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Design for F Class Blast Furnace Gas Firing 300 MW Gas Turbine Combined Cycle Plant

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ABSTRACT

In the past natural gas was used as the main fuel for heavy-duty type gas turbine. Because, the natural gas was the best fuel considering cost and availability.

Because recently the cost of natural gas presents an increasing trend, and its future availability is not very clear, there is a clear trend to use alternative fuels. Therefore, gas turbines should be capable of burning a variety of gas fuels including low heating value gases (e.g., Synthetic gas and steel mill gas), landfill gas and others. Compared with natural gas, these fuels vary in hydrocarbon composition, physical properties, impurities and others. In order to use these fuel gases in the gas turbines, advanced technology modifications to the combustor and combustion system are necessary.

Now, Mitsubishi Heavy Industries, Ltd. (MHI) designs and installs F Class Blast Furnace Gas Firing Combined Cycle Plant for the higher thermal efficiency by the adoption of higher firing temperature technology.

This paper introduces a recent application of F Class gas turbine firing Low Calorific gas as BFG and MHI design concepts for gas turbines utilizing by product gas in steel works.

1. INTRODUCTION

There is growing interest in saving energy by applying modern, high firing temperature gas turbines in electric utility and industrial facilities.

Japanese industries in particular want to use their by-product gas to best effect, and hereby reduce energy costs. Because, the energy source such as coal, gas and oil is the limited in Japan. Therefore, the energy cost of Japan is more expensive than the other countries.

In order to meet this requirement of Japanese industries, has been continuously developing a high efficiency gas turbine, embodying the state of the art in industrial gas turbine high-temperature turbine technology.

In recent steel industry field, the energy conservation has been expedited and the less oil blast furnace operation has been achieved. As a result, the balance of energy source in steel work has considerably altered. In this situation, the gas turbine combined cycle power plant is drawing attention as a high thermal efficiency power generation system, utilizing increased by-product gas effectively.

Under such circumstances, the design and development work are continuously conducting in MHI since 1950, based on our extensive experience with the by product gas firing technology in order to develop highly efficient, blast furnace gas (hereinafter abbreviated as BFG) firing combustors.

Based on the successful research & development and existing units we delivered the first unit of Low Calorific Heat (1,050 kcal/m^{3}N) BGF firing F Class gas turbine (1,300°C class) to Kimitsu Joint Thermal Power Co,. It is under erection, which will be put into commercial operation in 2004. The introduction of gas turbine will remarkably improve the overall power generation efficiency of the power company.

This paper outlines the experiences of the high-efficiency gas turbine combined cycle power plant, firing BFG in the steel works, and technical key points in the plan of new power plant.

2. SUMMARY OF FUEL GAS APPLICICATION

As of February 2003 MHI has gotten orders for a total of 435 units. As can be seen in Table 1, these gas turbines are designed to burn a variety of fuels.

Table 1. Mitsubishi gas turbine fuel applications
(as of February 2003)

Fuel	
1. Single Fuel	
(1) Natural Gas, LNG	176
(2) Refinery Process Gas, LPG	10
(3) Steel Mill Gas	9
(4) Distillate Oil	63
2. Dual Fuel	
(1) Natural Gas /Distillate Oil	146
(2) Natural Gas/Crude oil	6
(3) Refinery Process Gas/Distillate Oil	13
(4) LPG/Distillate Oil	
(5) Steel mill Gas/Distillate Oil	6
(6) Mine Gas/Distillate Oil	3
(7) Synthetic Gas/Distillate Oil	1
	2
Total	435

Figue.1 shows our fuel gas experience depending on the fuel gas heating value. Our gas turbines have operated successfully burning various fuel gases with heating value ranging from 600 kcal/m^{3}N to 20,000 kcal/m^{3}N. In particular, we have successfully operating experience in Low heating value gas, such as less than 1,000 kcal/m^{3}N.

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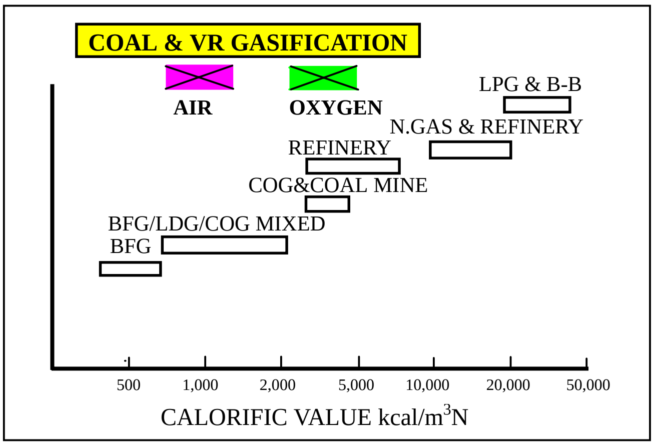


Fig.1 Mitsubishi gas turbine gas fuel experience

3. DEVELOPMENT AND HISTORY OF BFG FIRING GAS TURBINES

The first BFG firing gas turbine (850 kW) in Japan, of our unique design, was developed in 1958, and delivered as a prime mover for blast furnace blower to Yawata Steel Corp. (present Nippon Steel Corp.) Yawata Works. NO. 2 larger unit (4000 kW) was delivered in 1964 to the same Works. In 1965, we delivered the model MW171 gas turbine (15000 kW) to Sumitomo Metal Industries, Ltd. Wakayama Works.

In Europe, about 30 BFG firing gas turbine power plants were reportedly constructed from 1950 to 1965. The inlet temperature of these gas turbines seem to be about 750°C and regenerative cycle is adopted to most of them for the improvement of thermal efficiency. After then, the thermal efficiency of the gas turbine itself has been increased by raising the gas temperature and improving the efficiency of components.

Further, the energy recovery efficiency at the waste heat boiler is improved because of increase of the gas temperature. Accordingly, the overall thermal efficiency of the gas turbine combined cycle power plant was markedly improved. In 1982, Mitsubishi high efficiency model M151 gas turbine was delivered to Nippon Steel Corp. Kamaishi Works, with exceeding 1000°C of inlet temperature.

After developed M151, we have started to increase the output more than 100 MW with D Class gas turbine (1,150°C Class) depending in the many combined cycle plant operations.

MHI, In 1987, delivered a large capacity (145 MW) high efficiency gas turbine combined cycle power plant to the Chiba Works of Kawasaki Steel Corporation. This power plant uses low caloric by-product gas generated within the Chiba Works and has obtained a world record plant thermal efficiency of 46% (LHV base, net).

According to this 145 MW large capacity combined cycle plant operating experience, we have delivered the similar concept plant to Mizushima Joint Thermal Power Co., and Fukuyama Joint Thermal Power Co., which is 100°C higher gas turbine inlet temperature as DA Class, in 1994 and 1995.

And we delivered the first export unit of BFG firing gas turbine combined cycle plant to UNA, Netherland in 1997 depending on the successful operating records of above units. Table 2 shows the above experience and Fig. 2 shows the trend of gas turbine inlet temperature in BFG firing.

Table 2. Experience list of low heating value gas firing gas turbine

Customer	Application	Start-Up	Model	GT Unit Rating (Incl.G.C)	Combined Plant Rating	Fuel		Combustor
						Main	Stand-By	
NSC (Nippon Steel Co.) Yahata Works	Power Supply	1958	-	850 kW	-	BFG (770kcal/Nm^3)	-	Single-can
NSC Yahata Works	Power Supply	1964	-	4,000 kW	-	BFG (770kcal/Nm^3)	-	Single-can
Sumitomo Metal Co. Wakayama Works	Co-generation (WHB,G-M,BLOWER)	1965	MW171	15,000 kW	-	BFG (750kcal/Nm^3)	-	Single-can
Shikoku Elec.Pwr Co. Sakaide PS	Combined cycle	1970	MW301	34,000 kW	16,000 kW	COG (750kcal/Nm^3)	-	Multi-can
Mitsubishi Coal Mining Co. Minami Oyubari Plant	Co-generation (with air-preheater)	1970	MW101	9,000 kW	-	Coal mine (4,700kcal/Nm^3)	OIL	Multi-can
NSC Kamaishi Works	Combined cycle with existing STs	1982	M151	16,000 kW	23,000 kW	BFG (670kcal/Nm^3)	OIL	Single-can
Kawasaki Steel Co. Chiba Works	Combined cycle (Single Shaft)	1987	M701	87,400 kW	145,000 kW	BFG/COG (1,000kcal/Nm^3)	-	Multi-can with BP-V
Mitsubishi Gas-Chemical Co. Mizushima-factory	Co-generation (WHB)	1988	MF111	16,250 kW	-	BFG/COG (2,400kcal/Nm^3)	OIL	Multi-can
NSC Hirohata Works	Co-generation (WHB)	1989	M251	30,200 kW	-	LDG (1,815kcal/Nm^3)	OIL	Multi-can
Nissin Steel Co. Kure Works	Combined cycle with existing STs	1989	M251	32,000 kW	50,000 kW	BFG (700kcal/Nm^3)	-	Multi-can with BP-V
Nakayama Steel Co. Funamachi Works	Combined cycle (Single Shaft)	1991	M151	15,000 kW	37,000 kW	BFG/LDG (1,00kcal/Nm^3)	-	Multi-can with BP-V
Mizushima Joint Thermal Power Co.	Combined cycle (Single Shaft)	1994	M501	86,250 kW	145,000 kW	BFG/M (965kcal/Nm^3)	-	Multi-can with BP-V
Fukuyana Joint Thermal Power Co.	Combined cycle (Single Shaft)	1995	M501	86,250 kW	145,000 kW	BFG/M (965kcal/Nm^3)	-	Multi-can with BP-V
UNA Netherlands	Combined cycle (Single Shaft)	1997	M701	87,400 kW	145,000 kW	BFG/COG (1,000kcal/Nm^3)	-	Multi-can with BP-V
NSC Ooita Works	Combined cycle	2001	M251	31,000 kW	65,000 kW	BFG (700kcal/Nm^3)	-	Multi-can with BP-V
Nippon Petroleum Refining Co. Ltd. Negishi Refinery	Combined cycle (Single Shaft)	2003	M701F	295,000 kW	430,000 kW	Syn gas (2,680kcal/Nm3)	OIL	Multi-can with BP-V
Kimitsu Joint Thermal Power Co.	Combined cycle (Single Shaft)	2004	M701F	180,700kW	300,000 kW	BFG/COG (1,050kcal/Nm^3)	-	Multi-can with BP-V

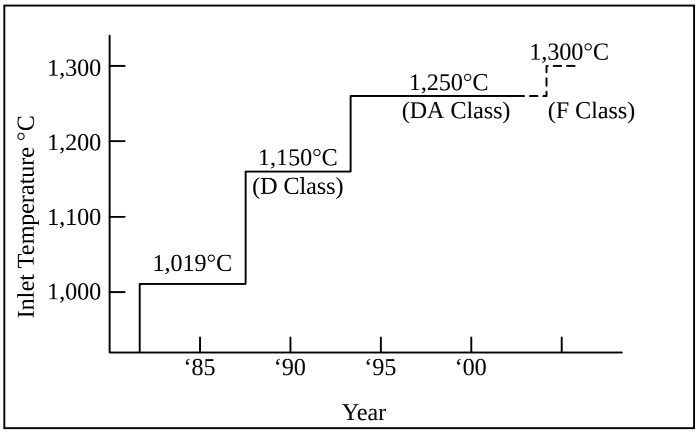


Fig.2 Trend of Mitsubishi BFG firing gas turbine inlet temperature

In Kimitsu Joint Power Co., a newly developed Mitsubishi model M701F gas turbine, firing BFG, has been installing for the highest thermal efficiency in the BFG firing power plant. The model M701F gas turbine is designed for firing BFG with an inlet temperature exceeding 1,300°C. It is the first gas turbine firing BFG for F Class, in the world.

4. DESIGN CONSIDERATION ON BFG FIRING GAS TURBINE

Comparing natural gas as clean fuel, BFG has the special characteristics such as lower heating value, impurities contents and so, on.

Table 3 shows the fuel gas characteristics comparison

Table 3. MHI Typical fuel characteristic table

FUEL		LNG	BFG	Mixed BFG	Oxygen Blown Syngas	Air Blown Syngas
COMPOSITION (%) Vol.	CH_4	88.0	—	2.02	0.21	2.9
	C_2H_6	7.11	—	0.24	—	—
	C_3H_6	3.58	—	—	—	—
	C_4H_{10}	1.24	—	—	—	—
	C_5H_{12}	0.05	—	—	—	—
	N_2	0.02	55.4	51.02	0.51	30.7
	CO	—	22.4	20.88	50.07	11.0
	CO_2	—	20.4	19.78	3.21	10.9
	H_2	—	2.0	6.05	44.66	16.8
	Ar	—	—	0.01	1.03	0.80
LHV ($Kcal/m^3N$)		9,762	706	1,000	2,680	1,020
Flammability Range (—)		3.2	2.1	4.2	25.8	9.8
Burning Velocity (cm/sec)		37	3	16	169	44

The standard gas turbine is designed for natural gas (i.e., LNG). If the fuel properties as BFG are different from standard natural gas, suitable modification of the fuel control system, supply system, combustor etc. will be required. Depending on the long term operating experience, MHI has already established the key technologies for BFG utilizing. Table 4 shows the summary of them.

Table 4. Development technology for BFG firing gas turbine

BFG CHARACTERISTIC	KEY POINT	TECHNOLOGY
· NARROWED FLAMMABILITY ZONE · LOW BURNING VELOCITY	· CAREFULLY DETERMINE THE AIR RATIO OF FUEL NOZZLE · KEEP THE HIGH BURNING EFFICIENCY · COMBUSTOR SECTION AREA	· MULTI CAN TYPE COMBUSTOR WITH AIR BYPASS VALVE
· LOW CALORIFIC VALUE	· LARGE CAPACITY GAS SUPPLY SYSTEM (GAS COMPRESSOR)	· HIGH EFFICIENCY AXIAL FLOW COMPRESSOR WITH VARIABLE PITCH VANE
· DIRTY DUST	· HIGH EFFICIENCY DUST REMOVAL · REMOVE THE DEPOSITION THE G.C. BLADE	· WET TYPE E.P. · COMPRESSOR CLEANING SYSTEM
· TOXIC	· NOT EXHAUST BFG TO OUTSIDE	· GAS COOLER · ADVANCED SHAFT SEAL

The key technology is the best matching design for compressor, combustor and turbine to maintain the stable operation in low heating value. The above design considerations are mainly described here in after.

4.1 Heating value

The heating value is the key parameter to decide the modification of gas turbine components. In the case of decreasing heating value, a modification will be necessary. Our design modification concepts are shown on Table.5.

Table 5. Design modification for various heating value

Heating Value	High 10,000～ 25,000kcal/m^3N	Standard (Natural Gas) 8,500 kcal/m^3N	Medium 2,000～ 7,000kcal/m^3N	Low 600～ 2,000kcal/m^3N
Air Compressor	Standard	Standard	Standard	Modification
Combustor	Standard (Minor mod.)	Standard	Standard (Minor mod.)	Modification
Turbine	Standard	Standard	Standard	Standard
Fuel system	Standard (Minor mod.)	Standard	Standard (Minor mod.)	Modification

To maintain the reliability and the hot component changeability, the identical turbine parts are applied for the various heating values. However, the other parts will be modified depending on the heating value.
For your reference, we explain the modifications introducing to BFG.

1) Air Compressor

Huge fuel gas flow is fed to gas turbine when BFG firing since it's heating valve is lower. Therefore, if the standard gas turbine as natural gas is applied, the surge problem on air compressor and over load on turbine will occur. So, to maintain the same gas flow on turbine, the air compressor is modified to decrease the air flow by adjusting the high of compressor blades (tip cut).

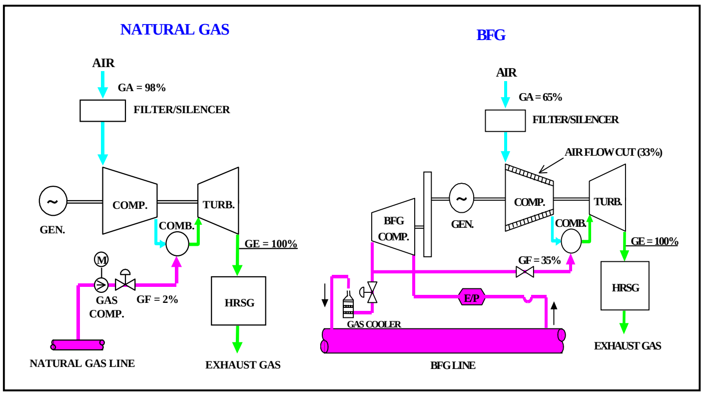


Fig.3 Flow balance vs. heating value

The above air compressor modification is applied in the existing operating units. On the other hand, there is air bleed system to correspond to BFG firing shown on Fig.4.

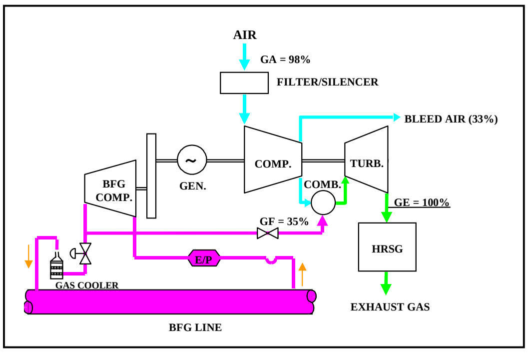


Fig.4 Bleed air system on BFG firing gas turbine

In case of air bleed system, the air compressor parts are not necessary to be modified. However, the performance is worse than the airflow cut modification shown on Fig.3 during the normal BFG firing mode, if there is no chance to utilize the bleed air.

2) Combustor

When burning BFG for gas turbine, the silo type combustor with pilot-torch is the suitable from the point of view of stable combustion only. However, considering the total evaluation of gas turbine including the reliability of turbine blades and others, the multi-cannular type combustor is the best type.
This combustor design with new concept is focused from the following points;

“Large amount of the air must be supplied for the combustion because of its substantial lower heating value and this gives the disadvantage for the control of the fuel to air ratio. Stable and high efficient combustion is required within the turndown ratio 2.5 in the gas turbine combustors.
The disadvantage for the combustor basket cooling because only a less air is available for cooling.”

To solve the above, multi-cannular combustor design is selected because of the smaller combustor basket surface area are available compared with the large silo type combustor design.

The specially designed variable geometry bypass valve is applied to compensate the air flow supplied to the combustion area Combustor configuration is illustrated on Fig.5.
Variable geometry bypass valve is equipped on the transition piece and adjusting the valve openings can regulate airflow supplied to the combustion area.

Prior to the combustor detail design, joint combustion development rig tests were carried out befitting the actual blast furnace/coke oven gas in the Steel Work. (Fig.6)

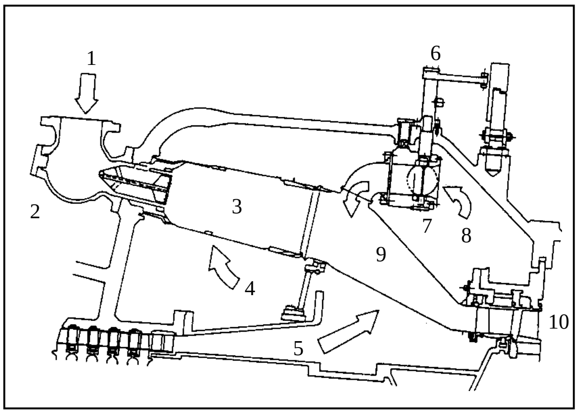


Fig.5 Combustor configuration

1. BFG+COG Fuel gas	6. Variable ring
2. Spherical elbow	7. Bypass valve
3. Combustor	8. Bypass air
4. Air for combustion	9. Transition Piece
5. Compressor discharge air	10. Turbine

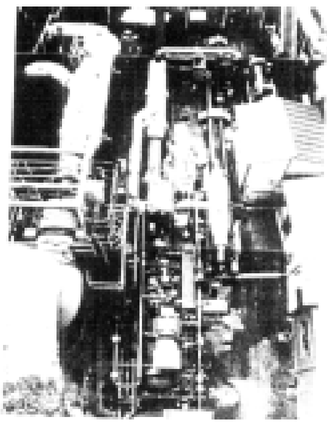


Fig.6 Combustor test rig

Rig test results are summarized as follows;

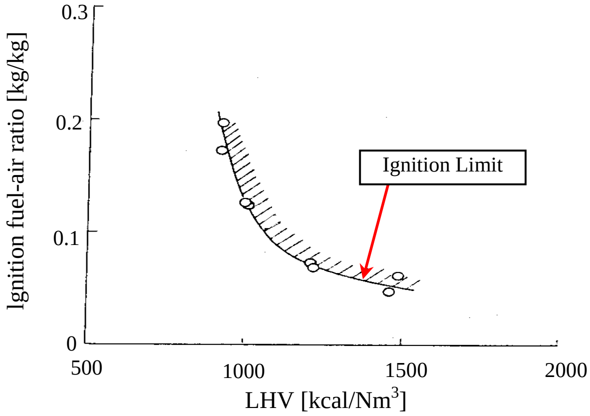


Fig.7 Critical fuel to air ratio for the ignition

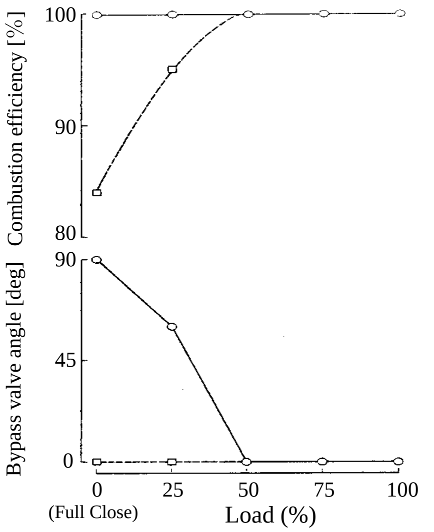


Fig.8 Combustion efficiency improvement using bypass valve

(i) Direct Ignition by the spark type igniter under the heating value condition higher than 900kcal/m^{3}N (3768kj/m^{3}N) can be achieved and the critical fuel to air ratio for the ignition was obtained as shown on Fig.7.
(ii) Stable and high efficient combustion can be obtained under the expected operating fuel to air ratio including no-load condition and the full load condition.
Under the part load condition, the variable geometry bypass valve improved the combustion efficiency as shown on Fig.8 .

3) Fuel Gas Supply System

According to the huge of fuel gas flow, the size of piping and valve are larger than natural gas.

To minimize the piping force to gas turbine proper, the size of piping should be reduced. However, the fuel gas pressure loss is increased in response to the smaller sizing. This pressure loss is the worse effect on the plant performance. Therefore, the suitable sizing should be considered for the total benefit. Depending on Low heating value gas firing experience, Mitsubishi could optimize this evaluation. Fig.9 shows the large size fuel gas supply system for reference.

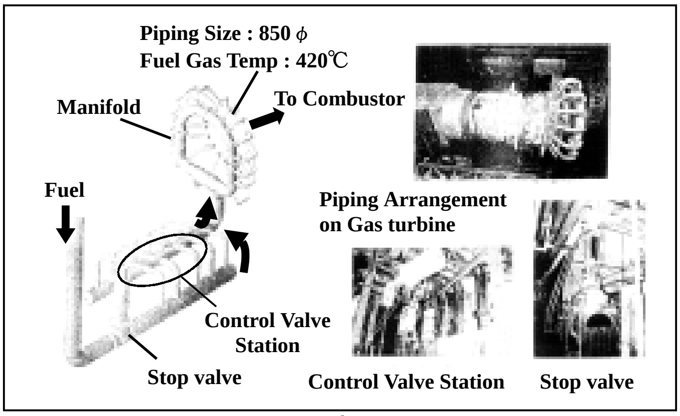


Figure.9 Fuel gas supply for low heating value gas

4.2 Heating value fluctuation

There are many items to be considered for design of combustion system. Relating to a fuel gas heating value, fluctuation of its heating value change a required pressure drop through the fuel-nozzle. This change must be maintained within allowable limit for stable combustion. For example, the combustor has its own characteristic frequency of acoustic waves established by its geometry or others and these phenomenon is shown as a spectrum of the dynamic pressure. When the pressure at inside of the combustor near the fuel-nozzle rises, the fuel flow through the fuel nozzle is reduced due to lower fuel-nozzle pressure drop.

Conversely a decrease in the combustor pressure makes a increase in the fuel flow. Thus the fuel flow may fluctuate and oscillate following the pressure frequency of combustor.

In case of the low fuel-nozzle pressure drop relative to the pressure frequency of combustor, the amplified pressure oscillation may occur in combustion system. An excess pressure oscillation may cause damage to the combustor component.

Considering the above phenomenon, our standard acceptable variation of Gas Index is within ±15% (Wobbe Index base) based on our experience. Please refer to the attached Fig.10.

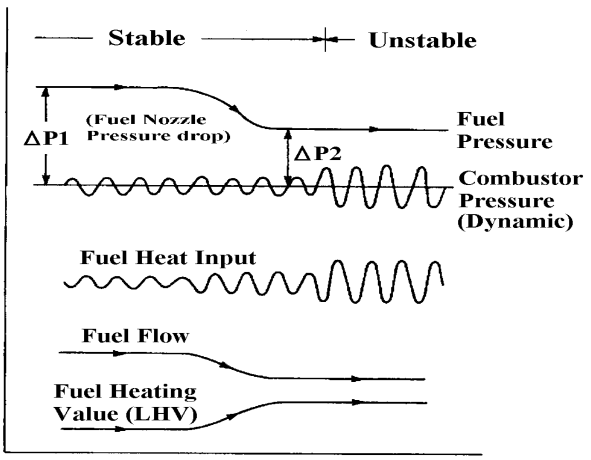
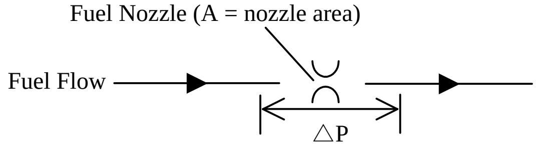


Fig.10 Schematic chart of dynamic phenomenon

Influence of Gas Index (Wobbe Index) to the fuel flow is as follows ;



Fuel flow (Q) $\quad Q \propto A \times \sqrt{\dfrac{\Delta P}{\gamma}}$

Fuel heat input (I) $\quad I \propto Q \times LHV \propto A \times \dfrac{LHV}{\sqrt{\gamma}} \times \sqrt{\Delta P}$

Gas Index (GI) $\quad GI = \dfrac{LHV}{\sqrt{\gamma}}$

Table 6 shows the effect of Gas Index on heat input and nozzle pressure drop. In case that Gas Index fluctuation is ±15%, heat input fluctuation is ±15% under nozzle pressure is constant, while nozzle pressure drop fluctuation is −17%～+23% under heat input is constant.

Table 6. Gas index vs. heat input and nozzle pressure drop

Gas Index	Design Point	±10%	±15%	±20%
A. ΔP= constant → Heat input variation Heat input $\propto GI \times \sqrt{\Delta P}$	Base	±10%	±15%	±20%
B. Heat input = constant → Nozzle pressure drop variation (ΔP) $\Delta P \propto \dfrac{1}{GI^2}$	Base	−17% ～ +23%	−24% ～ +39%	−31% ～ +56%

4.3 Contaminants in gas fuel

In the gas turbine, the tracemetal such as vanadium, sodium, potassium and others can combine with other element in the hot gas part to form low melting point compounds, which cause severe corrosion of turbine hot section parts. An example is vanadium pentoxide which melts at 690℃.

And also, these melting compounds can read to deposits on turbine blades which causes the plugging problem on air cooling holes. The compounds and deposits mechanism is shown on Fig.11.

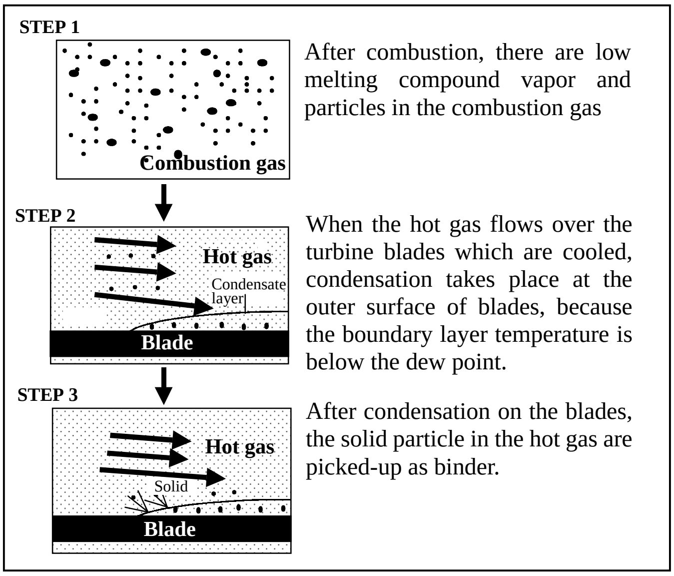


Fig.11 Compounds and deposits mechanism

Generally, the allowable limits of trace metal in the low calorific fuel gas as BFG should be lower than the high calorific fuel such as natural gas and liquid fuel, because the fuel flow quantity of low calorific fuel gas is larger at the same rating. And, the trace metal contaminants will also come from nonfuel sources such as ambient air, injection water and steam. Therefore, the total contaminants in the fuel and nonfuel have to be limited as follows;

Trace metal	Equivalent Limit (ppm weight)
Sodium plus Potassium	0.5 max.
Vanadium	0.5 max.
Calcium	10 max.
Lead	2 max.
Others total	2 max.

The following relationship will be used to calculate the above equivalent limit to each contaminant in the fuel and nonfuel.

Where;

- LHV_O = Standard lower heating value (≒ 10,000 kcal/kg)
- LHV_F = Lower heating value of actual fuel gas (kcal/kg)
- A/F = Air - fuel mass flow ratio
- S/F = Steam - fuel mass flow ratio
- W/F = Water - fuel mass flow ratio
- X_F = Contaminant concentration in fuel (ppm - weight)
- X_A = Contaminant concentration in air (ppm - weight)
- X_S = Contaminant concentration in injected steam (ppm -weight)
- X_W = Contaminant concentration in injected water

If there is no contaminants in nonfuel, the allowable trace metal limit in the fuel corresponding to the heating value is as follows;

Table 7. Example of allowable trace metal in low heating value

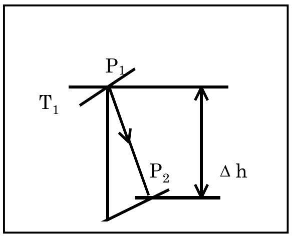
	Lower Heating Value 1,000 kcal/kg
Sodium plus Potassium	0.05 ppm weight
Vanadium	0.05 ppm weight
Calcium	1 ppm weight
Lead	0.2 ppm weight
Others metal	0.2 ppm weight

4.4 Performance impact

Comparing natural gas firing gas turbine combined cycle plant, thermal efficiency of BFG firing is reduced. The approx. 10 % as relative is lower for combined cycle plant base and 15% as relative is lower for gas turbine base.

The above efficiency drop could not be recovered since the thermodynamic theory of BFG firing gas turbine is different from natural gas firing. Main factors for the above is the compressor power and turbine output.

First, we explain the effect of turbine output between natural gas and BFG. Fig.12 Shows Turbine heat drop for both fuel gases.



$$\Delta h = T1 \times \frac{K}{K-1} \times R \times \left\{1-(P2/P1)^{\frac{K-1}{K}}\right\} \times \eta$$

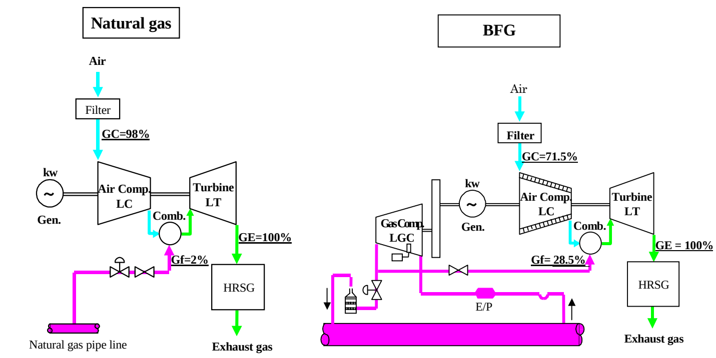
- Δ h ; Heat Drop
- T1 ; Turbine Inlet Temp.
- P_1 ; Turbine Inlet Press.
- P_2 ; Turbine Outlet Press.
- K ; Specific Heat Ratio
- R ; Gas Constant
- η ; Efficiency

Iotm \ Gas		Fuel		Reference		
		N. Gas	BFG	Air	CO_2	H_2O
CO_2 Content	%	4.1	16.7	-	-	-
K	(-)	1.31	1.28	1.40	1.30	1.33
R		29.91	27.71	29.27	19.26	47.06
Δ h	%	100 (base)	95	92	65	155

[Remarks] 1. η =Constant
2. P_2/P_1=1/15

Fig.12 Effect of turbine output

When firing BFG, CO_2 content level in turbine working fluid as combustion gas is increased. As result of CO_2 increasing, turbine output of BFG firing is reduced to be 95% from natural gas firing.



		Turbine Output LT	Fuel Flow Gf			Air Flow Gc		Comp. Power				Gen. Output Kw= Lt-(Lc+Lgc)		Ther. Eff.
								Lc	Lgc	Lc+Lgc	Ratio			
Fuel	LHV $Kcal/m^3N$	%	Flow %	Ratio	Heat Input Ratio	Flow %	Ratio	%	%	%	-	%	Ratio	Ratio
N. Gas	8,800	100	2.0	1.0	1.0	98.0	1.0	52.3	-	52.3	1.0	46.9	1.0	1.0
BFG	1,000	95.0	28.5	14.3	0.95	71.5	0.73	38.8	17.7	56.5	1.08	37.8	0.81	0.85

[Remarks]
1.Flow balance is Turbine exhaust gas base.
2.Power/Output base is Turbine output of natural gas firing.

Fig.13 Effect on total performance

Fig.13 shows the total performance effect considering compressor power and turbine output. When evaluating total compressor power coupled with air and BFG, it's power is 8% increasing than natural gas firing due to the less compressor efficiency. Therefore, 19% on generator output is lower. On the other hand, fuel gas heat input is reduced to be 95% since less compressor efficiency. Combined with the above output and heat input variations, thermal efficiency on BFG firing is reduced to be 85% than natural gas firing.

5. PLANT OUTLINE AND FEATURES OF 300 MW BFG FIRING COMBINED CYCLE PLANT

5.1 Plant outline

The overall equipment layout shown in Fig. 14 and the plant system flow shown in Fig. 15. Specification of main components are shown in Table 8.

5.2 Plant features

The features of the plant are described below.

(1) A combined cycle power plant system is selected in order to obtain a high thermal efficiency. The design value for plant thermal efficiency is taken as 50% (LHV base).

(2) A multi-cannular combustor with variable geometry bypass valve is developed and installed in order to allow low caloric gas operation over the entire operating range (from turbine start-up to full load operation).

(3) The gas turbine, generator, steam turbine, and gas compressor are coupled on a single shaft. (The gas compressor is connected to the shaft through step up gear.) (see Fig. 16) The total shaft length is the approx. 60 meters long and in order to prevent mutual interference resulting from longitudinal thermal expansion of the shaft, both ends of the steam turbine shaft are fitted with flexible couplings.

(4) The plant is started up by the main steam turbine using steam from an existing boiler. This eliminates the need for start-up device.

(5) A multi-cannular low NOx combustor with variable geometry bypass valve is provided so that low NOx operation could be carried out without having to inject steam or water into the combustor.

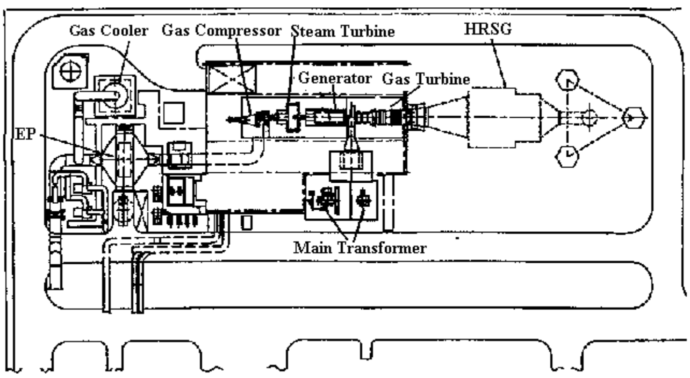


Fig. 14 Overall equipment layout

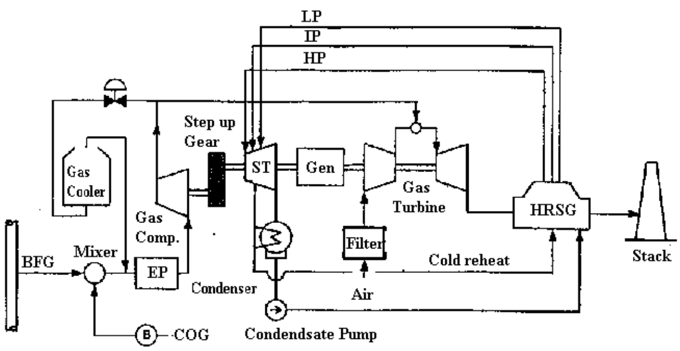


Fig. 15 Plant system flow
Fuel, steam, and exhaust gas flow systems.

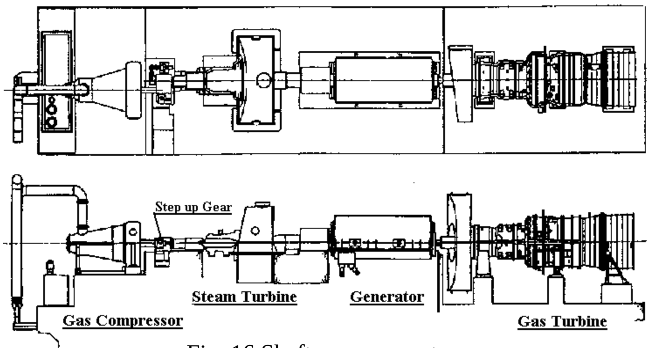


Fig. 16 Shaft arrangement

The gas turbine, generator, steam turbine, and gas compressor are could on a single shaft.

Table 8 Specifications of main components

	Component	Specifications
Gas turbine	Type	Simple open cycle, single shaft M701F
	Output	268,000 kW
	Speed	3000 rpm
Steam turbine	Type	Single cylinder, reheat triple pressure, condensing both end drive type
	Output	119,300 kW
	Inlet press. (HP)	10.30 MPa
	(IP)	2.94 MPa
	(LP)	0.49 MPa
	Inlet temp. (HP)	535°C
	(IP)	535°C
	(LP)	250°C
	Speed	3000 rpm
Generator	Type	AC synchronous generator
	Capacity	340,000 kVA
	Voltage	16 kV
	Speed	3000 rpm
Gas compressor	Type	Axial flow type with variable front vanes
	Power	87,300 kW
	Speed	5,030 rpm

(6) A gas decompression device with direct water cooling system, developed by Mitsubishi, is installed to enable to send back the high temperature and high pressure gas discharged from the gas compressor outlet to the gas supply line in an emergency or during normal shutdown.

(7) A full automatic control system makes it possible for one or two operators to control and monitor the plant from a central control room. This eliminates the need on-site control.

5.3 Equipment features

5.3.1 Gas turbine proper. For this plant, we selected our Model M701F, a simple open cycle, single shaft gas turbine. The reliability of this model has been well established from the successful operating record with natural gas and distillate oil.

In order to convert this standard model from natural gas firing to low caloric by-product gas firing, the design modifications shown on Table 5 are made.

Fig. 17 Shows section drawing of M701F gas turbine.

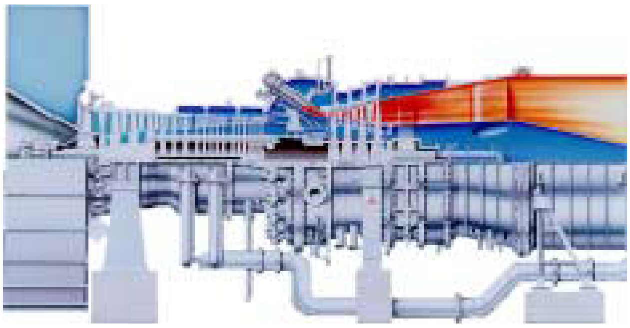


Fig. 17 M701F Section Drawing

5.3.2 Gas compressor. We selected a high efficiency axial flow type gas compressor which was scale designed from a gas compressor model with an extensive operating record.

In order to minimize a drop in efficiency under partial load, we control the fuel gas flow with variable pitched stator vanes when operating at loads of 65% or more. A dry-type segment seal is used for the shaft seals. Furthermore, in order to prevent gas leaks, nitrogen gas is injected between the segment seals at a pressure slightly higher than the ground pressure.

5.3.3 Steam turbine. Requirements for the steam turbine in this plant include :

(1) Safety accelerates all units trained in single shaft from the turning speed (about 3 rpm) up to the ignition speed (about 600 rpm);

(2) Operates at a high efficiency under rated conditions ;

(3) Has the high reliability required of main machine on the shaft line.

In order that the gas turbine may get up to the rated speed within only about 20 minutes after ignition, the cylinder and rotor of the steam turbine must be sufficiently warmed up before gas turbine ignition. Furthermore, the steam turbine must also be able to keep up with the rapid rate of the gas turbine when increasing the load.

In order to maximize reliability, it is best to construct a system that, as much as possible, is made up of parts and components which have an extensive service record. On the basis of this idea, we decided the following design principles:

- The steam turbine shall be of the reheat large capacity single cylinder single flow triple pressure condensing both end drive type;
- The steam turbine shall be able to smoothly start up the shaft on which all units are trained;
- The unit shall be able to handle the different operating modes experienced in industrial use such as DSS, WSS, part load operation, etc.
- The unit shall be highly efficient, highly reliable, and easy to maintain.

6. CONCLUSION

Presented is a description of our BFG firing gas turbine combined cycle power plant, the planning and the design procedures for the components and future prospects for the plants.

The Construction of our latest 300 MW BFG firing F Class combined cycle plant was begun is 2001. In 2004, this plant will be put into the commercial operation.

We will continue the research and development in this field and promote further improvement in thermal efficiency.

References

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