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Developments of Gas Turbine Combustors for Air-Blown and Oxygen-Blown IGCC

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1. Introduction

From the viewpoints of securing a stable supply of energy and protecting our global environment in the future, the integrated gasification combined cycle (IGCC) power generation of various gasifying methods has been introduced in the world. Gasified fuels are chiefly characterized by the gasifying agents and the synthetic gas cleanup methods and can be divided into four types. The calorific value of the gasified fuel varies according to the gasifying agents and feedstocks of various resources, and ammonia originating from nitrogenous compounds in the feedstocks depends on the synthetic gas clean-up methods. In particular, air-blown gasified fuels provide low calorific fuel of 4 MJ/m^3 and it is necessary to stabilize combustion. In contrast, the flame temperature of oxygen-blown gasified fuel of medium calorie between approximately 9–13 MJ/m^3 is much higher, so control of thermal-NOx emissions is necessary. Moreover, to improve the thermal efficiency of IGCC, hot/dry type synthetic gas clean-up is needed. However, ammonia in the fuel is not removed and is supplied into the gas turbine where fuel-NOx is formed in the combustor. For these reasons, suitable combustion technology for each gasified fuel is important. In this paper, I will review our developments of the gas turbine combustors for the three type gasified fuels produced from the following gasification methods through experiments using a small diffusion burner and the designed combustors' tests of the simulated gasified fuels.

- Air-blown gasifier + Hot/Dry type synthetic gas clean-up method.
- Oxygen-blown gasifier + Wet type synthetic gas clean-up method.
- Oxygen-blown gasifier + Hot/Dry type synthetic gas clean-up method.

Figure 1 provides an outline of a typical oxygen-blown IGCC system. In this system, 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 through a dust removing and desulfurizing process. The cleaned synthetic gas is then fed into the high-efficiency gas turbine topping cycle, and the steam cycle is equipped to recover heat from the gas turbine exhaust. This IGCC system is similar to LNG fired gas turbine combined cycle generation, except for the gasification and the synthetic gas cleanup process, primarily. IGCC requires slightly more station service power than an LNG gas turbine power generation.
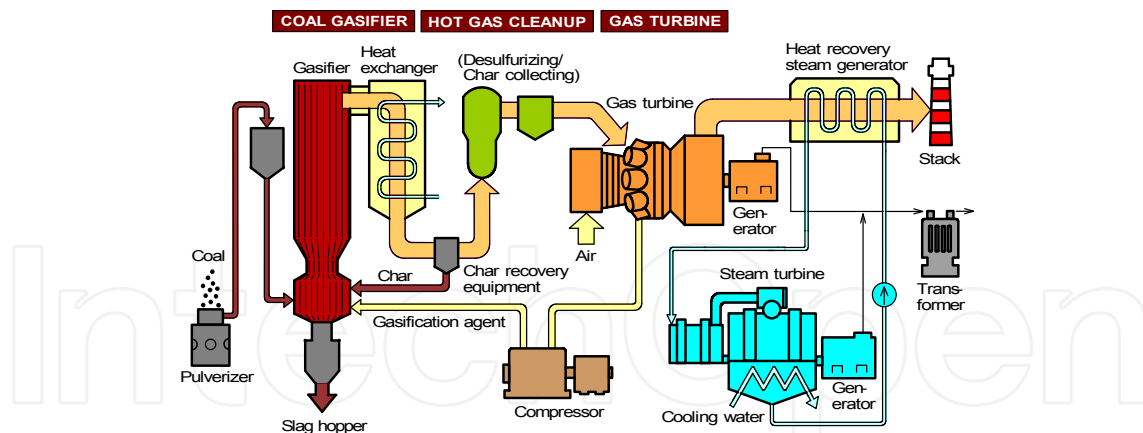
Fig. 1. Schematic diagram of typical IGCC system

1.1 Background of IGCC development in the world

The development of the gas turbine combustor for IGCC power generation received considerable attention in the 1970s. Brown (1982), summarized the overall progress of IGCC technology worldwide up until 1980. The history and application of gasification was also mentioned by Littlewood (1977). Concerning fixed-bed type gasification processes, Hobbs et al. (1993) extensively reviewed the technical and scientific aspects of the various systems. Other developments concerning the IGCC system and gas turbine combustor using oxygen-blown gasified coal fuel include: The Cool Water Coal Gasification Project (Savelli & Touchton, 1985), the flagship demonstration plant of gasification and gasified fueled gas turbine generation; the Shell process (Bush et al., 1991) in Buggenum, the first commercial plant, which started test operation in 1994 and commercial operation in 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; and an integrated coal gasification fuel cell combined cycle pilot plant, consisting of a gasifier, fuel cell generating unit and gas turbine, in test operation since 2002 by Electric Power Development Co. Ltd. in Japan. Every plant adopted the oxygen-blown gasification method. With regard to fossil-based gasification technology as described above, commercially-based power plants have been developed, and new development challenges toward global carbon capture storage (Isles, 2007; Beer, 2007) are being addressed.

Meanwhile, from 1986 to 1996, the Japanese government and electric power companies undertook an experimental research project for the air-blown gasification combined cycle system using a 200-ton-daily pilot plant. Recently, the government and electric power companies have also been promoting a demonstration IGCC project with a capacity of 1700 tons per day (Nagano, 2009). For the future commercializing stage, the transmission-end thermal efficiency of air-blown IGCC, adopting the 1773 K (1500°C)-class (average combustor exhaust gas temperature at about 1773 K) gas turbine, is expected to exceed 48%(on HHV basis), while the thermal efficiency of the demonstration plant using a 1473 K (1200°C)-class gas turbine is only 40.5%. IGCC technologies would improve thermal efficiency by five points or higher compared to the latest pulverized coal-firing, steam power generation. The Central Research Institute of Electric Power Industry (CRIEPI), developed an air-blown two-stage entrained-flow coal gasifier (Kurimura et al., 1995), a hot/dry synthetic gas cleanup system (Nakayama et al., 1990), and 150MW, 1773K(1500°C)-class gas turbine combustor technologies for low-Btu fuel (Hasegawa et al., 1998a). In order

to accept the various IGCC systems, 1773K-class gas turbine combustors of medium-Btu fuels by wet-type or hot/dry-type synthetic gas cleanup methods have undergone study (Hasegawa et al., 2003, 2007).

The energy resources and geographical conditions of each country, along with the diversification of fuels used for the electric power industry (such as biomass, poor quality coal and residual oil), are most significant issues for IGCC gas turbine development, as has been previously described: The development of biomass-fueled gasification received considerable attention in the United States and northern Europe in the early 1980s (Kelleher, 1985), and the prospects for commercialization technology (Consonni, 1997) appear considerably improved at present. Paisley and Anson (1997) performed a comprehensive economical evaluation of the Battele biomass gasification process, which utilizes a hot-gas conditioning catalyst for dry synthetic gas cleanup. In northern Europe, fixed-bed gasification heating plants built in the 1980s had been in commercial operation; the available technical and economical operation data convinced small district heating companies that biomass or peat-fueled gasification heating plants in the size class of 5 MW were the most profitable (Haavisto, 1996). However, during the period of stable global economy and oil prices, non-fossil-fueled gasification received little interest. Then, in the early 2000s when the Third Conference of Parties to the United Nations Framework Convention on Climate Change (COP3) invoked mandatory carbon dioxide emissions reductions on countries, biomass-fueled gasification technology began to receive considerable attention as one alternative. With the exception of Japanese national research and development project, almost all of the systems using the oxygen-blown gasification are in their final stages for commencing commercial operations overseas.

1.2 Progress in gas turbine combustion technologies for IGCCs

The plant thermal efficiency has been improved by enhancing the turbine inlet temperature, or combustor exhaust temperature. The thermal-NOx emissions from the gas turbines increase, however, along with a rise in exhaust temperature. In addition, gasified fuel containing NH_3 emits fuel-NOx when hot/dry gas cleanup equipment is employed. It is therefore viewed as necessary to adopt a suitable combustion technology for each IGCC in the development of a gas turbine for each gasification method.

Dixon-Lewis and Williams (1969), expounded on the oxidation characteristics of hydrogen and carbon monoxide in 1969. The body of research into the basic combustion characteristics of gasified fuel includes studies on the flammability limits of mixed gas, consisting of CH_4 or H_2 diluted with N_2, Ar or He (Ishizuka & Tsuji, 1980); a review of the flammability and explosion limits of H_2 and H_2/CO fuels (Cohen, 1992); the impact of N_2 on burning velocity (Morgan & Kane, 1952); the effect of N_2 and CO_2 on flammability limits (Coward & Jones, 1952; Ishibasi et al, 1978); and the combustion characteristics of low calorific fuel (Folsom, 1980; Drake, 1984); studies by Merryman et al. (1997), on NOx formation in CO flame; studies by Miller et al. (1984), on the conversion characteristics of HCN in $H_2\text{-}O_2\text{-}HCN\text{-}Ar$ flames; studies by Song et al. (1980), on the effects of fuel-rich combustion on the conversion of the fixed nitrogen to N_2; studies by White et al. (1983), on a rich-lean combustor for low-Btu and medium-Btu gaseous fuels; and research of the CRIEPI into fuel-NOx emission characteristics of low-calorific fuel, including NH_3 through experiments using a small diffusion burner and analyses based on reaction kinetics (Hasegawa et al, 2001). It is widely

accepted that two-stage combustion, as typified by rich-lean combustion, is effective in reducing fuel-NOx emissions (Martin & Dederick, 1976; Yamagishi et al, 1974).

On the other hand, with respect to the combustion emission characteristics of oxygen-blown medium calorific fuel, Pillsbury et al. (1976) and Clark et al. (1982) investigated low-NOx combustion technologies using model combustors. In the 1970s, Battista and Farrell (1979) and Beebe et al. (1982) attempted one of the earliest tests using medium-Btu fuel in a gas turbine combustor. Concerning research into low-NOx combustion technology using oxygen-blown medium calorific fuel, other studies include: Hasegawa et al. (1997), investigation of NOx reduction technology using a small burner; and studies by Döbbeling et al. (1994), on low NOx combustion technology (which quickly mixed fuel with air using the double cone burner from Alstom Power, called an EV burner); Cook et al. (1994), on effective methods for returning nitrogen to the cycle, where nitrogen is injected from the head end of the combustor for NOx control; and Zanello and Tasselli (1996), on the effects of steam content in medium-Btu gaseous fuel on combustion characteristics. In almost all systems, surplus nitrogen was produced from the oxygen production unit and premixed with a gasified medium-Btu fuel (Becker & Shetter, 1992), for recovering power used in oxygen production and suppressing NOx emissions. Since the power to premix the surplus nitrogen with the medium-Btu fuel is great, Hasegawa et al. studied low-NOx combustion technologies using surplus nitrogen injected from the burner (Hasegawa et al, 1998b) and with the lean combustion of instantaneous mixing (Hasegawa et al, 2003). Furthermore, Hasegawa and Tamaru(2007) developed a low-NOx combustion technology for reducing both fuel-NOx and thermal-NOx emissions, in the case of employing hot/dry synthetic gas cleanup with an oxygen-blown IGCC.

1.3 Subjects of gas turbine combustors for IGCCs

The typical compositions of gasified fuels produced in air-blown or oxygen-blown gasifiers, and in blast furnaces, are shown in Tables 1. Each type of gaseous mixture fuel consists of CO and H_2 as the main combustible components, and small percentages of CH_4. Fuel calorific values vary widely (2–13 MJ/m^3), from about 1/20 to 1/3 those of natural gas, depending upon the raw materials of feedstock, the gasification agent and the gasifier type.

Figure 2 shows the theoretical adiabatic flame temperature of fuels which were: (1) gasified fuels with fuel calorific values (HHV) of 12.7, 10.5, 8.4, 6.3, 4.2 MJ/m^3; and (2) fuels in which methane is the main component of natural gas. Flame temperatures were calculated using a CO and H_2 mixture fuel (CO/H_2 molar ratio of 2.33:1), which contained no CH_4 under any conditions, and the fuel calorific value was adjusted with nitrogen. In the case of gasified fuel, as the fuel calorific value increased, the theoretical adiabatic flame temperature also increased. Fuel calorific values of 4.2 MJ/m^3 and 12.7 MJ/m^3 produced maximum flame temperatures of 2050 K and 2530 K, respectively. At fuel calorific values of 8.4 MJ/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, each quantity of CO and H_2 constituent in the gasified fuels differed, chiefly according to the gasification methods of gasifying agents, raw materials of feedstock, and water-gas-shift reaction as an optional extra for carbon capture system. However, it could be said that the theoretical adiabatic flame temperature was only a little bit affected by the CO/H_2 molar ratio in the case of each fuel shown in Tables 1. That is to say, in air-blown gasified fuels, fuel calorific values are so low that flame stabilization is a problem confronting development of the combustor.

Fuel	BFG	COG	Gasified fuel									
Resource			Waste	RDP	Coal					wood		Heavy residue
Gasifier			Fluidized		Fixed	Fluidized	Entrained			Fluidized		Entrained
Coal supply					Dry	Dry	Dry	Dry	Slurry			
Developer					BGL	BC	IGC	Shell	Texaco	Lurgi		Texaco
Oxydizer			Air	Air+O$_2$	O$_2$	Air	Air	O$_2$	O$_2$	Air	O$_2$	O$_2$
Composition												
CO[%]	20	6	6	30	56.4	7.9-14.7	25.9-27.6	65.2-69.5	40.9	20	46	51.7
H$_2$[%]	3	56	1.6	22	25.6	13.2-15.0	10.9-9.4	28.8-31.0	29.9	10	13	43.1
CH$_4$[%]	-	30	0.9	0.4	6.6	1.5-2.8	1.4-0.5	0.01-0.03	0.1	5	10	0.2
CO$_2$[%]	20	-	12.4	4.1	2.8	10-12	6.7-5.4	1.0-2.8	9.5	14	23	3.2
H$_2$O[%]	-	-	23.4	5.9	-(a)	11.5-18.4	-(a)	-(b)	12.3	-	-	-(b)
NH$_3$[ppm]	-	-	-	-	-(a)	500-1000	1000	100-600	-(a)	-	-	-(a)
H$_2$S+COS[%]	-	-	-	-	20ppm	-(a)	404-714ppm	0.14-1.1	-(a)	-	-	1.6
Others[%]	N$_2$	C$_2$H$_2$ etc.	N$_2$	C$_2$H$_2$ etc.	8.6	45.9-47.3	54.2-56.1	-(a)	7.3	51	8	0.2
CO/H$_2$	7	0.1	3.8	1.4	2.2	-(a)	2.4-3.0	2.1-2.4	1.4	2	3.5	1.2
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	9.0	5.8	11.4	12.1

BFG:Blast furnace gas, COG:Coke-oven gas, RDF:Refuse derived fuel, Waste:Municipal solid waste, (a):No description, (b):Dry base
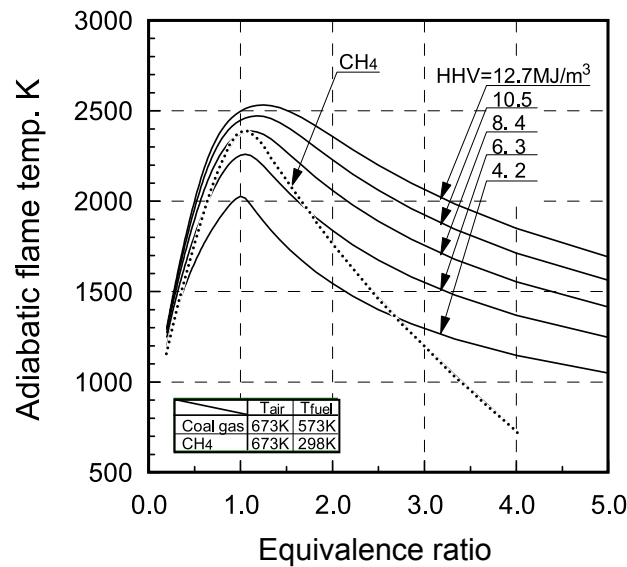
Table 1. Various gasified fuels

Fig. 2. Relationship between equivalence ratio and adiabatic flame temperature for gasified fuels and CH$_4$.

On the other hand, in the case of oxygen-blown gasified fuels, flame temperature is so high that thermal-NOx emissions must be reduced. Therefore, in oxygen-blown IGCC, N$_2$ produced by the air separation unit is used to recover power to increase the thermal efficiency of the plant, and to reduce NOx emissions from the gas turbine combustor by reducing the flame temperature. Furthermore, when hot/dry synthetic gas cleanup is employed, ammonia contained in the gasified fuels is not removed, but converted into fuel-NOx in the combustor. It is therefore necessary to reduce the fuel-NOx emissions in each case of air-blown or oxygen-blown gasifiers.

Because fuel conditions vary depending on the gasification method, many subjects arose in the development of the gasified fueled combustor. Table 2 summarizes the main subjects of combustor development for each IGCC method.

		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

Table 2. Subjects for combustors of various gasified fuels

2. Test facilities and method for gasified fueled combustors

This chapter indicates a typical example of a test facility and method for a single-can combustion test using simulated gasified fuels.

2.1 Test facilities

The schematic diagram of the test facilities is shown in Figure 3. 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 simulated gases derived from gasifiers 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.
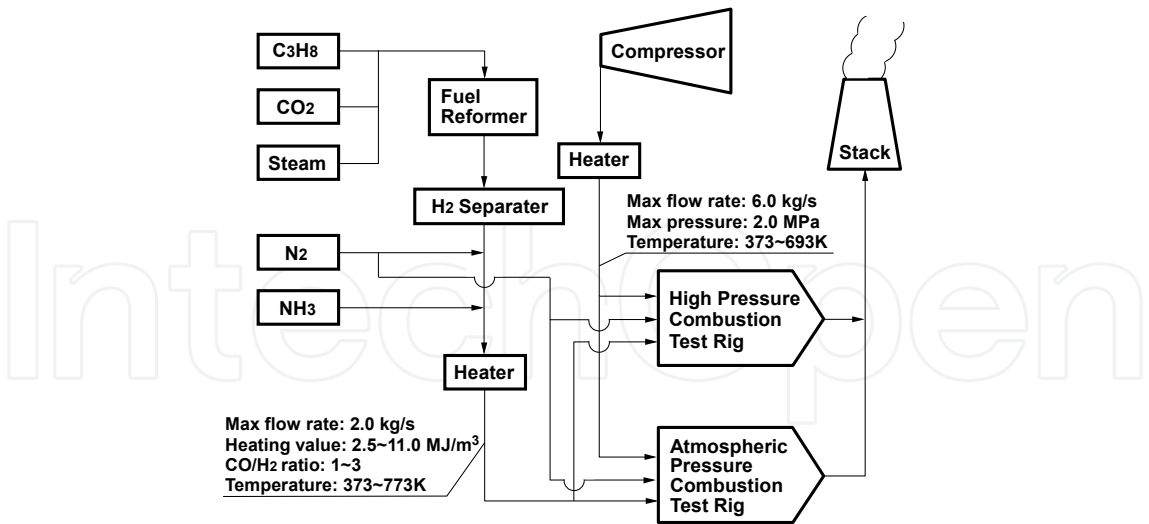

Fig. 3. Schematic diagram and specifications of test facility

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 4 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.
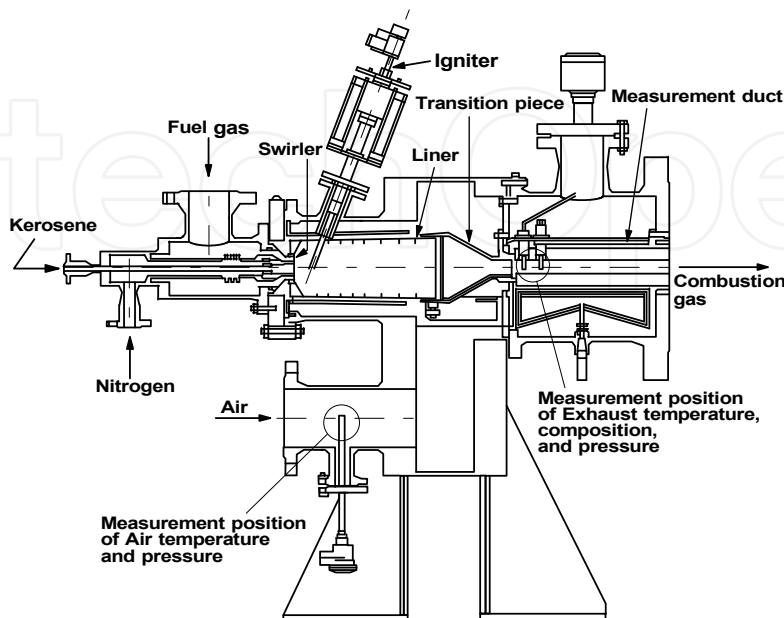

Fig. 4. Combustion test rig

2.2 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 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.

3. Gas turbine combustors for the gasified fuels

This chapter indicates the characteristics of the combustion technologies being applied to the gasified fuels classified into four types in Table 2. Based on the knowledge through experiments using a small diffusion burner and numerical analyses, prototype combustors were constructed, tested and their performances were demonstrated.

3.1 Combustor for air-blown gasification system with hot/dry type synthetic gas cleanup

3.1.1 Design concept of combustor

Figure 5 shows the relation between the combustor exhaust temperature and the air distribution in the gas turbine combustor using low-calorific gasified 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 tur-bine combustor utilizing low-calorific fuel should consider the following issues when the gas turbine temperature rises:

- Combustion stability; it is necessary to stabilize the flame of low-calorific fuel.
- Low NOx emission technology to restrain the production of fuel NOx.
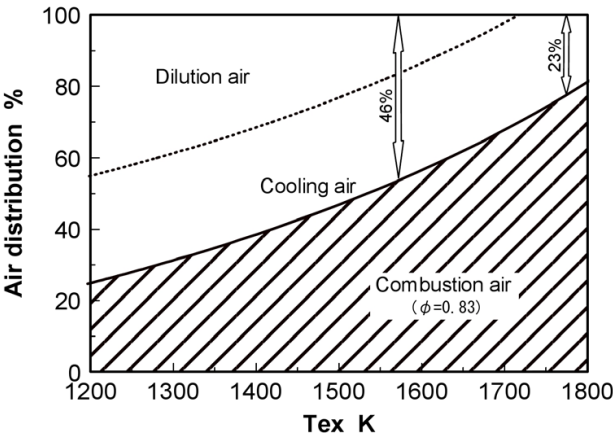- Cooling structure to cool the combustor wall efficiently with less amount of air.

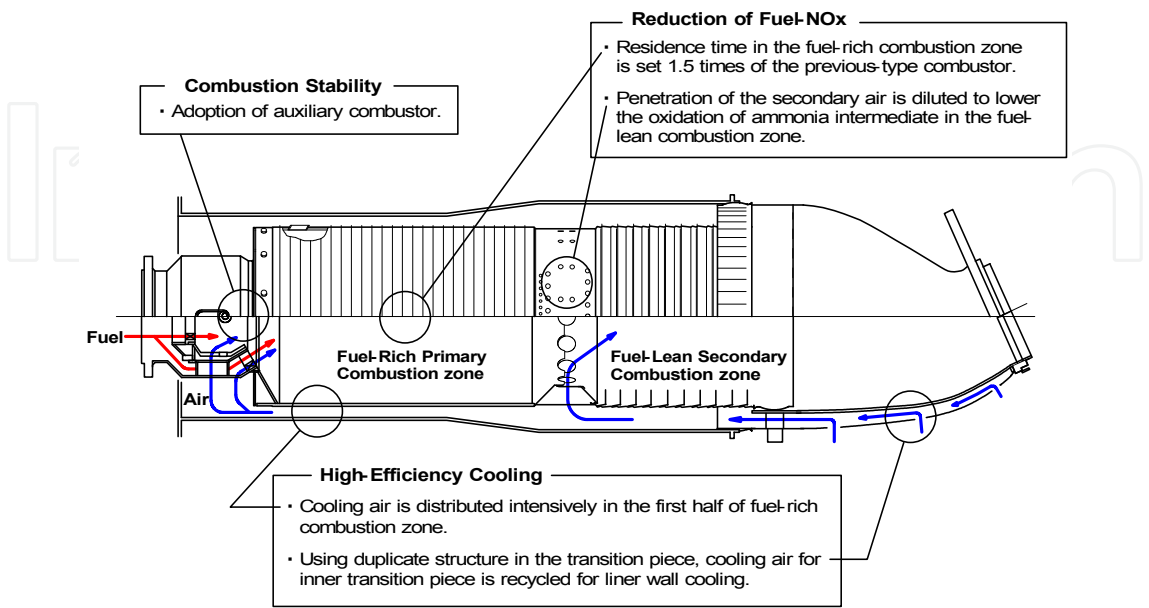Fig. 5. Air distribution design of a gas turbine combustor that burns low-Btu gasified fuel

Fig. 6. Design concept of 1773K-class low-Btu fueled combustor

Burner   Combustion Liner

Fig. 7. Tested combustor

Figure 6 presents characteristics of the designed and tested 1773K-class combustor. Figure 7 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.

3.1.1.1 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.

3.1.1.2 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:

*3.1.1.2.1 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 8 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.

$$S = \frac{2}{3} \times \frac{1-B^3}{1-B^2} \times \tan\theta_a \tag{1}$$

Where B (boss ratio of swirl vane)=0.66.
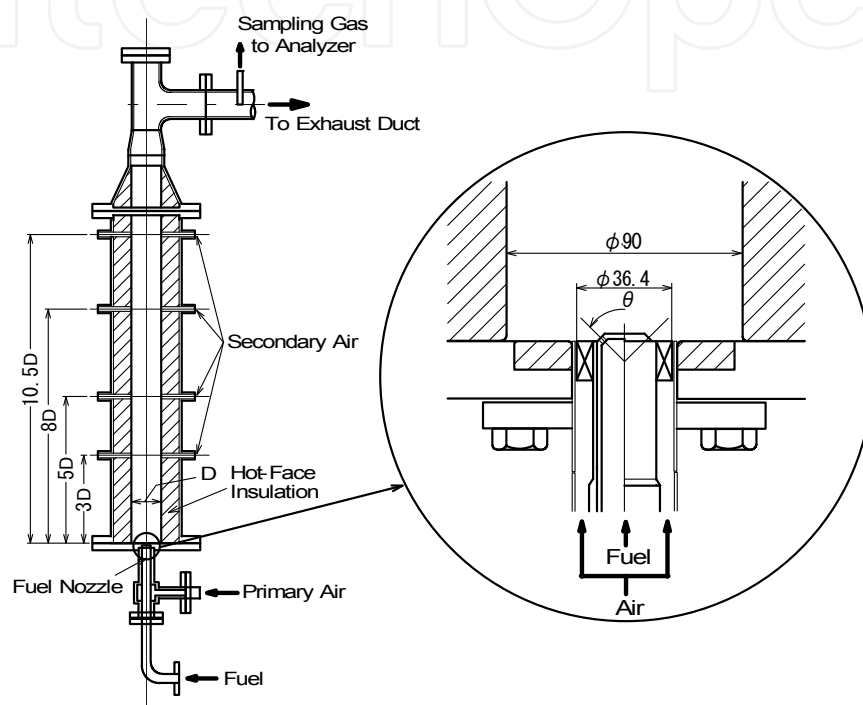
D : inner diameter of cylinder-style combustion chamber, 90mm
θ : injection angle of fuel nozzle, 90 degrees

Fig. 8. Combustion chamber and diffusion burner of basic experimental device

Figure 9 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.

$$C.R. = \frac{\left([\text{NOx}] - [\text{NOx}_{th}]\right) \times \left(\text{volume flow rate of exhaust}\right)}{[\text{NH}_3] \times \left(\text{volume flow rate of fuel}\right)} \tag{2}$$

To obtain the conversion rate of NH_3 to NOx, the concentration of thermal-NOx, '[NOxth]', 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.
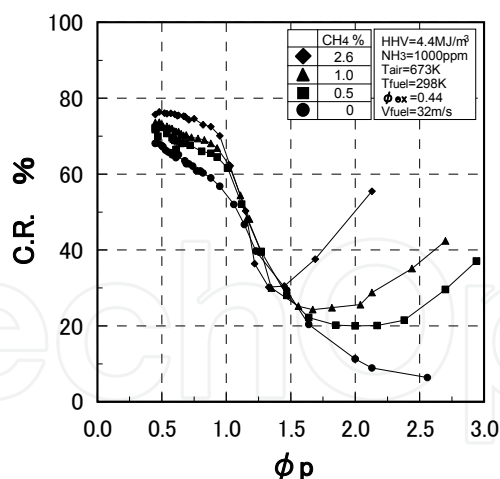
Fig. 9. Effect of methane content on conversion rate of ammonia in the fuel to NOx, defining by the experiments using a small diffusion burner (Hasegawa et al., 2001)

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. 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. Since the 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 primary combustion-air are injected from the burner, which has 30 degree swirl angle and 15 degree introvert angle.

3.1.1.2.2 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 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.

3.1.1.2.3 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.

3.1.2 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. The standard rated conditions in the combustion tests are summarized in Table 3. Combustion Intensity at the design point is 2.0×10^2 W/(m³•Pa).

T_{air}	700K
T_{fuel}	633K
T_{ex}	1773K
P	1.4MPa
ϕ_{ex}	0.62
Combustion Intensity	2.0×10^2 W/(m³•Pa)

Table 3. Rated test conditions

3.1.2.1 Combustion emission characteristics

Figure 10 shows the combustion emission characteristics, under the gas turbine operational conditions. When the gas turbine load was 25 percent or higher, which is the single fuel firing of gasified fuel, the conversion rate of NH₃ to NOx was reduced as low as 40 percent (NOx emissions corrected at 16 percent O₂ was 60ppm), while the combustion efficiency shows around 100 percent in each gas turbine load.
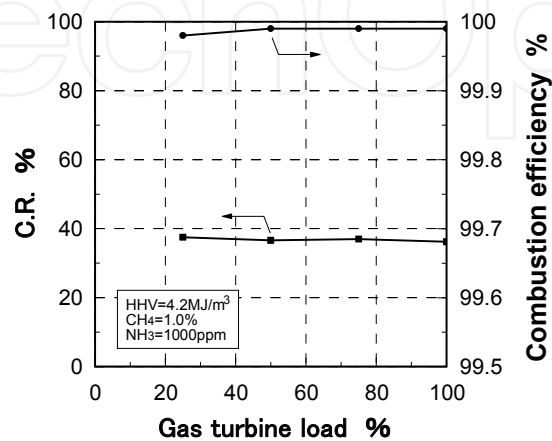

Fig. 10. Combustion emission characteristics

3.1.2.2 Thermal characteristics of combustor liner wall

Figure 11 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°C), 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.
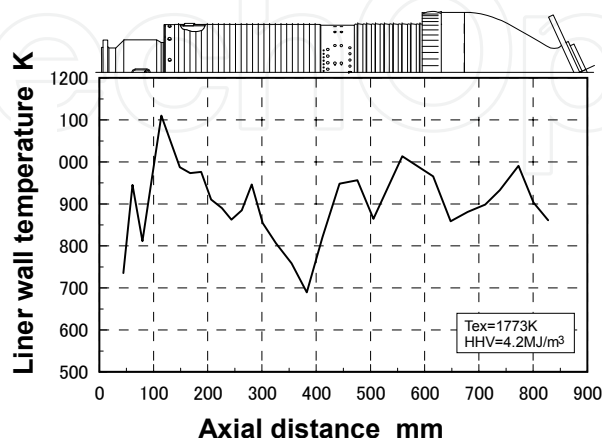

Fig. 11. Combustor wall temperature distribution

3.2 Combustor for oxygen-blown gasification system with wet type synthetic gas cleanup

3.2.1 Subjects of combustor

In the case of oxygen-blown IGCC, which has an air-separation unit to produce oxygen as gasification agent, medium-Btu gasified fuels are produced compared with the case of the air-blown gasified low-Btu fuels. That is, the maximum flame temperature of medium-Btu fuel is higher than that of each low-Btu fuel or high-calorie gas such as natural gas. 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 raw material such as coal into the gasifier and so on, gasified 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 fuel before injection into the combustor.

3.2.2 Design concept of combustor

Figure 12 presents characteristics of the designed, medium-Btu fueled 1573K (1300°C)-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.
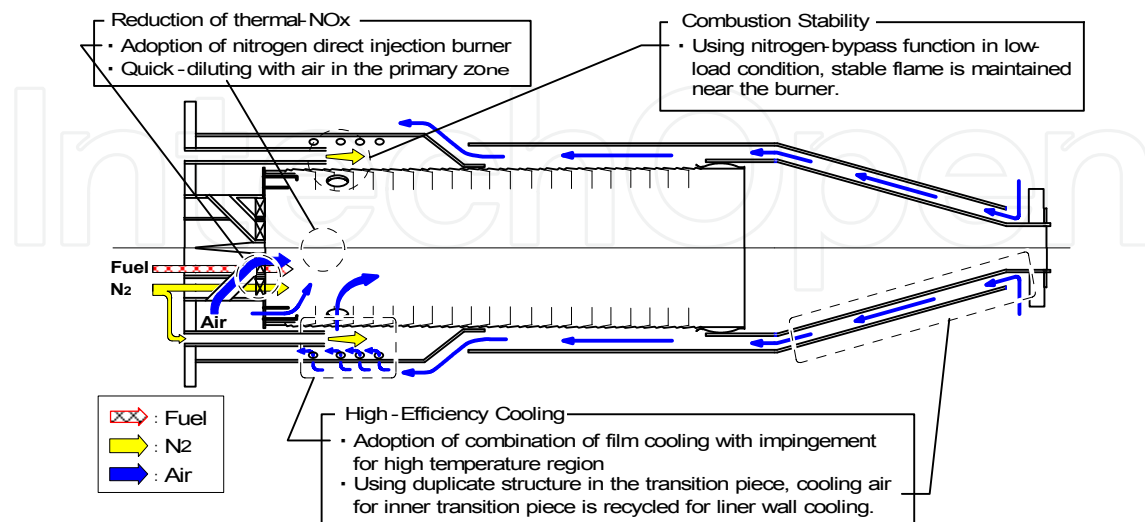

Fig. 12. Design concepts of medium-Btu fueled combustor for wet-type synthetic gas cleanup

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 13 presents an example of the test results using the small diffusion burner shown in figure 8, 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 5.1MJ/m^3. From figure 13, 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.
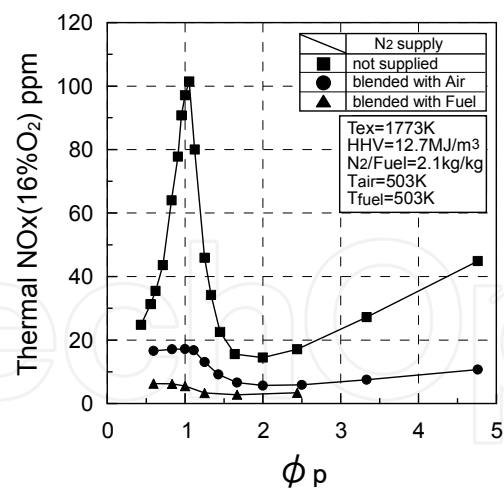
Fig. 13. Effect of nitrogen injection on thermal-NOx emission characteristics in two-stage combustion, using a small diffusion burner

Figure 14 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 14, 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.

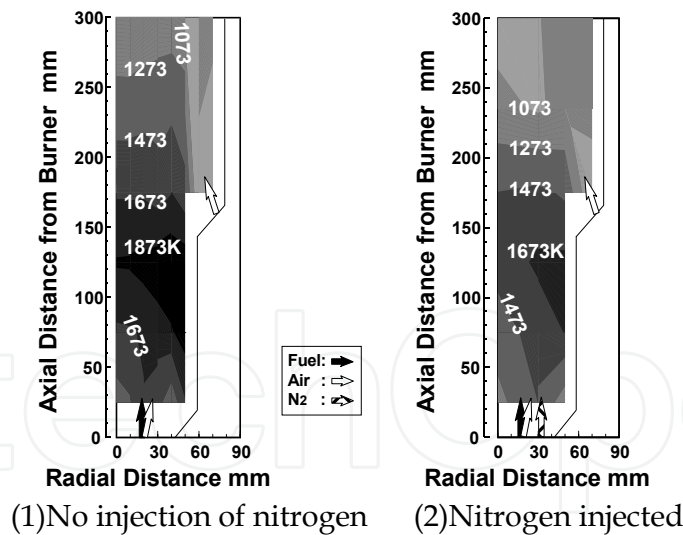(1)No injection of nitrogen   (2)Nitrogen injected

Fig. 14. Effect of nitrogen injection on combustion gas temperature distribution using a model combustor

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 12. 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.

3.2.3 Test results

Table 4 and 5 show the typical properties of the supplied fuel and the standard test conditions, respectively. Higher heating value of the supplied fuel, HHV, 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 commercial gasified fuel and temperature of supplied nitrogen is lower in the case of the test conditions than in the actual operations, the combustor performance is investigated 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, T_{ex}, is around 1700K.

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 4. Typical conditions of supplied fuel

T_{air}	603K
T_{fuel}	583K
T_{N_2}	333K
T_{ex}	1700K
N$_2$/Fuel ratio	0.3kg/kg
P	1.4MPa
Combustion Intensity	2.2×10^2 W/(m^3•Pa)

Table 5. Rated test conditions

Figure 15 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 1.0MPa of slightly less than that of the practical operation at the equivalent, rated load. When the gas turbine load was 25 percent or higher, which is the single fuel firing of gasified 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.
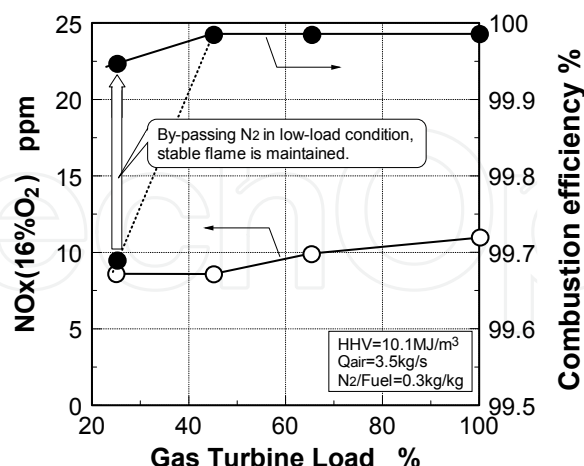

Fig. 15. Effect of the gas turbine load on combustion emission characteristics

3.3 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.

3.3.1 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.

3.3.2 Design concept of combustor

Figure 16 presents the configuration and its function of a designed, medium-Btu fueled 1773K (1500°C)-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.
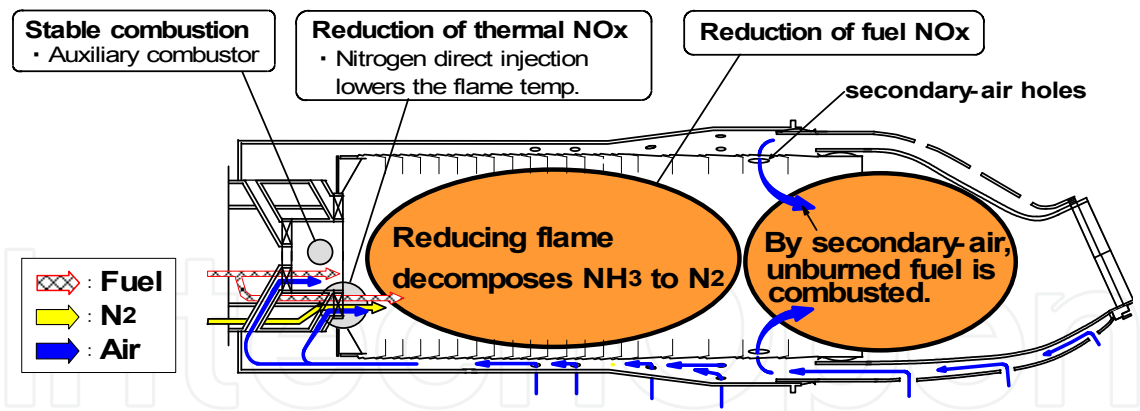
Fig. 16. Design concept of a medium-Btu fueled gas turbine combustor for hot/dry-type synthetic gas cleanup

Figure 17 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 following characteristics.
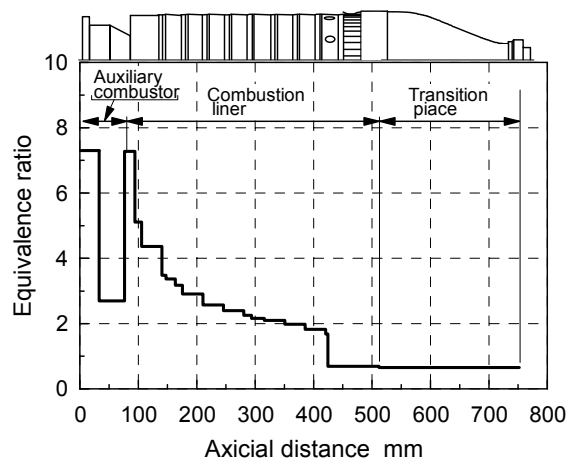

Fig. 17. Axial distribution of equivalence ratio at the rated load condition

3.3.2.1 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 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.

3.3.2.2 Nitrogen injection

From figure 13, 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 the primary equivalence ratio, 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 18 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 13 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 18, 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.
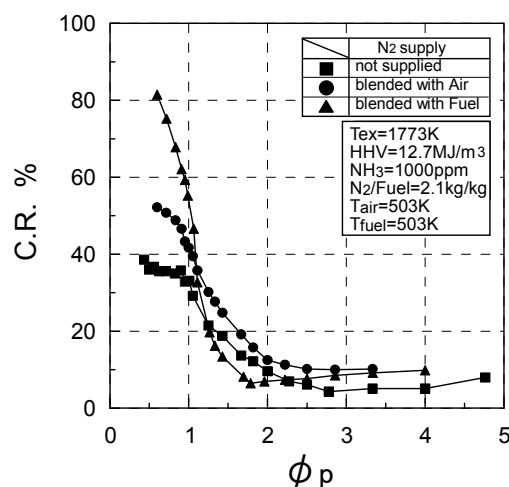

Fig. 18. Effect of nitrogen injection on the conversion rate of NH_3 to NOx in two-stage combustion, using a small diffusion burner

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.

3.3.2.3 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 8. Figure 19 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.
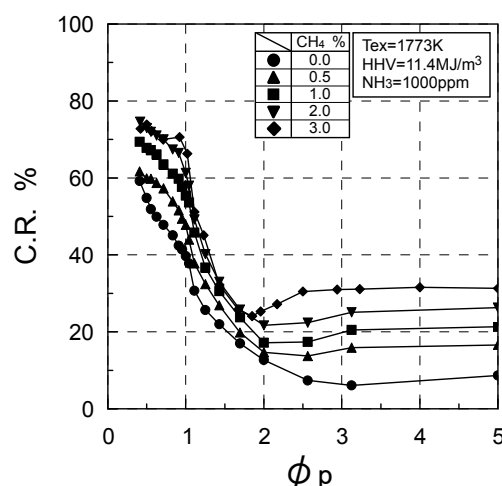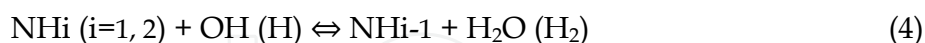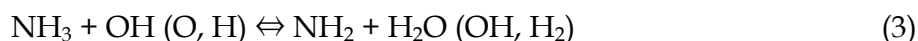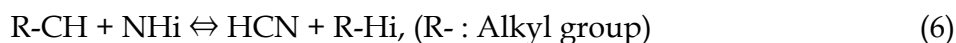

Fig. 19. Effect of the CH_4 concentration on conversion rate of NH_3 to NOx in two-stage combustion of medium-Btu fuel

The effect of the CH_4 concentration on the fuel-NOx produced by NH_3 in gasified 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 in the gasified fuels (Hasegawa, 1998c) has been confirmed by comparison with test results.
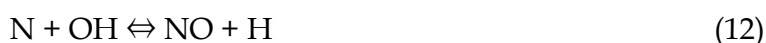
The nitrogen of NH_3 in the fuel has weaker bonding power than N_2. In the combustion process, NH_3 reacted with the OH, O, and H radicals and then easily decomposed into the intermediate NHi by the following reactions (Miller et al., 1983).

$$NH_3 + OH \text{ (O, H)} \Leftrightarrow NH_2 + H_2O \text{ (OH, } H_2) \tag{3}$$

$$NHi \text{ (i=1, 2)} + OH \text{ (H)} \Leftrightarrow NHi\text{-}1 + H_2O \text{ (}H_2) \tag{4}$$

When hydrocarbon is not contained in the fuel, NHi 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) and (Takagi et al, 1979).

$$CHi \text{ (i=1,2)} + N_2 \Leftrightarrow HCN + NHi\text{-}1 \tag{5}$$

$$R\text{-}CH + NHi \Leftrightarrow HCN + R\text{-}Hi, \text{ (R- : Alkyl group)} \tag{6}$$

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 fuel, the HCN increases, and NOx emissions originated from HCN in the fuel-lean secondary combustion zone increase.

$$HCN + OH \Leftrightarrow CN + H_2O \tag{7}$$

$$CN + O_2 \Leftrightarrow CO + NO \tag{8}$$

$$CN + O \Leftrightarrow CO + N \tag{9}$$

$$NH + OH \Leftrightarrow N + H_2O \tag{10}$$

$$N + O_2 \Leftrightarrow NO + O \tag{11}$$

$$N + OH \Leftrightarrow NO + H \tag{12}$$

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.

$$NO + NH \Leftrightarrow N_2 + OH \tag{13}$$

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) 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.

3.3.3 Test results

Supplied fuels into the combustor were adjusted as same propertied as that of the slurry-feed coal gasified fuel. In tests, the effects of the CH_4 concentrations, etc. in the supplied fuels on the combustion characteristics were investigated and the combustor's performances were predicted in the typical commercial operations. Figure 20 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 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.
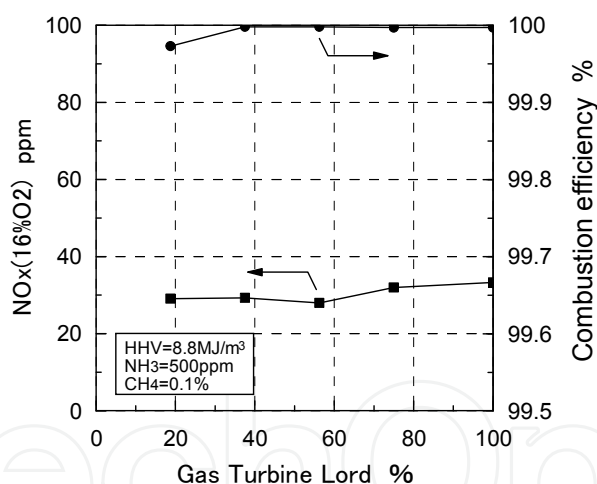

Fig. 20. Combustion emission characteristics

4. Conclusion

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 6, 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 turbine combustor.

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

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

* : Concentration corrected at 16% oxygen in exhaust

Table 6. Performances of gasified fueled combustors

5. Acknowledgment

6. Nomenclature

CO/H_2	Molar ratio of carbon monoxide to hydrogen in fuel [mol/mol]
C.R.	Conversion rate from ammonia in fuel to NOx [%]
HHV	Higher heating value of fuel at 273 K, 0.1 MPa basis [MJ/m³]
$N_2/Fuel$	Nitrogen over fuel supply ratio [kg/kg]
$NOx(16\%O_2)$	NOx emissions corrected at 16% oxygen in exhaust [ppm]
P	Pressure inside the combustor [MPa]
T_{air}	Temperature of supplied air [K]
T_{ex}	Average temperature of combustor exhaust gas [K]
T_{fuel}	Temperature of supplied fuel [K]
T_{N_2}	Temperature of supplied nitrogen [K]
ϕ_{ex}	Average equivalence ratio at combustor exhaust
ϕ_p	Average equivalence ratio in primary combustion zone

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Advances in Gas Turbine Technology

Edited by Dr. Ernesto Benini

Gas turbine engines will still represent a key technology in the next 20-year energy scenarios, either in stand-alone applications or in combination with other power generation equipment. This book intends in fact to provide an updated picture as well as a perspective vision of some of the major improvements that characterize the gas turbine technology in different applications, from marine and aircraft propulsion to industrial and stationary power generation. Therefore, the target audience for it involves design, analyst, materials and maintenance engineers. Also manufacturers, researchers and scientists will benefit from the timely and accurate information provided in this volume. The book is organized into five main sections including 21 chapters overall: (I) Aero and Marine Gas Turbines, (II) Gas Turbine Systems, (III) Heat Transfer, (IV) Combustion and (V) Materials and Fabrication.

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