CONCEPTUAL DESIGN OF HELIUM GAS TURBINE FOR MHTGR-GT

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Abstract

Conceptual designs of the direct-cycle helium gas turbine for a practical unit (450 MWt) and an experimental unit (1200kWt) of MHTGR were conducted and the results as shown below were obtained. The power conversion vessel for this practical unit can further be down-sized to an outside diameter of 7.4m and a height of 22m as compared with the conventional design examples. Comparison of the conceptual designs of helium gas turbines using single-shaft type employing the axial-flow compressor and twin-shaft type employing the centrifugal compressor shows that the former provides advantages in terms of structure and control designs whereas the latter offers a higher efficiency. In order to determine which of them should be selected, a further study to investigate various aspects of safety features and startup characteristics will be needed. Either of the two types can provide a cycle efficiency of 46 to 48%. The third mode natural frequencies of the twin-shaft type's low-pressure rotational shaft and the single shaft type are below the designed rotational speed, but their vibrational controls are made available using the magnetic bearing system. Elevation of the natural frequency for the twin-shaft type would be possible by altering the arrangements of its shafting configuration. As compared with the earlier conceptual designs, the overall system configuration can be made simpler and more compact; five stages of turbines for the single-shaft type and seven stages of turbines for the twin-shaft type employing one shaft for the low-pressure compressor and the power turbine and; 26 stages of compressors for the axial-flow type with the single shaft system and five stages of compressors for the centrifugal type with the twin-shaft system. An overall system configuration of the flange joint method to preclude leakages from gaps between the elements was developed, using a plate-finned recuperator and intercooler, and a helically-coiled precooler with low fins, and its feasibility is shown. A development program to lead to the commercial MHTGR-GT plant consisting of three phases including the fundamental design of commercial unit, demonstration of the components technologies and design of the demonstration unit, and fabrication and construction of the demonstration unit was also planned.

1. Introduction

As one of the energy sources to provide solutions to the prevailing environmental pollutions (global warming, industry-related pollution) and to dilemma in the energy supply, the high-temperature heat utilization system using the Modular High-temperature Gas-cooled Reactor (MHTGR) and the Gas Turbine Generator System are now under research and development. The gas-cooled reactor used for this power generation system has originally evolved from the research and development works in the U.S.A.[1 et al.], showing its salient features of; ① inherent and passive safety characteristics (meltdown-proof), ② availability of high-temperature heat, ③ a high thermal efficiency and ④ extremely low radioactive releases, all of which combine to show a new promising reactor for the next generation nuclear power plant. Particularly, fuel particles of approximately 1mm encapsulated in ceramics, etc., are used as the reactor fuel to provide a containment function of radioactive releases within the fuel particles perse and the gas-cooled reactor is provided with its inherent and passive safety characteristics to maintain the safety of the reactor because the heat generated from the fuel can all be released in terms of the natural heat release alone from the surface of the metallic pressure vessel.

Mitsubishi Heavy Industries, Ltd. has been promoting the research and development of the MHTGR under cooperation with Japan Atomic Energy Research Institute and the other associated organizations over many years. Also, the company is the contractor of the HTTR (High Temperature Test Reactor) under development as a national project in Japan and has undertaken its design and construction works. Furthermore, we have been engaged in research and development works of both types of MHTGRs using the steam cycle and the gas turbine cycle and the energy utilization system involved in heat utilization and electric power generation. Among these research and development activities, this paper presents the conceptual designs of an experimental model unit (heat input 1200kWt) and a utility unit (450 MWt) for the gas turbine generator system including the outline of our study results of the major components used, the fundamental characteristics of these main components as well as the developmental issues. It should be noted that the fundamental concept and configuration of the MHTGR-GT have previously been developed principally by General Atomics.

2. Working Fluids

2.1 Selection of the working fluids

Several candidate working fluids have been investigated as the working fluid for the MHTGR and their physical properties are shown in Table 1 and Fig.1[2]. In the present study, inert helium was selected from these candidate fluids because of its excellent heat transfer property.

Table 1 Physical Properties of Working Fluids

Gas	Molecular Equation	Atomic Number	Molecular Weight	Gas Constant (kJ/kg K)	Thermal Conductivity (W/mK)	Density (kg/m³)	Specific Heat at 0C (kJ/kgK)		S.H. Ratio
							Cp	Cv	Cp/Cv
Helium	He	1	4.0026	2.0772	0.1462	0.17850	5.19	3.116	1.66
Argon	Ar	1	39.498	0.20813	0.0163	1.783771	0.522	0.312	1.66
Hydrogen	H_2	2	2.0159	4.1244	0.1683	0.089885	14.188	10.06	1.409
Oxygen	O_2	2	31.9988	0.25983	0.0266	1.42900	0.917	0.655	1.399
Nitrogen	N_2	2	28.0134	0.29680	0.0242	1.25046	1.041	0.743	1.400
Air	—	—	28.964	0.28706	0.0242	1.29304	1.006	0.718	1.402
Steam	H_2O	3	18.0153	0.46151	—	—	—	—	—
Carbon Dioxide	CO_2	3	44.01	0.18892	0.0146	1.97700	0.826	0.631	1.301

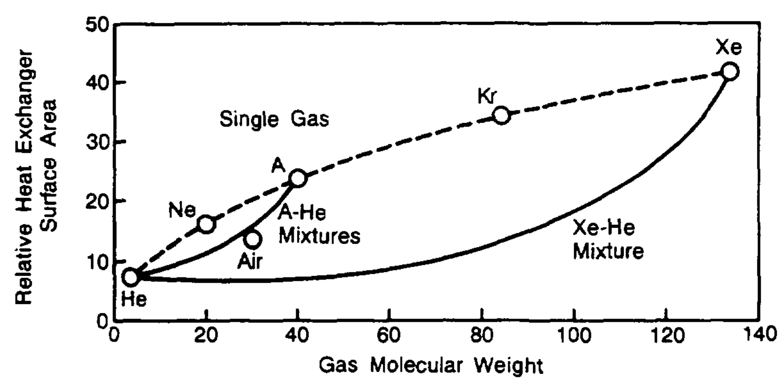


Fig. 1 Relative Heat Exchanger Surface Area vs. Various Gases
(By courtesy of Airesearch)

2.2 Features of helium

As compared with the other working fluids listed in Table 1, helium has physical properties; ① a large thermal conductivity, ② a large specific heat, ③ a large gas constant, ④ a small molecular weight and ⑤ a small gas density. In accordance with such physical properties, helium has the features of its working fluid; ① compact design of heat exchangers used, ② heat drop (output) can be large at a small temperature difference, ③ a large velocity is available at a small pressure ratio, ④ it leaks even through a micro pore and ⑤ it has a large volume per unit mass.

In the open cycle gas turbine, its size must be increased when the working fluid has a small gas density whereas in the closed cycle gas turbine, the turbine size can be reduced by elevating the base pressure, utilizing the excellent features of helium. Comparison of helium and air relative to their features of working fluids for the gas turbine is shown in Figs.2-1 to 2-2 (refer to Table 1 together). In the case of a pressure ratio 2.8, the theoretical velocity of the helium gas is approximately 2.62 times that of air and its dimensionless flow rate is 0.165 times as great. Since the output is proportional to the square of the theoretical velocity and to the product of the flow rate, the output from the helium turbine is 1.13 times the case using air under the same working conditions. Because helium has a small molecular weight leading to its leakage through a micro-pore, its leakage is, in general, assumed to be greater than those of the other fluids, but as shown in Fig.2-2, its leak mass flow is approximately one-sevenths of the air leakage.

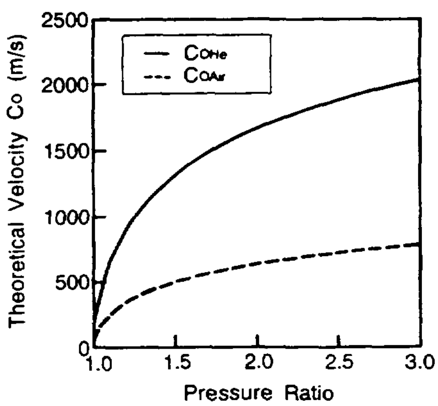


Fig. 2-1 Theoretical Velocity of Helium Gas and Air

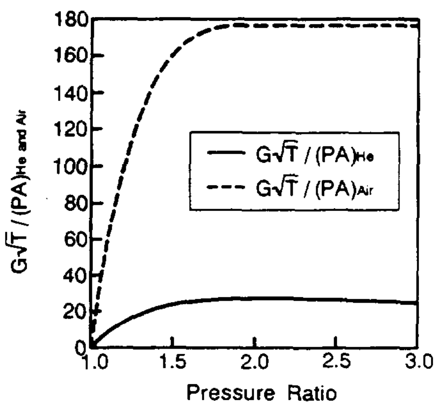


Fig. 2-2 Non-dimensional Flow Rate of Helium and Air

2.3 Features of the turbomachine using helium as its working fluid

In the open cycle gas turbine, the turbine inlet temperature has been set to an elevated point of 1350℃ and the pressure ratio to approximately 30 to improve the thermal efficiency. When this fact is considered, the output from the open cycle gas turbine can be approximately 3.53 times the output from the closed cycle helium gas turbine under the same base pressure condition. Meanwhile, the output from the closed cycle helium turbine can be raised to a level comparable to the output from the open cycle gas turbine by setting the base pressure at more than about 3.53 times the atmospheric pressure.

The base pressure (compressor inlet pressure) of the gas turbine for the MHTGR has been set at approximately 25 ata (2.55 MPa) and its output is approximately seven times the output from the open cycle gas turbine. Although the specific heat of helium is approximately five times that of air, its pressure ratio is well below the level of this difference and the volumetric change of the working fluid is significantly small leading to small changes of its flow paths. Due to this fact, the change of the blading height can be small from the first to the last stages of compressors and turbines.

3. Outlet Temperatures of the Reactor

The MHTGR has a negative reaction property that its reactivity is reduced as the temperature increases. If the internal gas should be released out due to some accidental failures, resulting in a pressure drop and an elevated temperature of the fuel particles, the reactivity would drop, causing the thermal output to be reduced to several percent of the design level. The MHTGR has been designed to provide the so-called inherent and passive safety characteristics; the heat capacity generated during this failure can all be released out of the outer wall of the pressure vessel. The outlet temperature (turbine inlet temperature) of the MHTGR has been set at 850℃ such that the maximum temperature of the fuel during an accident can always be maintained below 1600℃ which can allow the MHTGR. In this paper, study has been performed on the basis of this temperature 850℃. However, the most recent researches show a potential for an elevation of the maximum temperature due to improved fuel particles, re-adjustments of the bypass flows in the reactor and an improvement of the cooling method for the reactor vessel, etc.

4. Cycle Calculations and Component Efficiencies

4.1 Cycle efficiency calculations

The calculated cycle efficiencies are shown in Fig.3. When a recuperater is used, the cycle efficiency increases at the lower side of the pressure ratio as the recuperater exchanging heat capacity increases and as the system pressure loss is reduced. As seen from the theoretical value (component efficiency 100%) of the recuperated cycle, this is because a raised pressure ratio would reduce the exhaust gas temperature resulting in the smaller heat recovery by the recuperater. The efficiency of the recuperated/intercooled cycle is above the theoretical efficiency of the simple cycle in a pressure ratio range below 3.5. Because the efficiency of the

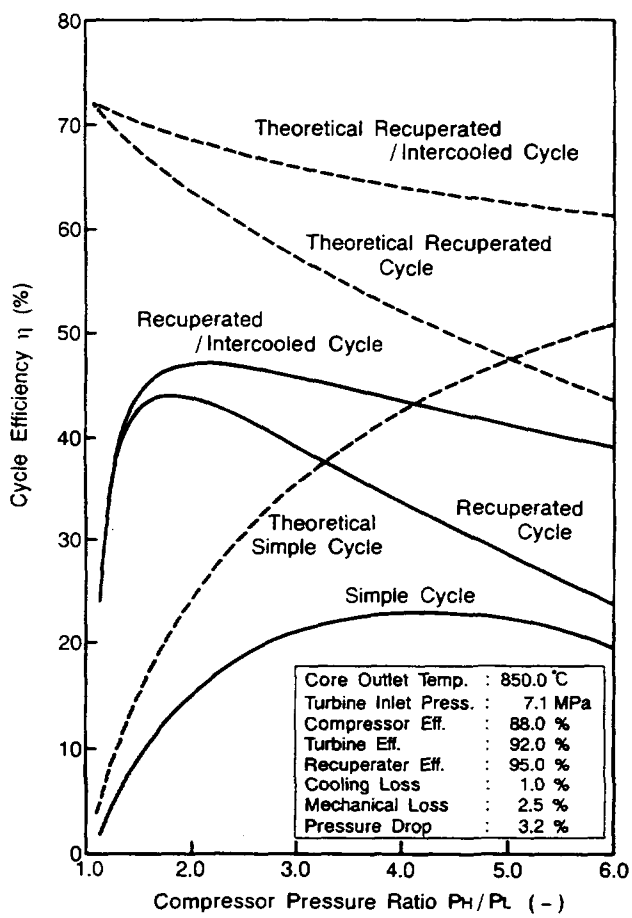


Fig. 3 Cycle Efficiency of Direct Cycle

feasible heat exchanger could be estimated at 95% and the system pressure loss at around 3%, a pressure ratio of 2.8 was selected from Fig.3. Assuming that efficiencies of the components are identical, the efficiency of the simple cycle is approximately 20% and that of the recuperated/intercooled cycle is 46%, respectively at the pressure ratio of 2.8.

4.2 Effects of the component efficiencies and pressure loss on the cycle efficiency

The effects of efficiencies of the main components (turbine, compressor, recuperater, intercooler, etc.) of the gas turbine, the system pressure loss and compressor and turbine inlet temperature on the overall cycle efficiency are shown in Fig.4. If the turbine efficiency, the compressor efficiency or the recuperater efficiency changes by 1%, the cycle efficiency also changes by approximately 0.5%. If the turbine inlet temperature or the pressure loss changes by 1%, the cycle efficiency changes by 0.3% or by 1%, respectively. Because the pressure loss shows the most serious influence on the cycle efficiency, pressure losses through pipings, etc., are needed to be minimized.

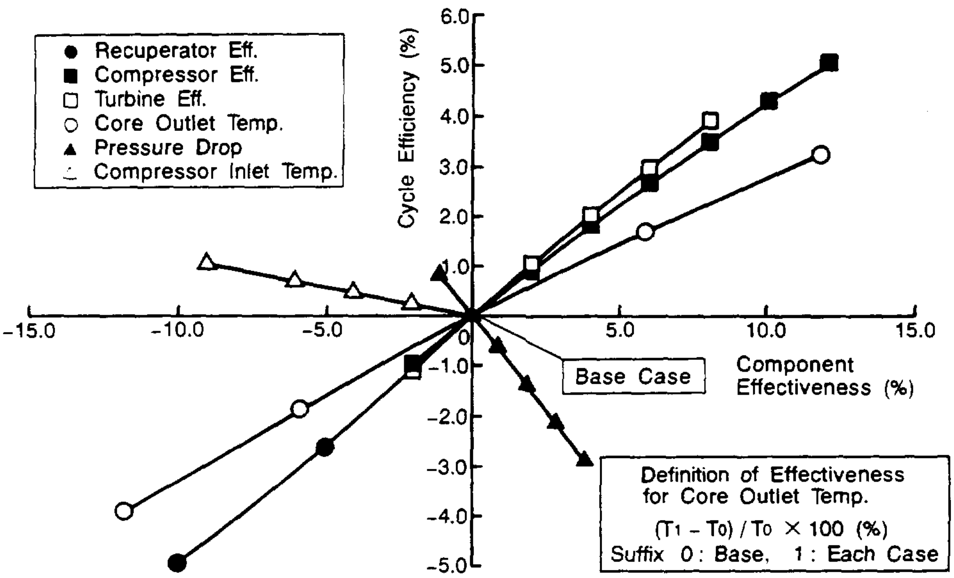


Fig. 4 Effect of Components Effectiveness on Cycle Efficiency (Direct Recuperated / Intercooled Cycle)

5. Type Selection

5.1 Direct cycle and indirect cycle

The direct cycle uses the working fluid in the reactor to directly drive the turbine while the indirect cycle heats another working fluid through heat exchanger to utilize this heated fluid to drive the turbine. The direct cycle system can have a high cycle efficiency realized because its system can be simplified with a high turbine inlet temperature being made available. The indirect cycle system is supposed to show a higher safety reliability because it can minimize a potential for pollution of the turbomachine by radioactive substances due to use of another working fluid through the heat exchangers, and the freedom of its design work can be higher because the working fluid for the turbomachine can be freely chosen. Also, in utilization of the high-temperature, the direct cycle must take the type identical to the indirect cycle type. Both have advantages of their own and the view of the system advantages can vary, depending on what factors one should place emphasis on.

Currently it is shown that the fuel particle has a high reliability and that the plant safety can be assured even with the direct cycle system, and since the potential feasibility of the indirect cycle system could be high if the direct cycle system is realized, the conceptual design of the direct cycle system has been conducted in our present research.

5.2 Single-shaft type and twin-shaft type

The twin-shaft system is composed of two independent shafts; the power turbine directly connected to the generator and the turbines to drive the compressors. Because this selection allows a free selection of the rotational speed of the drive turbine for the compressor, elevated efficiencies of the compressor and its drive turbine can be obtained. However, it requires the installation of a motor to drive the compressor during startup and there is a problem of the difficult control of its overspeed for various assumed accidents. The MHTGR has a helium tank and the compressors can also be started by utilizing the high-pressure helium gas stored in this tank[3]. In the single-shaft system, the rotational speeds of the turbine and the compressor are needed to set to the same speed of the generator, which enlarges outside diameters of the turbine and the compressor, reducing the blade height which leads to difficulties in the design of efficiency improvements. Also, assurance of the rotational shaft stability can be a major problem because the shaft length is extended with the natural frequency of the shafting system being low.

The aforementioned problems have been examined by performing outline designs of two cases of heat outputs 450 MWt and 1200 kWt. In the case of the heat output 450 MWt, the efficiency difference in the compressor between the twin-shaft system and the single-shaft system is 1 to 1.5% and the efficiency difference in the turbine is around 0.5% and the efficiency improvement and the compact design are possible, but the benefit of the twin-shaft system is small in terms of the shaft system vibration.

In the case of the heat output 1200 kWt, the single-shaft system shows a poor efficiency because of a reduced blade height, and because the shaft becomes very small and long, it is determined that maintaining the shaft system stability can be difficult.

6. Design Requirements and Specification

The design requirements have been decided based on our study results of the cycle efficiency calculations, etc., as described earlier and on the study results in U.S.A. [4 et al.]. The results of these design requirements are also applicable to the cases slightly deviating from the given conditions. The specifications of the respective components derived from our conceptual design are compared with another data[4, 5, 6, 7] in Table 2. Both data are nearly identical except for the details of estimate and distribution of losses.

7. Design of the Turbomachine

The fundamental technologies required for the design of the turbomachine have already been proven technologies of aerospace and industrial gas turbines and the turbomachine can be designed using those technologies. However, the gas turbine for the MHTGR requires a design simultaneously provided with the lightweight design for aerospace application and the long-term endurance capability for utility service, and there has been no such design experience, which can be deemed a major developmental issue. The turbomachine should most adequately be designed and its periodical inspection interval should be established by examining and analyzing official permit and authorization laws in various countries, regulations regarding the periodical inspection, etc., proven lives of the components for industrial and aerospace gas turbines, and the factors to limit their lives, etc.

7.1 Turbomachine Type

With regard to the turbomachine type, different types of the compressor and their advantages or disadvantages are shown in addition to the differences between the single-shaft and twin-shaft types as described earlier. Although earlier conceptual designs have dealt with the axial flow turbine and the axial flow compressor, the axial flow compressor has an extended shaft length and a large number of its stages, which increases its costs. Accordingly, our conceptual design has also been done on the case employing the centrifugal compressor where the number of required stages and the manufacturing costs can be curtailed and piping to the intercooler is more easily made available. Although the common practice of the twin-shaft type

Table 2 Specification of MHTGR-GT

System			MHI-GT	GT-MHR	MHTGR-GT
Core	Thermal Power	MWt	450.0	600.0	450.0
	Coolant Pressure	MPa	7.07	7.01	7.03
	Flow Rate	kg/s	244.0	328.0	243.0
	Outlet Temp.	℃	850.0	850.0	850.0
	Inlet Temp.	℃	494.0	497.0	494.0
	Gas Turbine Power	MWe	208.0	301.0	228.0
	Net Cycle Efficiency	%	46.3	47.2	47.7
Effectiveness	Compressor	%	88.0	90.0	90.0
	Turbine	%	92.0	92.0	92.0
	Recuperator	%	95.0	95.0	95.0
	Mechanical Loss*1	%	2.5	2.5	2.5
	Cooling Loss*2	%	1.0	0.3	0.3
	Generator Eff.	%	98.0	97.5	97.5
	Pressure Loss*3	%	3.2	4.4	3.5
Turbine	Inlet Temp.	℃	850.0	850.0	850.0
	Outlet Temp.	℃	515.8	517.8	514.1
	Pressure Ratio	—	2.66	2.64	2.67
Compressor	Outlet Temp.	℃	35.0	33.0	33.0
	Inlet Temp.	℃	86.5	110.8	110.8
	Pressure Ratio	—	2.80	2.80	2.80
Recuperator	Heat Load	MWt	516.0	658.0	484.0
	Hot Inlet Temp.	℃	515.8	517.8	514.1
	Hot Outlet Temp.	℃	108.0	131.1	130.9
	Cold Inlet Temp.	℃	86.5	110.8	110.8
	Cold Outlet Temp.	℃	494.3	497.5	493.9
Precooler	Heat Load	MWt	92.0	167.0	124.0
	Inlet Temp.	℃	108.0	131.1	130.9
	Outlet Temp.	℃	35.0	33.0	33.0
Intercooler	No.	—	2	1	1
	Heat Load	MWt	65.2	65.2	41.4
	Inlet Temp.	℃	86.5	110.8	110.8
	Outlet Temp.	℃	35.0	33.0	33.0

Note *1 Ratio of House Load to Gas Turbine Power for GT-MHR
*2 Ratio of Generator Windage Loss to Gas Turbine Power for GT-MHR
*3 Total Pressure Loss to Turbine Inlet Pressure

is to use a common shaft for the compressor and its drive turbine and use the power turbine to drive the generator, the use of a common shaft for the low pressure stage of the compressor and the power turbine has been proposed to make available the compressor startup by the generator and to preclude its overspeed, and the latter method has been studied in this paper.

7.2 Compressors

The results of our conceptual designs of helium gas turbines employed axial flow and centrifugal compressors are compared with the conventional designs in Table 3 and in Fig.5. In our design of the compressors, the axial flow type has been employed for the single-shaft type and the axial flow and centrifugal types for the twin-shaft type. In all the cases, the peripheral speed has been raised and the number of compressor stages reduced as compared with the conventional designs. The case of the single-shaft type has been divided to the H.P. group and the L.P. group with an intercooler provided between the two groups, consisting of 13 stages of axial flow compressors, respectively, for the H.P. group and the L.P. group. In the case of the twin-shaft type, the number of stages has been reduced to five stages, one-fifth of that for the axial flow type, by employing the centrifugal type. The outside diameter of the impeller for the centrifugal compressor is as large as approximately 1.6m for the L.P. stage and the maximum peripheral speed is as high as approximately 500 m/s for the H.P. stage, and therefore, stainless steel or titanium alloy is used for the impeller material. Also, the twin-shaft type uses the shaft configuration of the shaft for the L.P. stage penetrating through the shaft for the H.P. stage such that the turbomachine can be started by the generator which drives the L.P. compressor at the startup to raise the pressure in the system.

Table 3 Turbomachine Salient Features

System	Nuclear Gas Turbine			Power Generation Unit	
				Heavy Duty Industrial GT	Aeroderivative Gas Turbine
Plant	MHI-MHTGR-GT		GR-MHR	MS9001F (GE)	LM6000 (GE)
Power (MWe)	208		286	226	42
Working Fluid	Helium		Helium	Air	Air
Thermodynamic Cycle	Recuperated and Intercooled		Recuperated and Intercooled	Simple Cycle	Simple Cycle
Turbine Inlet Temp. (℃)	850		850	1,288	1,243
Compressor Pressure Ratio	2.8		2.8	15	30
Mass Flow Rate (kg/sec)	244		320	613	123
Machine Orientation	Vertical		Vertical	Horizontal	Horizontal
Shafting Type	Twin	Single	Single	Single	Twin
Overall Length (m)	7.9	8.9	13	14.5	4.5
Overall Diameter (m)	2.8	2.8	2.8	4.8	2.5
Overall GT Weight (kg)	65,000	70,000	82,000	300,000	5,600
Rotational Speed (rpm)	3,600/6,000	3,600	3,600	3,000	10,225/3,000
Compressor Number of Stages Max. Tip Diameter (mm)	Centrifugal 3HP+2LP 1,762	Axial 13LP+13HP 1,580	 14LP + 19Hp 1,683	 18 2,515	 5LP + 14HP 1,372/737
Turbine Number of Stages Max. Tip Diameter (mm) Tip Speed (m/sec) Blade Cooling	 2HP+5LP 1,905 506/359 Uncooled	 5 2,500 471 Uncooled	 11 1,778 335 Uncooled	 3 3,251 510 Cooled	 2HP + 5LP 889/1,321 476/249 Cooled
Bearing Type	Active Magnetic		Active Magnetic	Oil Lubricated	Oil Lubricated
Number of Bearings (Thrust/Journal)	2/4	1/4	1/4	1/2	1/6

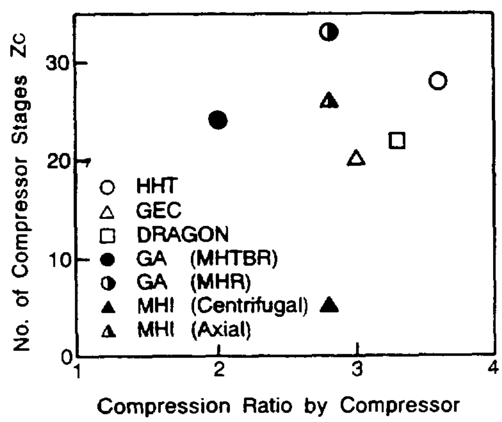


Fig. 5 Comparison of Number of Compressor Stages for Direct Cycle

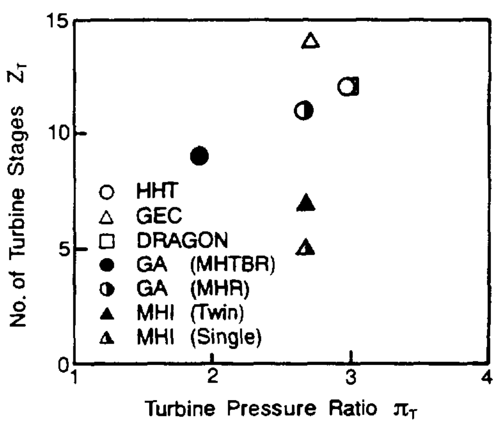


Fig. 6 Comparison of Number of Turbine Stages for Direct Cycle

7.3 Turbines

The results of our conceptual design of the turbine are compared with the conventional designs in Table 3 in Fig.6. The number of turbine stages is five stages for the single-shaft type and seven stages for the twin-shaft type. For the single-shaft type, the increase of the shaft length has been avoided, aimed at a high-load design. The twin-shaft type which shows allowance for the load and the number of stages is found more advantageous over the single-shaft type in terms of performance. The outside diameter of the turbine is 2.5m for the single-shaft type and 1.9m for the twin-shaft type. In either of the types, the gas temperature at the moving blade inlet is 850℃ or less which is below the heat-resistant temperature of the blade material, eliminating the necessity of cooling. For the moving blade, a wide-chord blade to provide a reduced number of blades and improve its vibration-resistant strength and a lightweight hollow blade to alleviate stresses to the disc and the connection between the disc and the blade has been employed. Also, because the blade tip clearance is increased when a back-up bearing is provided accompanied with the employment of the magnetic bearing, the moving blade is provided with shround and tip-fins to minimize leakage from the tip clearance. The disc has a large diameter and the heat-resistant temperature of the material suited to such a large disc is approximately 600℃, which requires the disc to be cooled. The performance drop due to this cooling is estimated to be 0.5 to 1%.

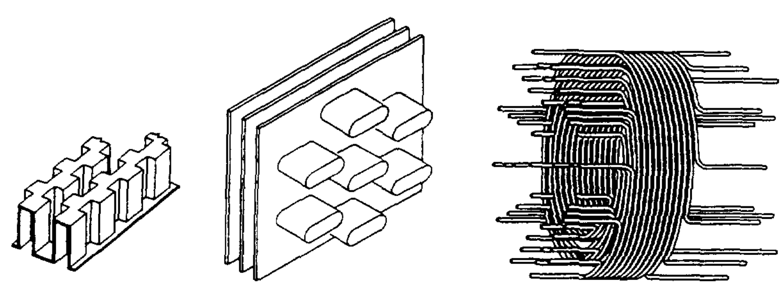
8. Heat Exchangers

Heat exchangers required for the gas turbine are a recuperator, a preheater and an intercooler, and their high efficiency performances and compact designs are demanded. Table 4 lists specifications of these heat exchangers and Fig.7 shows configurations of the heat exchanger elements.

Table 4 Heat Exchanger Salient Feature

Unit	Recuperator			Intercooler / Precooler	
	MHI	GT-MHR	Industrial GT	MHI	GT-MHR
Gas Turbine Power (MWe)	208	286	10 to 60	204	286
Working Fluid (High/Low)	He/He	He/He	Air/Gas	He/Water	He/Water
Unit Thermal Rating (MW)	515	630	15 to 30	65.2/92	131/164
Exchanger Type	Plate-Fin	Plate-Fin	Plate-Fin	Helical Coil	Tubular
Number of Modules	8	6	2 to 4	2/1	—
Flow Rate (kg/sec)	244	320	45 to 240	244	320
Hot Gas Inlet Temp. (K)	516	510	519 to 575	87/108	112/131
Pressure Difference (MPa)	4.5	4.6	0.9	3.6/4.5	4/2.3
Effectiveness	0.95	0.95	0.84 to 0.89	0.83/0.84	0.93/0.95
Overall Pressure Loss (%)	1.6	2.0	3.5 to 4	0.2/0.2	0.47/0.75
Typical Surface Density(m/m)	714	1,906	624	488/295	500
Heat Transfer Coeff. (W/mK)	LP : 1,857 HP : 1,876	LP : 2,555 HP : 3,120	LP : 115 HP : 425	He :1,780/585 Wa: 8600/7230	He : 1,250 Water:11,000
Thermal Density (MW/m)	8.9	17	1	4.0/0.57	3.6
Typical Flux (W/cm)	1.5	2.5	0.1	0.82/0.2	0.75
Material	316 StSt	316 StSt	409 StSt	1/2Cr-1/2Mo	1/2Cr-1/2Mo

*1 : Heat Transfer Area of Hot Gas Side to Total Recuperator Heat Transfer Section Volume
*2 : Typical Specification of First Stage Intercooler
*3 : Low-Fin Tube



(a) Recuperator (Plate-Fin)   (b) Intercooler (Wavy-Fin Flat Tube)   (c) Precooler (Helical Coil)

Fig. 7 Heat Exchanger Elements

8.1 Recuperator

For the recuperator, the plate-finned compact heat exchanger has been selected because a high heat transfer efficiency of 95% is demanded due to a large heat transfer of approximately 520 MW. Depending on the design specification, various fin types are selected for plate-finned heat exchangers and they are broadly used as small heat exchangers. For this system, the offset fin which can provide higher heat transfer efficiency has been employed. The offset fin is a small type of 1.9mm in height, 0.1mm in plate thickness and 1.0mm in fin-to-fin spacing, but this would pose no practical problems because the working fluid is high-purity, inert helium, which will eliminate concerns about oxidation or fouling.

8.2 Intercooler

The intercooler is used to reduce power consumption for the compressors and improve the efficiency of the regenerative gas turbine system. In this section, helium in the pressure rise process of the compressors is cooled by the cooling water running through a wavy-finned flat tube. The system design has been done on the basis of installation of two intercoolers, and Table 5 shows the specification of the first stage of the intercooler.

8.3 Precooler

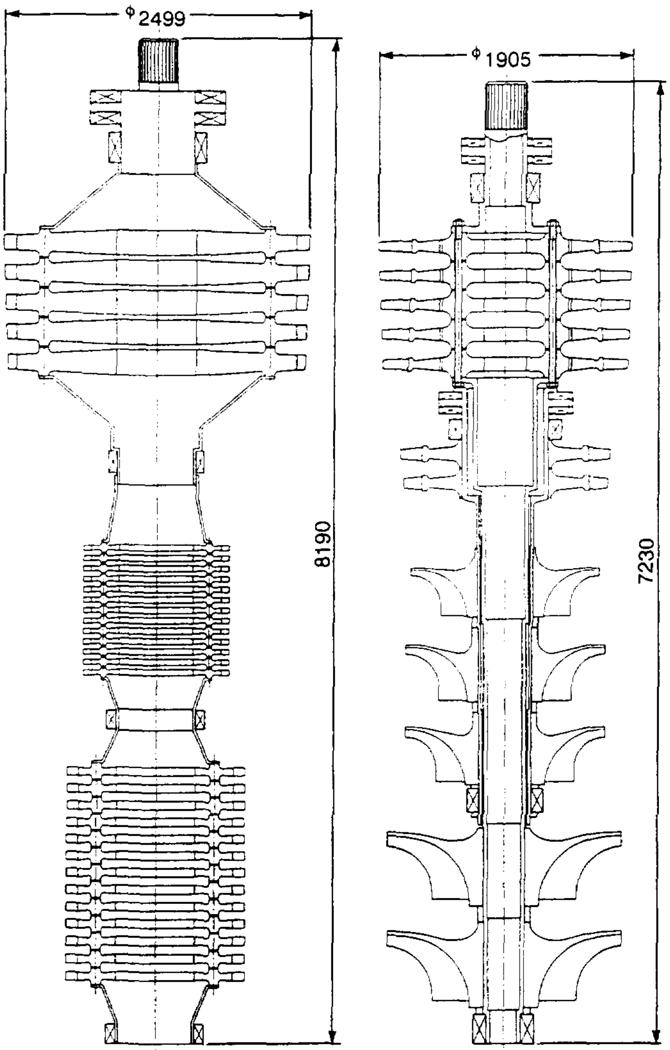
For the precooler, the helical coil type heat exchanger which has proven results for various machine type has been selected and the low-fin tube has been employed to reduce the height. If a plate-finned or fin-and-tube type heat exchanger is employed for the precooler, it can reduce the installation space for the precooler. This subject will be dealt with in the next stage of our research work.

9. Study on Shafting and Bearings

Fig.8 shows the arrangements of shafting and the bearing for the single-shaft type and the applied to the practical unit of 450 MW. There are several examples of the shafting arrangements of the twin-shaft type for the gas turbine, and the fundamental arrangement uses one shaft for the gas generator and the other for the power turbine, which can reduce the overall shafting length and allow the rotational speed of the compressor shaft to be freely selected, providing option for a higher efficiency, but this arrangement requires addition of a motor, etc., to drive the gas generator at startup. As shown in the figure, the twin-shaft type in this design uses the low-pressure compressor shaft inserted through the high-pressure shaft because a common shaft has been designed for the low-pressure compressor and the power turbine for the compressor to be driven by the generator.

9.1 Shafting

Fig.9 shows the calculated results of the rotational shaft vibrations. The first mode is the parallel mode, the second X mode and the third the bending mode. As shown in Fig.9, the designed rotational speeds of the L.P. shaftings of the single-shaft type and the twin-shaft type are in a region of rotational speeds higher than the third mode and it is difficult to exceed the natural frequency of the third mode when oil lubricated bearings are used, but the natural frequency of the third mode can be exceeded if the magnetic bearing system which can control the shaft vibration is employed. However, it is desirable to maintain the designed rotational speed below the natural frequency of the third mode, and hence, it is needed to devise the elevation of the natural frequency by reducing the shaft lengths of the turbine and the compressor in order to raise rigidity of the shaft bending. For the two-shaft type, the L.P. shaft is placed through the hollow H.P. shaft because the startup using the generator has been made available as already described. This causes the length of the L.P. shaft to be extended, resulting in a reduced natural frequency. If a common shaft were used for the gas generator turbine and the compressor with the power turbine separated from this common shaft and if the high-pressure helium stored in the helium tank were used for the startup, the length of the L.P. shaft would be reduced, causing the natural frequency of the third mode to exceed the designed rotational speed. In this paper, the most serious problem of shafting has been examined.



(a) Single-Shaft, Axial Compressor (b) Twin-Shaft, Centrifugal Compressor

Fig. 8 Helium Turbine Rotor

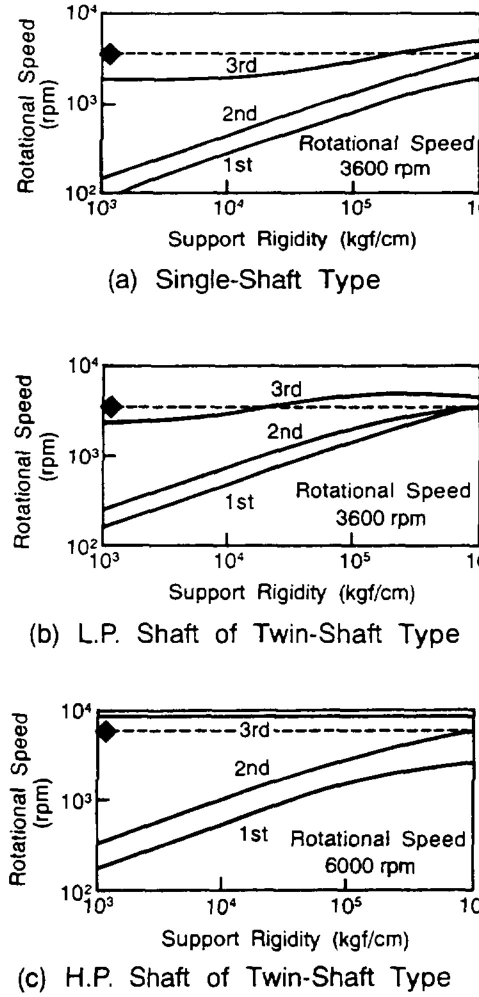


Fig. 9 Calculated Results of Rotational Shaft Vibration

9.2 Magnetic bearing

The magnetic bearing is provided with the following features; ① no lubricant is needed, ② a small loss, ③ the shaft vibration is controllable and ④ the load capacity does not depend on the rotational speed. For the closed cycle gas turbine, ① the gas bearing using the working fluid as a lubricant and ② the non-lubricant magnetic bearing can be pointed out as candidate bearings to avoid ingress of the lubricant to the working fluid. The gas bearing poses problems of the cooling method and small deformations of the bearing and the shaft because the gas bearing has a small load capacity and a large loss. Meanwhile, the thrust bearing of the magnetic bearing system can support a large load because it has a small loss and provides a large support area. In the journal bearing, control of the shaft vibrations from the low to high modes is available. Therefore, it is best suited for the bearing system for the vertical shafting arrangement of an extended shaft length where the thrust bearing supports an overall weight of the rotor.

The selection chart of the outside diameters of the magnetic thrust bearing is shown in Fig.10. Although the dimensions of the thrust and the journal bearings are over the range of proven experience, they can still be manufactured in terms of technical capability. However, the magnetic bearing is supposedly needed to be provided with ball bearings as the backup

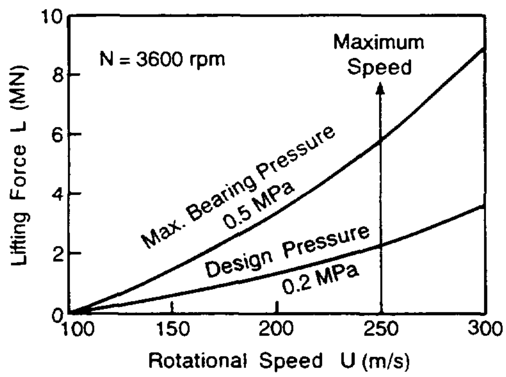


Fig. 10 Lifting Force of Magnetic Thrust Bearing Relative to Peripheral Speed

bearings at the shaft ends in order to avoid contacts at startup and stop and a large clearance must be provided to avoid the contact of this bearing with the shaft during the normal operation. Accordingly, it becomes necessary to provide large clearances for turbine and compressor tips, which reduces their performances. It is desirable to use a backup power supply as an alternative to the backup bearing and design alleviation of damage to the contact area between the shaft and the bearing and their developments are expected for.

10. Design of the Overall System Configuration

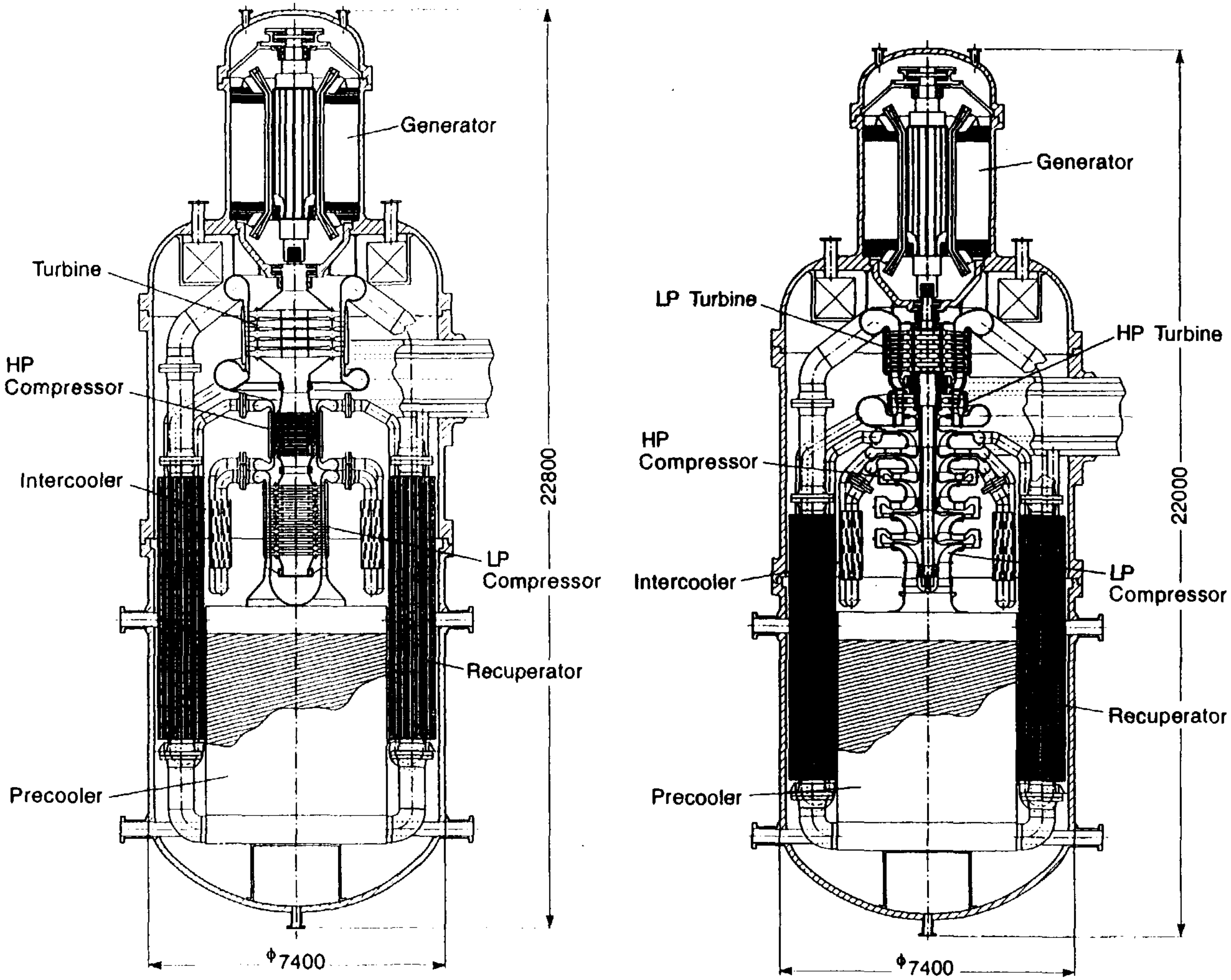
The structural sections of the utility unit and the experimental model unit as designed in this research are shown in Figs.11 and 12. For the utility unit, a configuration of the generator and the gas turbine contained in one module, the recuperator in one module, the intercooler in one module and the preheater in one module, has been employed to carry out the maintenance servicing and the periodical inspection after removing each module from the pressure vessel in order to minimize the required work in the vessel. Flanges will be provided between the respective modules to preclude gas leakage from between the modules. The flange connection method must take into account thermal expansions and deformations, etc., of the assemblies and components, and they will be examined in the stage of the detail design work.

The support structure for the gas turbine will be retained on a support plate between the generator and the turbine and will use the hanger type. Accordingly, the casing structure of the gas turbine is to have a rigidity required to support the shafting and to be of the self-containment structure which could contain fragments of a damaged blade if such a damage should happen.

For the assembly method of the modules, the flange connection or the insert type can be conceived. The former has a problem of the alleviation method of stresses due to deformations caused by thermal elongation, etc., while the latter has a problem of the sealing method. In addition, the arrangements of pipings and the components must be carefully designed, taking into considerations access of maintenance personnel to the vessels, the working space and the workability, etc.

11. Development Program

The main flow of this development program is shown in following table. The preliminary research work in the initial period in this program has already been shown in this paper and the overall system configuration and the outlined configurations of the respective components have conceptually designed, based on the existing technological information and MHI's own design database. In Phase 1 in the next step, the fundamental design of the utility unit will be performed and the component technologies and the design technology required for the development of the utility unit will be fulfilled. In Phase 2, verifications of performances



(a) Single-Shaft Type

(b) Twin-Shaft Type

Fig. 11 Power Conversion Modules

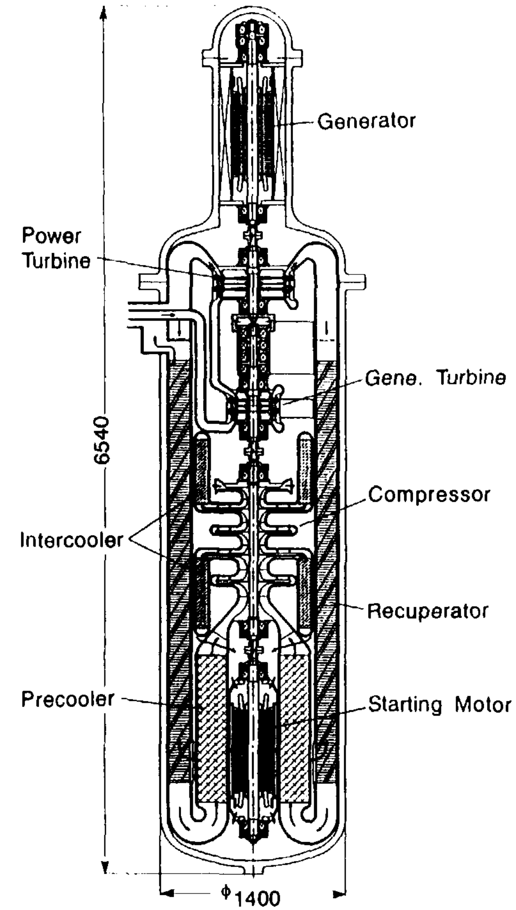


Fig. 12 1200 kWt Test Model Unit

and functions of the required components as well as the design of a demonstration unit will be carried out, ready to respond to the examination of the Safety Counsil, etc. In Phase 3, fabrication and construction of a demonstration plant will take place.

Development Program

Phase 1	Phase 2	Phase 3
· Fundamental Design of Utility Unit · Fulfillment of Component & Design Technologies	· Design of Demonstration Unit · Demonstration of Performances & Functions of Components	· Fabrication & Construction of Demonstration Unit

12. Conclusions

Assuming a full-scale development of MHTGR-GT, the authors have completed the conceptual design of the recuperated cycle intercooled helium gas turbine of the closed cycle on the basis of MHI's own technologies and various other information currently available. As a result, study results nearly identical to the results already obtained in the U.S.A. and other countries have been obtained and a potential for a more compact design has also been confirmed. Furthermore, developmental problems regarding the respective components and the total system have also been clarified and the development program up to the stage of practical use of this gas turbine has been summarized. In future, we intend to realize the Modular High-Temperature Gas-cooled Reactor in cooperation with Japan Atomic Energy Research Institute and other organizations concerned as well as manufacturers in this field both domestic and abroad.

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