dry type synthetic gas cleanup method.

Figure 7 shows the theoretical adiabatic flame temperature of fuels, which were: 1) gaseous fuels with fuel calorific values (HHV) of 12.7, 10.5, 8.4, 6.3, 4.2MJ/m^3, and 2) methane as the main component of natural gas. Flame temperatures were calculated using a CO/H_2 mixture (CO/H_2 molar ratio of 2.33:1), which contained no CH_4 under any condition, and the fuel calorific value was adjusted with nitrogen. In the case of gasified coal fuel, as the fuel calorific value increases, the theoretical adiabatic flame temperature also increases. Fuel calorific values of 4.2MJ/m^3 and 12.7MJ/m^3 produce maximum flame temperatures of 2050K and 2530K respectively. At fuel calorific values of 8.4MJ/m^3 or higher, the maximum flame temperature of the gasified fuel exceeded that of methane, while the fuel calorific value was as low as one fifth of methane. Furthermore, as the fuel calorific value increases, the equivalence ratio, which indicates maximum flame temperature, also increases. This is because the effect of thermal dissociation increases as the adiabatic flame temperature increases. That is to say, in the case of air-blown gasified fuels, fuel calorific values are so low that flame stabilization is a problem which confronts a development of the combustor.

On the other hand, in the case of oxygen-blown gasified fuels, flame temperature is so high that thermal-NOx emissions have to be reduced. Therefore, in oxygen-blown IGCC, N_2 produced by the air separation unit is utilized for recovering power in order to increase the thermal efficiency of the plant and also to reduce the emission of NOx by reducing the flame temperature. Furthermore, when the hot/dry synthetic gas cleanup is employed, ammonia contained in the gasified fuels is not removed and converted into the fuel-NOx. It is necessary to reduce the fuel-NOx emissions in each case of air-blown or oxygen- blown gasifiers.

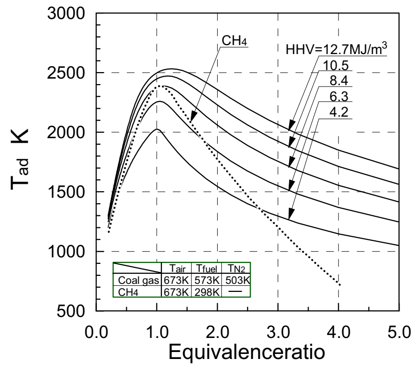


Fig.7 Relationship between equivalence ratio and adiabatic flame temperature for gasified coal fuels and CH_4

Subjects of Gas Turbine Combustors for IGCCs

The typical compositions of gaseous fuel produced in air or oxygen blown gasifiers are shown in Table 1. Each type of gaseous fuel produced raw material with CO and H_2 as the main combustible components, and small amounts of CH_4. Fuel calorific values varied widely (4.2-13.0MJ/m^3), from about one tenth to one third of that of natural gas, depending upon the raw material, the gasification agent and the gasifier type. For example, a gasified fuel derived from biomass contained 30-40 percent steam. That is, since the fuel conditions were various, there were all sorts of subjects in the development of the gasified fueled combustors. Table 2 summarizes the main subjects of the combustor development for each IGCC method.

Table 2 Subjects for combustors of various gasified fuels

		Synthetic gas cleanup	
		Wet type	Hot/Dry type
Gasification agent	Air	・Combustion stability of low-calorific fuel	・Combustion stability of low-calorific fuel ・Reduction of fuel-NOx
	O_2	・Surplus nitrogen supply ・Reduction of thermal-NOx	・Surplus nitrogen supply ・Reduction of thermal- and fuel-NOx emissions

GAS TURBINE COMBUSTORS FOR THE GASIFIED FUELS

COMBUSTOR FOR AIR-BLOWN GASIFICATION SYSTEM WITH HOT/DRY TYPE SYNTHETIC GAS CLEANUP

DESIGN CONCEPT OF COMBUSTOR

Figure 8 shows the relation between the combustor outlet gas temperature and the air distribution in the gas turbine combustor using low-calorific gasified coal fuel. To calculate air distribution, the overall amount of air is assumed to be 100 percent. The amount of air for combustion is first calculated at 1.2 times of a theoretical air(ϕ=0.83), 30 percent of the total air is considered as the cooling air for the combustor liner wall, and the remaining air is considered as diluting air. According to this figure, as the gas turbine temperature rises up to 1773K, the ratio of cooling and diluting air decrease significantly, and the flexibility of the combustor design is minimized. To summarize these characteristics, it can be said that the design concept of the gas turbine combustor utilizing low-calorific fuel should consider the following issues when the gas turbine temperature rises:

(1) Combustion stability; it is necessary to stabilize the flame of

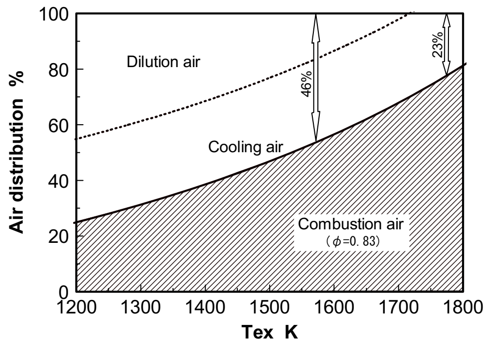


Fig.8 Air distribution design of a gas turbine combustor that burns low-Btu gasified coal fuel

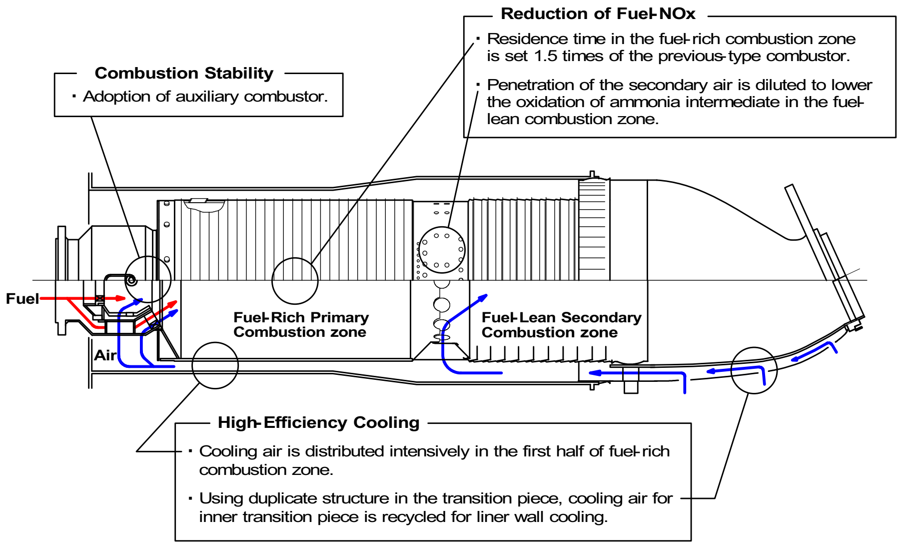


Fig.9 Design concept of 1773K-class low-Btu fueled combustor

Fig.10 Tested combustor

low-calorific fuel.

(2) Low NOx emission technology to restrain the production of fuel NOx from NH_3 in the fuel.

(3) Cooling structure to cool the combustor wall efficiently with less amount of air.

Figure 9 presents characteristics of the designed and tested 1773K-class combustor. Figure 10 illustrates the external view of the burner of the combustor. The main design concept of the combustor was to secure stable combustion of a low-calorific fuel in a wide range of turn-down operation, low NOx emission and enough cooling-air for the combustor liner. The combustor is designed for advanced rich-lean combustion which is effective in decreasing fuel NOx emissions resulting from fuel bound nitrogen.

Assurance of Flame Stabilization

In order to assure flame stability of low-calorific fuel, an auxiliary combustion chamber is installed at the entrance of the combustor. The ratio of the fuel allocated to the auxiliary combustion chamber is 15 percent of the total amount of fuel. The fuel and the combustion air are injected into the chamber through a sub-swirler with a swirling angle of 30 degree. By setting the stoichiometric condition in this chamber under rated load conditions, a stable flame can be maintained. The rest of the fuel is introduced into the main combustion zone from the surrounding of the exit of the auxiliary combustion chamber.

Fuel-NOx Reduction

To restrict the production of fuel NOx that is attributable to NH_3 contained in the fuel, a two-stage combustion method (rich-lean combustion method) is introduced. The tested combustor has a two chamber structure, which separates the primary combustion zone from the secondary combustion zone. In addition, the combustor has two main design characteristics for reducing fuel NOx as indicated below:

Air to Fuel Ratio in Primary Combustion Zone. The equivalence ratio of the primary combustor is determined setting at 1.6 based on the combustion tests previously conducted using a small diffusion burner (Hasegawa et al., 2001).

Figure 11 shows an outline of the experimental device of the small diffusion burner. The combustion apparatus consists of a cylinder-style combustion chamber with an inner diameter, 'D', of 90mm and a length of 1,000mm, and a primary air swirler and fuel injection nozzle. The combustion chamber is lined with heat insulating material and the casing is cooled with water. In order to simulate two-stage combustion, secondary air inlets at a distance from the edge of the fuel injection nozzles of 3×'D' are used. The diameter of the secondary air inlets at the entry to the combustion chamber is 13mm, and six inlets are positioned on the perimeter of one cross-section. The tested burner consists of a fuel injection nozzle and a primary air swirler. There are twelve injection inlets with a diameter of 1.5mm on the fuel injection nozzle with an injection angle, θ, of 90-degree. The primary air swirler has an inner diameter of 24.0mm, an outer diameter of 36.4mm, and twelve vanes with a swirl angle, θ_a, of 45-degree. Swirl number, 'S', which is calculated from the following equation, is 0.84.

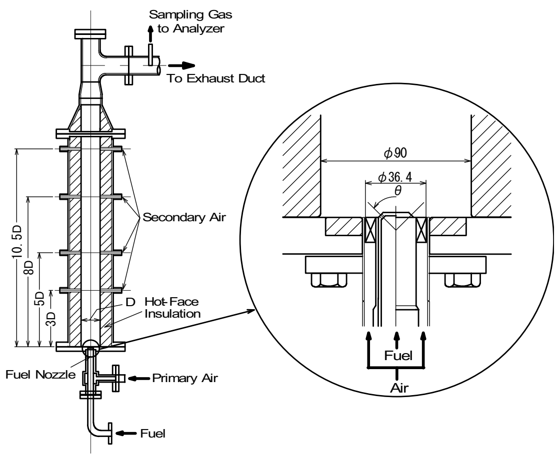
$$S = \frac{2}{3} \times \frac{1-B^3}{1-B^2} \times \tan\theta_a \quad \cdots\cdots (1)$$

Where B (boss ratio of swirl vane)=0.66.

Figure 12 presents an example of the test results which indicates the influence of the equivalence ratio of the primary combustion zone to the conversion rate of NH_3 to NOx, C.R., at the exit of the secondary combustion zone. It also indicates the influence of the CH_4 concentration in the fuel.

$$C.R. = \frac{([NOx]-[NOx_{th}])\times(\text{volume flow rate of exhaust})}{[NH_3]\times(\text{volume flow rate of fuel})} \quad \cdots (2)$$

To obtain the conversion rate of NH_3 to NOx, the concentration of thermal-NOx, '$[NOx_{th}]$', was first measured after stopping the supply of NH_3, then the concentration of total NOx, '[NOx]', was measured while NH_3 was supplied, and finally fuel-NOx was calculated by deducting the concentration of thermal-NOx from that of total NOx. In the tests investigating fuel-NOx emissions, 1000ppm of NH_3 is contained in the low-Btu fuel which consists of CO, H_2 (CO/H_2 molar ratio of 2.33:1), and small amount of CH_4. In the case of changing CH_4 concentration, fuel calorific value was adjusted by N_2 dilution.



D : inner diameter of cylinder-style combustion chamber, 90mm

θ : injection angle of fuel nozzle, 90 degree

Fig.11 Combustion chamber and diffusion burner of basic experimental device

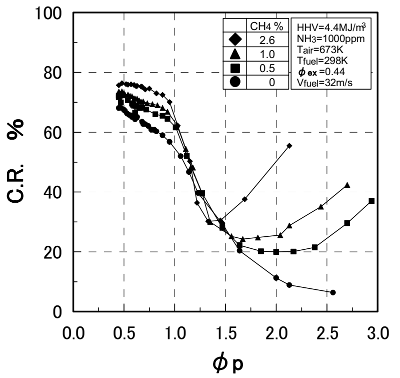


Fig.12 Effect of methane content on conversion rate of ammonia in the fuel to NOx, defining by the experiments using a small diffusion burner (Hasegawa et al., 2001)

From the test results, it is known that the conversion rate of NH_3 to NOx is affected by both the equivalence ratio in the primary combustion zone using the two-staged combustion method and CH_4 concentration in the fuel. When the fuel contains CH_4, HCN produced in the primary-combustion zone is easily converted to NOx in the secondary combustion zone along with the decomposition of NH_3. Therefore, there is a particular equivalence ratio, which minimizes the NOx conversion rate. Based on the fact that low-calorific fuel derived from the IGCC subject to development contained approximately 1.0 percent of CH_4, the equivalence ratio in the primary-combustion zone was set at 1.6. The fuel and the combustion air are injected into the tested combustor through the main swirler, which has 30 degree swirl angle and 15 degree introvert angle, to make these gases premixed.

Introduction Method of Secondary Air. An innovative idea was applied for secondary air introduction. With the decomposition of fuel N, a large portion of the total fixed nitrogen (TFN) produced in the primary combustion zone, including NO, HCN and NH_i, is converted to NOx in the secondary combustion zone. The influence of secondary air mixing conditions on the NOx production was examined from the viewpoint of reaction kinetics with modular model where each combustion zone means a perfect stirred reactor, neither the effect of diffusion nor that of radiant heat transfer of the flame are taken into account. As a result, it was found that the slower mixing of the secondary air made the conversion rate of NH_3 to NOx decline further (Hasegawa et al., 1998a). Based on this result, an exterior wall was installed at the secondary-air inlet section in the tested combustor to make an intermediate pressure zone of the dual structure. By providing this dual structure, the flow speed of the secondary air introduced to the combustor decreased to 70m/s, compared to 120m/s without an exterior wall, thus the secondary air mixing was weakened.

Cooling of Combustor Liner Wall. In order to compensate for the declined cooling air ratio associated with the higher temperature of the gas turbine, 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 that flowed into the transition piece from the exterior wall cools the interior wall with an impingement method, and moves to the combustor liner at the upper streamside.

For the auxiliary combustor and the primary combustion zone in which temperatures are expected to be especially high, the layer-built cooling structure that combined impingement cooling and film cooling was employed. For the secondary combustion zone, the film cooling method was used.

In addition to the above design characteristics, the primary air inlet holes are removed in order to maintain the given fuel-rich conditions in the primary combustion zone. Also, the overall length of the combustor, including the auxiliary chamber, is 1317mm, and the inside diameter is 356mm. The standard conditions in the combustion tests are summarized in Table 3. Combustion Intensity at the design point is 2.0×10^2 W/(m^3·Pa).

Table 3 Standard test conditions

T_{air}	:	700 K
T_{fuel}	:	633 K
T_{ex}	:	1773 K
Ur	:	16 m/s
P	:	1.4 MPa
ϕ_{ex}	:	0.62
Combustion Intensity	:	2.0×10^2 W/(m^3·Pa)

TEST FACILITIES AND TEST METHOD

Test Facilities

The schematic diagram of the test facilities is shown in figure 13. 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 prescribed calorie, and then coal-derived simulated gases were produced.

This facility had another nitrogen supply line, by which nitrogen was directly injected into the combustor. Air supplied to the combustor was provided 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 the prescribed temperature.

The combustion test facility had two test rigs, each of which was capable of performing full-scale atmospheric pressure combustion tests of a single-can for a "several"-hundreds MW-class, multican-type combustor as well as half-scale high-pressure combustion tests, or full-scale high-pressure tests for around a 100MW-class, multican-type combustor. Figure 14 shows a cross-sectional view of the combustor test rig under pressurized 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. An automatic gas analyzer analyzed the components of the combustion gases. After that, the gas temperature was lowered through a quenching pot, using a water spray injection system.

Measurement system

Exhaust gases were sampled from the exit of the combustor through water-cooled stainless steel probes located on the centerline of a height-wise cross section of the measuring duct. The sample lines of exhaust gases were thermally insulated with heat tape to maintain the sampling system above the dew point of the exhaust gas. The exhaust gases were sampled from at an area averaged points in the tail duct exit face and continuously introduced into an emission console which measured CO, CO_2, NO, NOx, O_2, and hydrocarbons by the same methods as the test device for basic studies using the small diffusion burner. The medium-Btu simulated fuel were

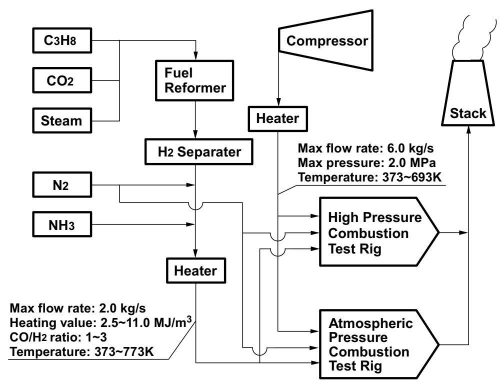


Fig.13 Schematic diagram of test facility

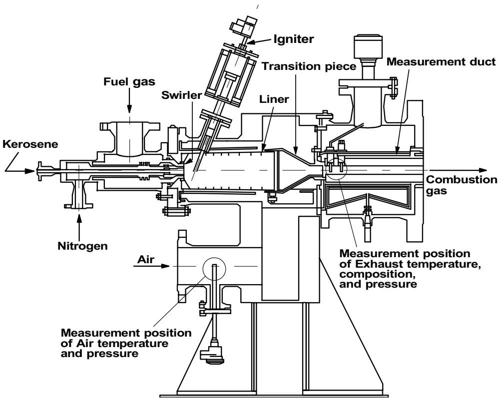


Fig.14 Combustion test rig

sampled from the fuel gas supply line at the inlet of combustor, and constituents of CO, H_2, CH_4, H_2O, CO_2 and N_2 were determined by gas chromatography. Heating values of the simulated 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 walls were measured by sheathed type-K thermocouples with a diameter of 1mm attached to the liner wall with a stainless foil welding. The temperature distributions of the combustor exit gas were measured with an array of three pyrometers, each of which consisted of five type-R thermocouples.

TEST RESULTS

Combustion tests are conducted on under atmospheric pressure conditions. Concerning the pressure influence on the performance of the combustor, a half scale combustor, which has been developed by halving in dimension, was tested under pressurized conditions. Supplied fuels into the combustor were adjusted as same components as that of air-blown entrained-flow gasified coal fuel shown in table 1.

Combustion Emission Characteristics

Figure 15 shows the combustion emission characteristics, under

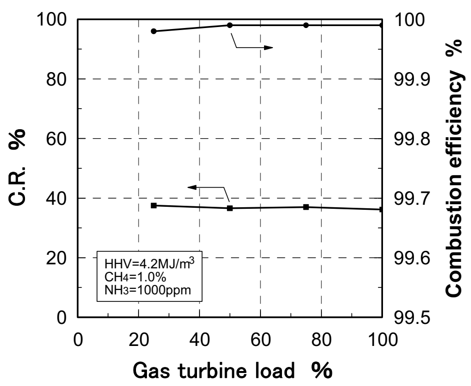


Fig. 15 Combustion emission characteristics

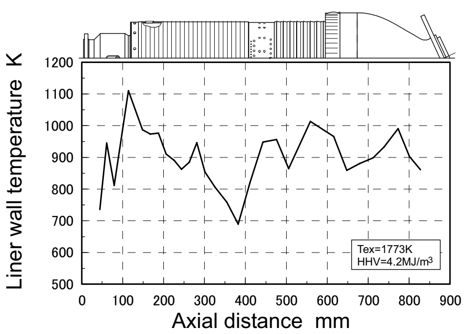


Fig.16 Combustor wall temperature distribution

the gas turbine operational conditions. When the gas turbine load was 25 percent or higher, which is the single fuel firing of gasified coal fuel, the conversion rate of NH_3 to NOx was reduced as low as 40 percent (NOx emissions corrected at 16 percent O_2 was 60ppm), while the combustion efficiency shows around 100 percent in each gas turbine load.

Thermal Characteristics of Combustor Liner Wall

Figure 16 shows the temperature distribution of the combustor liner wall at the rated load condition. From this figure, it could be said that the overall liner wall temperature almost remained under 1123K (850℃), the allowable heat resistant temperature, while the wall temperature increased to an adequate level and a stable flame was maintained in both the auxiliary-combustion chamber and the primary combustion zone.

COMBUSTOR FOR OXYGEN-BLOWN GASIFICATION SYSTEM WITH WET TYPE SYNTHETIC GAS CLEANUP

SUBJECTS OF COMBUSTOR

In the case of oxygen-blown IGCC, the employment an air-separation unit to produce oxygen as gasification agent is the feature of oxygen-blown gasification system, compared with the case of the air-blown gasification ones. That is, the maximum flame temperature of medium-Btu gasified fuel, produced in an oxygen-blown gasifier, is higher than that of each air-blown low-Btu fuel or high-calorie gases such as natural gas which consists mainly of methane. Thermal-NOx emissions are expected to increase in the case of medium-Btu fueled combustors.

Furthermore, in the oxygen-blown IGCC system, large quantity of nitrogen is produced in the air separation unit. In almost all of the systems, a part of nitrogen is used to feed coal into the gasifier and so on, gasified coal fuels are premixed with the rest of the nitrogen and injected into the combustor to increase electric power and to decrease thermal-NOx emissions from the gas turbine. It is necessary to return a large quantity of the surplus nitrogen (as much as the fuel flow rate) to the cycle from the standpoint of recovering power for oxygen production. So, we intend to inject the surplus nitrogen directly into higher temperature regions from the burner and to decrease thermal-NOx emissions produced from these regions effectively. Analyses confirmed that the thermal efficiency of the plant improved by approximately 0.3 percent absolutely by means of nitrogen direct injection into the combustor, compared with a case where nitrogen is premixed with gasified coal fuel before injection into the combustor.

DESIGN CONCEPT OF COMBUSTOR

Figure 17 presents characteristics of the designed, medium-Btu fueled 1573K (1300℃)-class combustor based on the above considerations. The main design concept for the tested combustor was to secure a low-NOx and stable combustion of medium-Btu fuel with nitrogen injection in a wide range of turn-down operations. The overall length of the combustion liner is 650mm and the inside diameter is 230mm.

According to the combustor cooling, a convection method was employed in the transition piece, and moves to the combustor liner on the upstream side. For the primary combustion zone where temperatures are expected to be especially high, the dual-cooling structure was employed, in which the cooling air was impinged from the air flow guide sleeve to the combustion liner and used as film cooling air for the combustion liner. For the secondary combustion zone, the film-cooling method was used.

To restrict thermal-NOx production originating from nitrogen fixation and CO emissions, the burner was designed with nitrogen injection function, based on combustion tests previously conducted using a small diffusion burner (Hasegawa et al., 2001) and a small model combustor (Hasegawa et al., 2003).

Figure 18 presents an example of the test results using the small diffusion burner shown in figure 11, which indicate the influence of

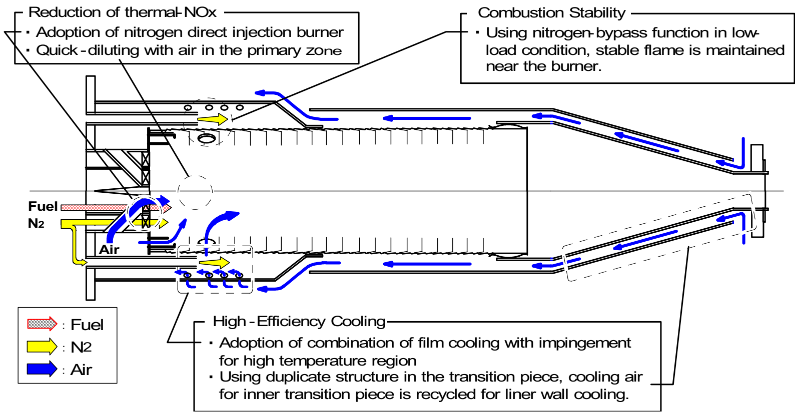


Fig. 17 Design concepts of medium-Btu fueled combustor for wet-type synthetic gas cleanup

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 5.1MJ/m^3. From figure 18, we know that nitrogen blended with fuel or air injected from the burner has a great influence over decreasing NOx emissions from nitrogen fixation. On the other hand, not shown in here, in the case where nitrogen blended with air was injected into the combustor, CO emissions decreased as low as medium-Btu gasified fuel not blended with nitrogen, while CO emissions significantly increased when fuel was blended with nitrogen. That is, in the medium-Btu fuel combustion with nitrogen injection, all of the surplus nitrogen should be injected into the primary combustion zone to reduce the thermal-NOx emissions and should not be blended with fuel, or the primary zone should be fuel lean condition for a low NOx and stable combustion in a wide range of turn-down operations.

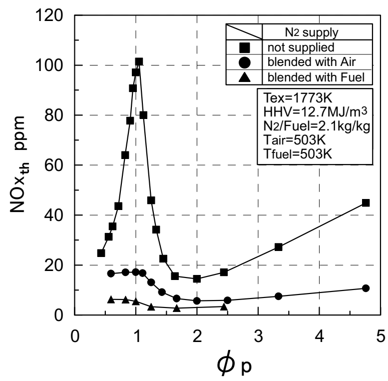
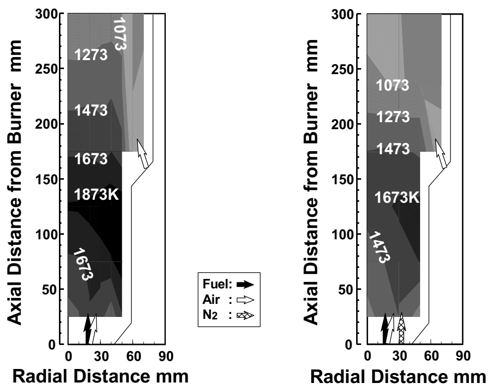


Fig.18 Effect of nitrogen injection on thermal-NOx emission characteristics in two-stage combustion, using a small diffusion burner



(1)No injection of nitrogen (2)Nitrogen injected

Fig.19 Effect of nitrogen injection on combustion gas temperature distribution using a model combustor

Figure 19 shows the combustion gas temperature distribution in the both cases of no nitrogen injection and of nitrogen injection of 1.0kg/kg N_2/Fuel from the burner under atmospheric pressure condition, using a model combustor. In tests, the combustor outlet gas temperature is set at 1373K. From figure 19, we know that nitrogen injection from the burner has a great influence over decreasing hot regions by around 200K in this test conditions. So, in this way of nitrogen injection, thermal-NOx production was restrained one fifth that of the case no nitrogen injection.

Based on these basic test results, we arranged the nitrogen injection intakes in the burner and adopted the lean primary combustion, as shown in figure 17. The nitrogen injected directly into a combustor has the effect of decreasing power to compress nitrogen, compared with the case where the nitrogen was blended with fuel or air evenly. And it is possible to control the mixing of fuel, air and nitrogen positively by way of nitrogen being injected separately into the combustor. The nitrogen direct injection from the burner dilutes the flame of medium-Btu fuel. Furthermore we intended to quench the flame as soon as possible, both by sticking the combustion air injection tubes out of the liner dome and by arranging the secondary combustion air holes on the upstream side of the combustion liner. Design of the combustor was intended for the medium-Btu fuel, the nitrogen injection function was combined with the lean combustion technique for a low NOx combustion. By setting the primary combustion zone to fuel lean state under the rated load condition, the NOx emissions are expected to decrease, and by bypassing nitrogen to premix with the combustion air under partial load conditions, a stable flame can be maintained in a wide range of turn-down operations.

TEST RESULTS

Table 4 and 5 show the typical properties of the supplied fuel and

Table 4 Typical conditions of supplied fuel

Composition	CO	30.4 vol%
	H_2	27.5 vol%
	CH_4	6.8 vol%
	CO_2	35.3 vol%
HHV		10.1 MJ/m^3

Table 5 Rated test conditions

T_{air}	:	603 K
T_{fuel}	:	583 K
TN_2	:	333 K
T_{ex}	:	1700 K
N_2/Fuel ratio	:	0.3 kg/kg
P	:	1.4 MPa
Combustion Intensity	:	2.2×10^2 W/(m^3·Pa)

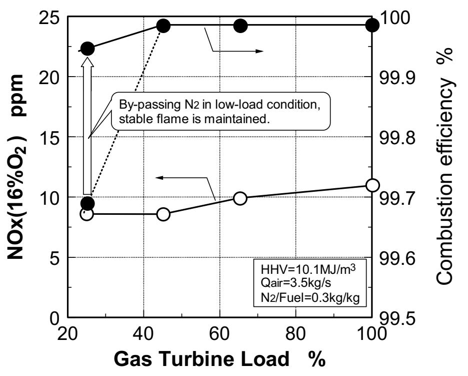


Fig.20 Effect of the gas turbine load on combustion emission characteristics

the standard test conditions, respectively. Higher heating value of the supplied fuel was set at 10.1MJ/m^3, CH_4 was contained higher concentration of 6.8 percent. A part of surplus nitrogen produced from the air-separation unit was used to feed coal or char into the gasifier and the flow rate of the rest was about 0.9 times the fuel flow in the actual process. Since the density of the supplied fuel is higher than that of the gasified coal fuel and temperature of supplied nitrogen is lower in the case of the test conditions than in the actual operations, we also investigate the combustor performance in the case of 0.3kg/kg N_2/Fuel ratio, in which firing temperature of the burner outlet corresponds to the case of actual operations. The rated temperature of combustor-outlet gas is around 1700K and the combustion intensity at the design point is 2.2×10^2W/(m^3·Pa).

Figure 20 shows the relationship between the gas turbine load and the combustion emission characteristics, under the condition where the pressure in the combustor is set to lower level of 1.0MPa at the equivalent, rated load. When the gas turbine load was 25 percent or higher, which is the single fuel firing of gasified coal fuel, the NOx emission was reduced as low as 11ppm (corrected at 16 percent O_2), while the NOx emission tends to increase slightly with the rise in the gas turbine load. Considering the effects of pressure, it could be said that NOx emission was surmised as low as 12ppm (corrected at 16 percent O_2) at any gas turbine load.

On the other hand, combustion efficiency shows around 100 percent in the case where the gas turbine load was 25 percent or higher, by bypassing nitrogen to premix with the combustion air at low load conditions.

COMBUSTOR FOR OXYGEN-BLOWN GASIFICATION SYSTEM WITH HOT/DRY TYPE SYNTHETIC GAS CLEANUP

In order to improve the thermal efficiency of the oxygen-blown IGCC, it is necessary to adopt the hot/dry synthetic gas cleanup. In this case, ammonia contained in the gasified fuels could not be removed and fuel-NOx is emitted from the gas turbine. It is necessary to develop to low NOx combustion technologies that reduce fuel-NOx emissions originating from ammonia in the fuel at the same time as reducing thermal-NOx ones.

SUBJECTS OF COMBUSTOR

From the characteristic of medium-Btu, gasified fuel as mentioned above, it could be said that the design of a gas turbine combustor with nitrogen supply, should consider the following issues for an oxygen-blown IGCC with the hot/dry synthetic gas cleanup:

(1) Low NOx-emission technology: Thermal-NOx production from nitrogen fixation using nitrogen injection, and fuel-NOx emissions originating from ammonia using a two-stage combustion must be simultaneously restrained.

(2) Higher thermal efficiency: Nitrogen injection must be tailored so as to decrease the power to compress nitrogen, which is returned into the gas turbine in order to recover a part of the power used for the air-separation unit.

DESIGN CONCEPT OF COMBUSTOR

Figure 21 presents the configuration and its function of a designed, medium-Btu fueled 1773K (1500℃)-class combustor based on the above considerations. The main design concepts for the tested combustor were to secure stable combustion of medium-Btu fuel with nitrogen injection in a wide range of turn-down operations, and low NOx combustion for reducing fuel-NOx and thermal-NOx emissions. In order to secure stable combustion, we installed an auxiliary combustion chamber at the entrance of the combustor. To reduce thermal-NOx emissions, the nitrogen injection nozzles was set up in the main-swirler, which is installed at exit of the auxiliary combustion chamber. The overall length of the combustion liner is 445mm and the inside diameter is 175mm.

Figure 22 illustrates the axial distribution of equivalence ratio at the rated load condition. In order to reduce the fuel-NOx emissions, we adopted the two-stage combustion, in which a fuel-rich combustion was carried out in the primary zone maintaining the equivalence ratio of 0.66 at exit of the combustor. And the designed combustor has another following characteristics.

Assurance of Flame Stabilization

The ratio of the fuel allocated to the auxiliary combustion chamber is 30 percent of the total amount of fuel. The fuel and air are injected into the chamber through a sub-swirler with a swirling angle

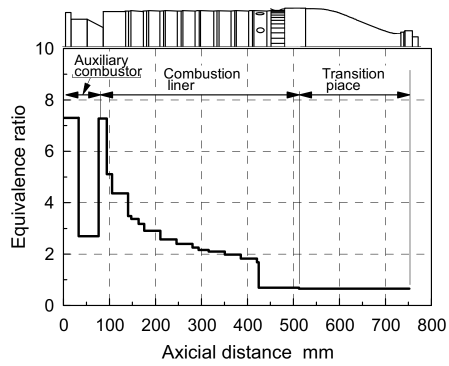


Fig.22 Axial distribution of equivalence ratio at the rated load condition

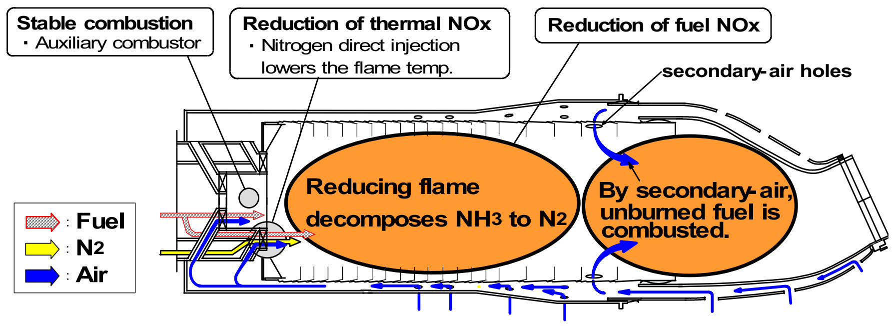


Fig.21 Design concept of a medium-Btu fueled gas turbine combustor for hot/dry-type synthetic gas cleanup

of 30-degree. By setting the mean equivalence ratio in the auxiliary chamber at 2.4 under rated load conditions, a stable flame can be maintained in the fuel-rich combustion zone and reduction of NH_3 to N_2 could proceed in lower load conditions. The rest of the fuel is introduced into the main combustion zone from the surrounding of the exit of the auxiliary combustion chamber.

Nitrogen Injection

From figure 18, we just noticed that nitrogen supply, which is blended with fuel or primary air, drastically decreases thermal-NOx emissions, and also NOx emissions decreases with rises in ϕ_p in the case of using the two-stage combustion. That is, thermal-NOx emissions decrease significantly by setting a fuel-rich condition when ϕ_p is 1.3 or higher in the case of nitrogen premixed with fuel, and by setting ϕ_p at 1.6 or higher in the case of nitrogen premixed with primary combustion air.

With regard to fuel-NOx emissions on the other hand, figure 23 indicates the effects of nitrogen injection conditions on the conversion rate of NH_3 in the fuel to NOx, C.R. in the same conditions with figure 18 except for fuel containing NH_3. In the tests investigating fuel-NOx emissions, 1000ppm of NH_3 is contained in the medium-Btu fuel. In the case of a fuel blended with nitrogen, fuel was diluted, or fuel calorific value decreased to 5.1MJ/m^3 and NH_3 concentration in the fuel decreased to 400ppm. From figure 23, whether with or without nitrogen supplied, the staged combustion method effectively decreased the fuel-NOx emissions, or C.R. drastically decreased as the primary equivalence ratio, ϕ_p, become higher than 1.0, which is a stoichiometric condition, and shows the minimum value at the appropriate ϕ_p. Those optimum ϕ_p become lower when the medium-Btu fuel was blended with nitrogen, while the optimum ϕ_p was in a wide range in the case of nitrogen blended with the primary combustion air injected from the burner, and C.R. showed a tendency to become a little higher than in the other two cases. Furthermore, under lean-lean combustion conditions with a lower ϕ_p than 1.0, in the case of nitrogen premixed with medium-Btu fuel, C.R. becomes higher than in the case of nitrogen premixed with the primary combustion air.

From the above, it was shown that the supply method of nitrogen premixed with medium-Btu fuel possibly decreases total emissions of thermal-NOx and fuel-NOx, but careful attention must be paid to the homogeneity of mixture of fuel and nitrogen, or thermal-NOx emissions will increase. In the case of nitrogen premixed with the primary combustion air, total NOx emissions grow slightly higher than the case of nitrogen premixed with fuel, and the power to compress nitrogen increases a little or the thermal efficiency of the plant decreases. That is, it is necessary to blend nitrogen with medium-Btu fuel evenly in the combustor, in which the lowest power to compress nitrogen is needed for nitrogen supply into the gas turbine, and not to collide the medium-Btu fuel with combustion air directly.

Based on these basic experimental results, we arranged the nitrogen injection intakes between fuel and air intakes in the main-swirler surrounding the primary-flame from the auxiliary combustion chamber for low thermal-NOx emissions. Additionally the fuel, the combustion air, and the nitrogen from the burner are separately injected into the combustor through a swirler, (which has a 30-degree swirl angle and a 15-degree introverted angle), to collide medium-Btu fuel with air in an atmosphere where nitrogen is superior in amount to both fuel and air.

Fuel-NOx/Thermal-NOx Reduction

In order to decrease fuel-NOx emissions, we adopted fuel-rich combustion in the primary zone and set the equivalence ratio in the primary-combustion zone is determined based on the combustion test results using a small diffusion burner shown in figure 11. Figure 24 presents a relation between the primary equivalence ratio, ϕ_p, and the conversion rate of NH_3 to NOx, C.R., with CH_4 concentration as a parameter in two-staged combustion. In test, the average temperature of the exhaust, T_{ex}, is set to 1773K and fuel calorific value is 11.4MJ/m^3 for fuel containing 1000ppm of NH_3, CO and H_2 of 2.33 CO/H_2 molar ratio. In the same way as low-Btu fuels, the primary equivalence ratio that minimizes the conversion rate of NH_3 to NOx is affected by CH_4 concentration in the fuel. Because the supplied fuel contains 3 percent of CH_4, the equivalence ratio in the primary-combustion zone was set around 1.9 and the equivalence ratio in the auxiliary-combustion chamber was around 2.4 to maintain the flame stabilization and to improve reduction of NH_3, simultaneously.

The effect of the CH_4 concentration on the fuel-NOx produced by NH_3 in gasified coal fuel was studied using the elementary reaction kinetics (Hasegawa et al., 2001). The model of the flow inside the combustor introduced the Pratt model (Pratt et al., 1971) and each stage combustion zone is assumed to be a perfectly stirred reactor. The reaction model employed here was proposed by Miller and Bowman(1989), values for thermodynamic data were taken from the JANAF thermodynamics tables(Chase et al., 1985) or calculated based on the relationship between the Gibbs' standard energy of formation and the chemical equilibrium constant. The values of Gibbs' standard energy of formation were obtained from the CHEMKIN database (Kee et al., 1990). The GEAR method (Hindmarsh, 1974) was used for the numerical analysis. Also, it is assumed that the species are evenly mixed, and diffusion and stirring processes are not taken into consideration in the reaction process. The appropriateness of the model for reaction NH_3 with NO (Hasegawa, 1998c) and the oxidation of ammonia in premixed flame of methane (Miller and Bowman, 1989) has been confirmed by comparison with test results.

The nitrogen of NH_3 in the fuel has weaker bonding power than N_2. In the combustion process, NH_3 reacted with the OH, O, and H radicals and then easily decomposed into the intermediate NH_i by the following reactions (Miller et al., 1983).

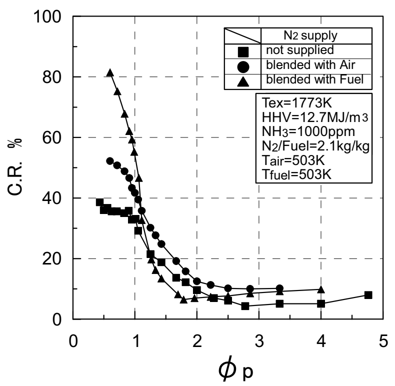


Fig.23 Effect of nitrogen injection on the conversion rate of NH_3 to NOx in two-stage combustion, using a small diffusion burner

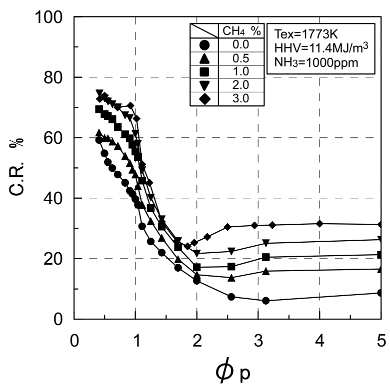


Fig.24 Effect of the CH_4 concentration on conversion rate of NH_3 to NOx in two-stage combustion of medium-Btu fuel

$NH_3 + OH(O,H) \Leftrightarrow NH_2 + H_2O(OH,H_2)$ ··························(3)

$NH_i(i=1,2) + OH(H) \Leftrightarrow NH_{i-1} + H_2O(H_2)$ ························ (4)

When hydrocarbon is not contained in the fuel, NH_i is converted into N_2 by reacting with NO in the fuel-rich region. If fuel contains CH_4, HCN is produced by reactions 5 and 6 in the fuel-rich region and the HCN is oxidized to NO in the fuel-lean zone (Heap et al, 1976),(Takagi et al, 1976 and 1978) and (Kato, et al., 1977).

$CH_i(i=1,2) + N_2 \Leftrightarrow HCN + NH_{i-1}$ ··································(5)

$R\text{-}CH + NH_i \Leftrightarrow HCN + R\text{-}H_i$, (R- : Alkyl group) ···········(6)

Some HCN is oxidized into NO by reactions 7 and 8, and the rest is decomposed into N radical by the reaction 9. NH radical is decomposed into the NO by reactions 10, 11, and 12. With the rise in CH_4 concentration in gasified coal fuel, the HCN increases, and NOx emissions originated from HCN in the fuel-lean secondary combustion zone increase.

$HCN+OH \Leftrightarrow CN+H_2O$ ···(7)

$CN+O_2 \Leftrightarrow CO+NO$ ···(8)

$CN+O \Leftrightarrow CO+N$ ···(9)

$NH+OH \Leftrightarrow N+H_2O$ ··· (10)

$N+O_2 \Leftrightarrow NO+O$ ·· (11)

$N+OH \Leftrightarrow NO+H$ ·· (12)

On the other hand, some NH radical produced by the reactions 3, 4 and 5 are reacted with Zel'dovich NO, Prompt NO and fuel-N oxidized NO, which produced by above reactions, and decomposed into N_2 by the reaction 13.

$NO+NH \Leftrightarrow N_2+OH$ ··· (13)

That is, it is surmised that each of increase in thermal-NOx concentration and fuel-NOx affected the alternative decomposition reaction of intermediate NH radical with NO, so the each of NOx emissions originated from the nitrogen in the air or fuel-N decreased.

These new techniques those adopted the nitrogen direct injection and the two-stage combustion, caused a decrease in flame temperature in the primary combustion zone and the thermal-NOx production near the burner was expected to be controlled. On the contrary, we were afraid that the flame temperature near the burner was declined too low at lower load conditions and so a stable combustion cannot be maintained. The designed combustor was given another nitrogen injection function, in which nitrogen was bypassed to premix with the air derived from the compressor at lower load conditions, and a stable flame can be maintained in a wide range of turn-down operations. Also, because the nitrogen dilution in the fuel-rich region affected the reduction characteristics of NH_3, the increase in nitrogen dilution raised the conversion rates of NH_3 to NOx. This tendency showed the same as that of the case where nitrogenous compounds in fuel increased, indicated by Sarofim et al.(1975), Kato et al.(1976), Fenimore(1972) and Takagi et al.(1977). That is, it is necessary that the nitrogen bypassing technique is expected to improve fuel-NOx reduction in the cases of higher concentration of NH_3.

TEST RESULTS

Table 6 shows the composition of the supplied fuel used in this paper and the typical commercial gasified fuel. In tests, the effects of the concentrations of CH_4 and NH_3 in the supplied fuels on the combustion characteristics were investigated and the combustor's performances were predicted in the typical commercial operations. Figure 25 estimates the combustion emission characteristics under the simulated operational conditions of 1773K-class gas turbine for IGCC in the case where gasified fuel contains 0.1 percent CH_4 and 500ppm NH_3. Total NOx emissions were surmised as low as 34ppm (corrected at 16 percent O_2) in the range where the gas turbine load was 25 percent or higher, which is the single fuel firing of gasified coal fuel, while the NOx emissions tend to increase slightly with the rise in the gas turbine load. In the tests of the simulated fuel that contained no NH_3, thermal-NOx emissions were as low as 8ppm (corrected at 16 percent O_2). On the other hand, we can expect that combustion efficiency is around 100 percent under operational conditions of the medium-Btu fueled gas turbine.

Table 6 Comparison of supplied fuel in tests and typical case of commercial gasified fuel

	Supplied fuel in test	Commercial gasified fuel
Constituent		
CO	31.4 %	40.9 %
H_2	28.6 %	29.9 %
CH_4	0～3.0 %*1	0.1 %
CO_2	32.0 %	9.5 %
H_2O	0.0 %	12.3 %
N_2	5.0 %	7.3 %
NH_3	0～3000 ppm*2	500 ppm
HHV	8.8 MJ/m^3	9.0 MJ/m^3
LHV	8.1 MJ/m^3	8.2 MJ/m^3

*1: In the case of varying CH_4 concentration in the fuel, the CO and H_2 constituents were adjusted to maintain the fuel calorific value and the CO/H_2 molar ratio constantly, and the combustor's performance of commercial gasified fueled-combustion was predicted.

*2: NH_3 concentration was different according to the gasification methods and raw materials.

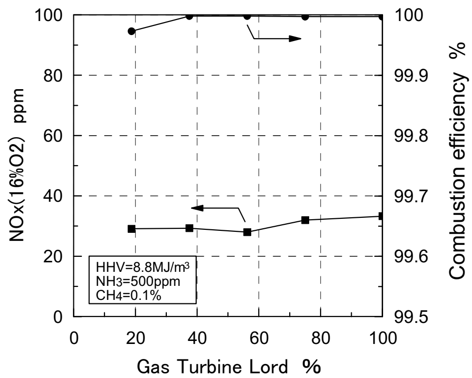


Fig.25 Combustion emission characteristics

CONCLUSIONS

Based on basic combustion test results using small burners and model combustors, Japanese electric industries proposed the correspond combustion technologies for each gasified fuels, designed combustors fitted with a suitable nitrogen injection nozzle, two-stage combustion, or lean combustion for each gasified fuel, and demonstrated those combustors' performances under gas turbine operational conditions. As summarized in Table 7, the developed combustors showed to be completely-satisfied with the performances of 1773K-class gas turbine combustor in the actual operations. That is, these combustion technologies reduced each type of NOx emissions for each gasified fuel, while maintaining the other combustor's characteristics enough. Furthermore, developed technologies represent a possible step towards the 1873K-class gas

Table 7 Performances of gasified fueled combustors

		Synthetic gas cleanup	
		Wet type	Hot/Dry type
Gasification agent	Air	・1573K-class gas turbine combustor for BFG ・thermal-NOx ≦20ppm	・1773K-class combustor ・NOx emissions ≦60ppm ・thermal-NOx ≦ 8ppm ・P.F.(rated) ≦ 8%
	O_2	・1573K-class combustor ・thermal-NOx ≦11ppm ・P.F.(rated) =10～13%	・1773K-class combustor ・NOx emissions ≦34ppm ・thermal-NOx ≦8ppm ・P.F.(rated) ≦ 7%

P.F.(Pattern factor) = $(T_{max} - T_{ex})/(T_{ex} - T_{air}) \times 100$[%]

T_{max} : Local maximum temperature of combustor exhaust [K] turbine combustor.

To keep stable supplies of energy and protect the global environment, it will be important that human beings not only use the finite fossil fuel, such as oil and coal, but also have to reexamine unused resources and reclaim waste, and develop the highly effective usage of such resources. The IGCC technologies could have potential the highly-efficient use of various resources those are not use today widely for power generation.

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