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DESIGN OF CLOSED-CYCLE HELIUM TURBINE NUCLEAR POWER PLANTS

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ABSTRACT

This paper describes the design of two closed-cycle helium turbine electric power plants using a modular high temperature gas-cooled reactor (MGR) as heat source. The MGR-GT utilizes a direct cycle with the turbine in the primary coolant loop while the MGR-GTI uses an intermediate heat exchanger. Both plants are based on current technologies and offer busbar efficiencies in excess of 45% at generating costs competitive with other nuclear or fossil systems. The MGR-GTI indirect cycle faces less technical challenge and is easier to license under current technical and institutional realities, whereas the MGR-GT direct cycle is better able to accommodate future developments in fuel and materials and is an ultimate design capable of capturing the full economic potential through further design simplicity. The two designs thus provide an evolutionary path for the commercialization of MGR gas turbine technology.

1. INTRODUCTION

Current generation nuclear reactors have suffered high costs and public mistrust in many countries. The complexity of existing reactors as a consequence of their reliance on a defense-in-depth safety strategy has also limited their use to nations with highly-developed nuclear energy infrastructures. The modular high temperature gas-cooled reactor (MGR) is an attractive option for the next generation of nuclear power. It relies on intrinsic physical laws for inherent safety. The reactor's ability to survive worst case accidents, without relying on safety control systems or operator intervention, permits experiments to prove reactor safety. This allows a "Licensing by Tests" approach to nuclear regulation, providing the strongest possible basis for satisfying the social and political prerequisites for next-generation nuclear power (Lidsky, 1990).

Combining the MGR with a closed-cycle gas turbine creates a high performance, low-cost electric power plant. The high temperature capability (>850°C) of the reactor can be fully utilized by the gas turbine to achieve high cycle efficiency while the reactor's low power rating (200-450 MWth) can be conveniently matched by a small gas turbine without sacrificing economies of scale. The closed-cycle is able to operate at higher pressure than open-cycle systems and to use noble gases such as helium to enhance thermodynamic design. This allows small system size, high power density, and lower plant costs.

Over the half century since the introduction of the closed-cycle gas turbine by Ackeret and Keller, only 15 plants of less than 150 MWe in total output were built worldwide, of which only one was a helium system (McDonald, 1992, and Zenker, 1988). Such a justifiable lack of enthusiasm was largely the result of many past technological difficulties that limited closed-cycle performance and economics. Recent development in enabling technologies, however, call for re-examining the closed-cycle option. MGR closed-cycle helium turbine power plants, for example, can now be built — using proven fuels and materials, based on established industrial gas turbine and heat exchanger capabilities — that yield more than 45% busbar efficiency at a competitive generating cost. The economic potential of the closed-cycle helium turbines may also be significant for alternative energy sources such as natural gas and biomass. This paper focuses on two nuclear plant designs and demonstrates that closed-cycle helium turbine power plants combined with the MGR can be a near-term option for safe, economic electric production.

2. SYSTEM FEATURES AND TECHNOLOGY STATUS

A practical closed-cycle gas turbine consists of only six major components in a single circuit, as shown in Fig.1. Fig.2 shows the strong relationship between cycle efficiency and turbine inlet temperature and recuperator effectiveness. Increasing recuperator effectiveness from 0.85 to 0.95 improves the cycle efficiency by approximately 6%, while a simultaneous increase in turbine inlet temperature from 750 to 950°C increases the cycle efficiency by more than 12%. A combination of an 850°C turbine inlet temperature and a 0.95 recuperator effectiveness, both achievable based on today's fuels and heat exchangers, allows for a 49% cycle efficiency. In fact, MGR reactors have been successfully operated at temperatures up to 950°C with current fuels and a further increase in fuel temperatures is possible with the new fuel technology being developed. This raises the potential for future efficiencies to exceed 50%.

Existing industrial experience with combustion air turbines provides a well-established technical basis for closed-cycle helium

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turbines because of their physical and aerodynamic similarities. Interestingly, helium turbines face less technical challenge in many respects than air turbines. The inlet temperatures (750-850°C) for current helium turbine designs are very conservative with regard to the blades of the first rotor stages, being several hundred degrees lower than combustion air turbines. In fact, helium turbines are better suited to higher temperatures since oxidation is excluded — many high temperature and high strength materials such as molybdenum alloys may be used — and disks and blades are less stressed due to smaller rotor dimensions. Considering aerodynamic design, the sonic velocity of helium is three times that of air, which removes Mach number limitations on blade design so that higher rotor speed can be utilized to achieve better aerodynamic blade design and efficiency. Given the turbine inlet temperatures of interest, conventional blade materials such as IN 738 LC are sufficient for use in rotor construction and no blade cooling is necessary.

The fact that the specific heat and thermal conductivity of helium is five times that of air has compound influence on heat transfer equipment and system ducting. First, because of the larger specific heat, the work done per unit helium mass flow is five times that done by the air flow between the same cycle temperature limits, and the cycle pressure ratio and volume expansion rate are much smaller for helium than for air. These differences lead to a much higher power density, smaller system size and less cycle pressure loss than for open-cycle systems, greatly simplifying the construction of the helium plants. Second, with the much higher thermal conductivity and higher permissible flow velocity, the heat exchanger equipment in a helium turbine system is only about one third of the size of the equipment in a comparable air plant.

Indirect heat rejection is characteristic of any closed-cycle system, both steam and gas turbine alike. However, the problems facing steam turbine cycles do not apply to a large extent in gas turbine systems. In contrast to the heat rejection at a virtually constant temperature in the steam cycle, the gas turbine cycle rejects its heat over a wide band of high temperatures as indicated in Fig.1. The high temperature heat rejection makes dry cooling economically feasible because considerably less air flow and cooling tower size are involved than in a steam cycle. The gas turbine cycle is also advantageous in wet cooling as it would require only a fraction of the water quantity and heat transfer surface of a steam plant. Another advantage of the gas turbine cycle is the ability to use its high temperature waste heat for district heating or desalinization without affecting electric generating performance.

For a closed-cycle gas turbine plant, partial power operation over a broad load range may be achieved through pressure level control at constant efficiency and at constant temperatures in all major components (Bammert, et al., 1974, and Yan and Lidsky, 1991a). Thus the plant is suitable to both base and isolated power schemes without serious economic consequence or excessive thermal cycling. Control operation involves few components and permits a high degree of automation. With reliable gas turbine components operating in a clean, inert gas and at modest operating conditions, the plant maintenance requirements and costs are minimized.

In summary, a high performance, low-cost closed-cycle helium turbine based on an MGR can be built with a conservative implementation of current fuel, material, gas turbine and heat exchanger technologies.

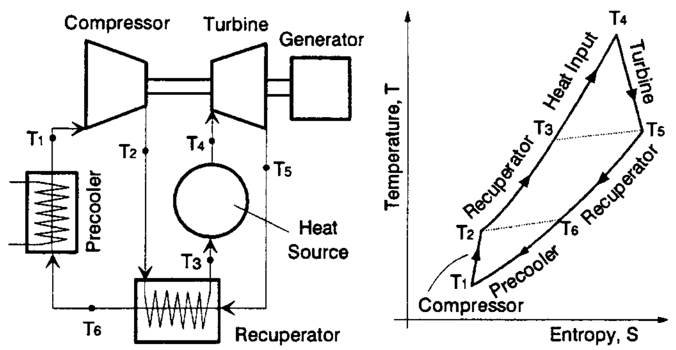


Fig.1: GAS TURBINE CLOSED-CYCLE

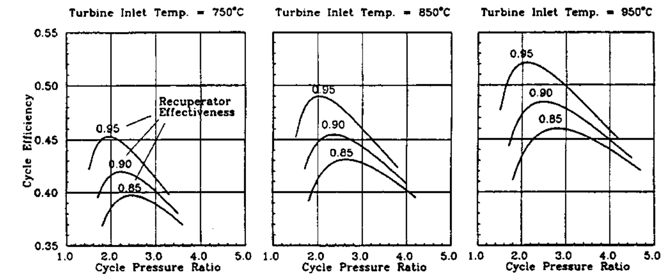


FIG. 2: CLOSED-CYCLE EFFICIENCIES

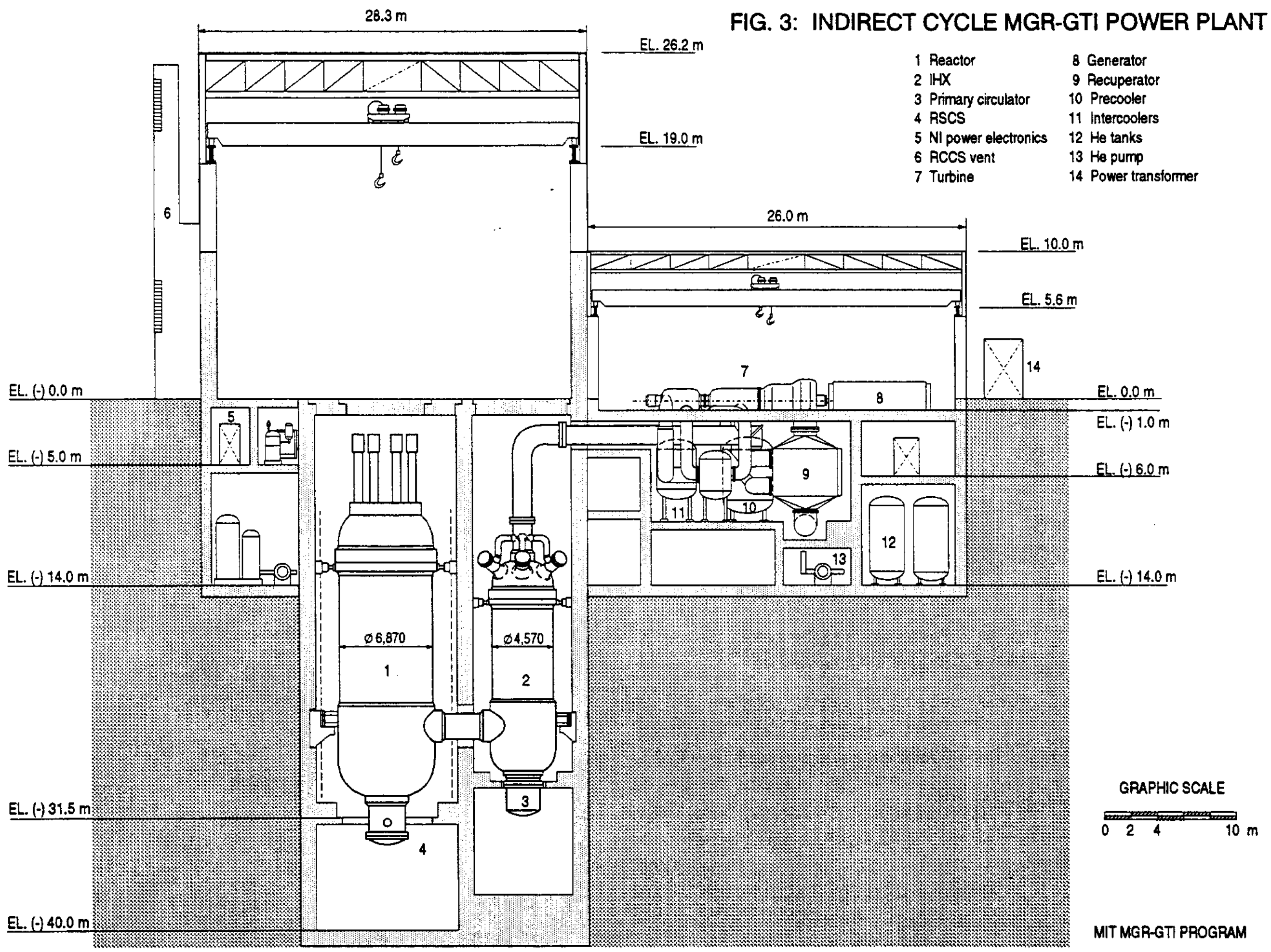
3. INDIRECT CYCLE PLANT DESIGN - MGR-GTI

The indirect cycle power plant uses a 350 MWth MGR in a primary loop to supply heat through an intermediate heat exchanger (IHX) to a closed-cycle gas turbine power conversion system located in a secondary loop. The plant layout shown in Fig.3 is divided into two major areas: a nuclear island consisting of the reactor and IHX silos and a power conversion area housing the gas turbine system. All safety-related components and structures are located in the nuclear island, permitting procurement and construction of the balance of plant to conventional standards. Separating the gas turbine from the reactor primary loop avoids fission product contamination in the turbine equipment and the hazardous impact of a rotor failure on the reactor. The following is a design summary; details of the design have been documented elsewhere (Yan and Lidsky, 1992a).

3.1 Cycle Process

The cycle design is illustrated in Fig.4. The core outlet temperature is selected at 850°C. The hot primary flow exiting the core travels into the IHX in which it transfers the heat to the secondary flow. The cold primary flow exiting the IHX at 489°C is circulated by a primary circulator around the primary pressure boundary, while providing cooling to the pressure vessels, and then into the core. On the secondary side, the cycle pressure ratio is 3.0 with a turbine inlet temperature of 775°C. The relatively high pressure ratio is selected to allow the secondary gas stream to enter the IHX at a low temperature in order to facilitate the cooling of the primary pressure vessels. Compression intercooling is used in order to obtain high cycle efficiency. Fig.5 illustrates the intercooling effect on the cycle efficiency. Two stages of intercooling are selected since the cycle efficiency gains little when the number of intercooling stages becomes three or more.

Table 1 summarizes major plant design and performance data. The plant busbar efficiency is 45.1% based on the selected cycle parameters.



3.2 Reactor Module

The reactor module consists of the reactor, the IHX, the primary circulator, and a reactor shutdown cooling system (RSCS). The reactor core, core supporting structure, and control and refueling devices are located in the reactor vessel (DOE, 1986). The current reactor employs a prismatic core; however, the pebble-bed core is equally applicable. A helical-coil heat exchanger tube bundle is contained in the IHX vessel. The reactor and IHX steel vessels are interconnected by a coaxial crossduct. The primary circulator and the RSCS are mounted at the bottom of the IHX and reactor vessels, respectively. The RSCS provides cooling for decay heat removal during reactor shutdown.

During normal operation, the reactor vessel is externally cooled by a reactor cavity cooling panel mounted on the reactor silo wall while the IHX vessel is internally insulated from the gas stream to maintain low vessel temperature and to reduce heat loss. The design temperature and pressure for the reactor and IHX vessels are 454°C and 7.70 MPa, including added design margins. The ferritic steel 12Cr-1Mo-V or 9Cr-1Mo-V may be used for vessel construction; the former is a qualified material in the German KTA HTR high temperature design code (Schubert and Nickel, 1991) while the latter has been recently submitted for code approval of the ASME Code Case N-47. Both these materials are applicable for temperatures up to 650°C. The inner diameter is 6.55 m for the reactor vessel and 4.35 m for the IHX vessel; the thickness requirements for the reactor and IHX vessels are respectively 16 cm and 11 cm using either of these steels.

It is possible to redesign the current reactor internal design by modifying the core inlet flowpath or power distribution in the side reflector so that the reactor vessel design temperature could be reduced to 425°C or below. In this case, the low chromium steel 2¹/4Cr-1Mo, which is a qualified material for temperatures up to 600°C in the ASME Code Case N-47 and the Japanese HTTR high temperature design code (Hada, et al., 1991), may be employed for vessel construction.

The IHX helical-coil tube bundle, as shown in Fig.6, is a proven construction for the MGR high temperature environment. The tube bundle consists of 2400 tubes with a tube dimension of 23.0x2.0 mm (tube outer diameter x wall thickness) and an average tube length of 25 m, fabricated of Ni-base alloy Inconel-617. The overall size of the tube bundle is 3.52 m outer diameter and 6.0 m height.

The maximum metal temperature of the heat transfer tubes is 810°C. Several design techniques have been employed to ensure high temperature structural reliability. First, the pressure balance across the heat transfer tubes is chosen to minimize the pressure loads on the heat transfer tubes; only a slightly higher pressure (0.40 MPa) of the secondary flow than the primary coolant is maintained for leakage purpose. Second, the helical-coil tube bundle has an axially-symmetrical construction which, in combination with a specially-designed center pipe for tube support, allows free thermal

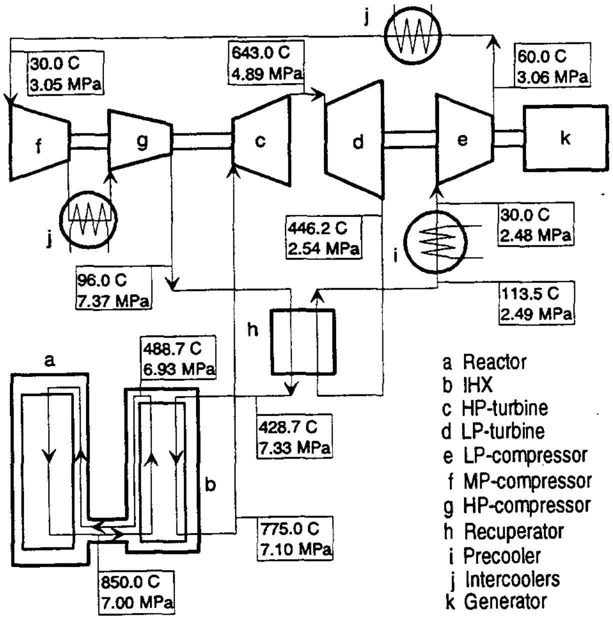


FIG.4: CYCLE DESIGN FOR THE MGR-GTI

Table 1: DESIGN DATA OF MGR GAS TURBINE PLANTS

	Indirect-Cycle	Direct-Cycle
Reactor System		
Reactor power (MWth)	350	350
Core temperature (inlet/outlet) (C)	494/850	576/850
Core coolant flowrate (kg/s)	186.6	245.6
Core coolant pressure (MPa)	7.0	8.0
Reactor system pressure drop (%)	1.50	1.13
IHX effectiveness	0.86	N/A
Gas Turbine System		
Turbine flowrate (kg/s)	194.6	245.6
Turbine inlet pressure (MPa)	7.06	7.80
Compressor inlet pressure (MPa)	2.48	3.71
Cycle pressure ratio	3.0	2.2
Cycle pressure drop (%)	7.30	8.83
Cycle intercooling	2 stages	No
Recuperator effectiveness	0.95	0.95
Turbine inlet temperature (C)	775	850
Turbine polytropic efficiency	0.92	0.93
Compressor inlet temperature (C)	30	30
Compressor polytropic efficiency	0.90	0.93
Generator efficiency	0.985	0.985
Overall Plant Performance		
Gross power output (MWe)	166.0	163.0
Station loads (MWe)	8.1	3.5
Net power output (MWe)	157.9	159.5
Net Plant Efficiency (%)	45.1	45.6

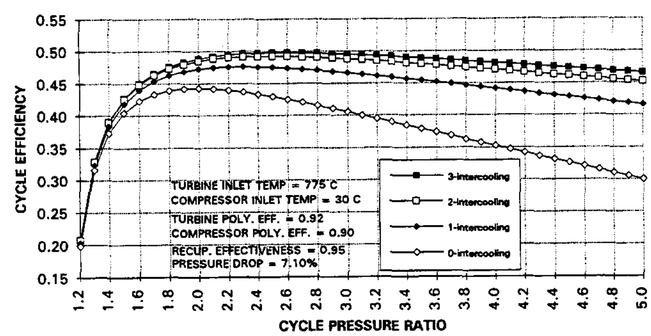


FIG. 5: INTERCOOLING EFFECTS ON CYCLE EFFICIENCY

expansion of the tube bundle in both radial and axial directions. This permits rapid thermal stress dissipation and reduces low-cyclic fatigue in the heat transfer tubes. The high-cyclic fatigue is controlled by the radiation plates inserted between the tube rows, which proves to be an effective means in suppressing flow-induced tube vibration caused by flow vortex shedding in high pressure flow. A detailed design study for the IHX concluded that the current tube bundle is capable of satisfying the lifetime operation (40 years) and the structural reliability requirements (Yan, et al., 1992b).

Further simplification of the reactor module can be achieved in an in-line vessel configuration as shown in Fig.7. The IHX tube bundle is now installed below the reactor core, and the primary circulator and the RSCS are mounted horizontally on the vessel sideline. The advantage of this configuration is the elimination of the coaxial crossduct and a separate IHX vessel and silo; considerable cost saving can be generated. Mounting the IHX tube bundle below the core elevation prevents hot gas natural circulation and harmful heatup in the tube bundle during primary system pressurized conduction colddown. In addition, the in-line vessel configuration also allows for improved seismic responses over the side-by-side vessel structure. Maintenance requirements are minimal for the He/He tube bundle. The small individual tube headers will provide access to the tube bundle for minor inspection and maintenance such as tube plugging. Should tube bundle replacement be necessary, which is highly unlikely during plant lifetime, the tube bundle can be removed through the vessel bottom flange and lifted up through the auxiliary silo on the right to the grade level.

3.3 Balance of Plant

The balance of plant (BOP) is a conventional gas turbine power conversion station located outside the nuclear island, as shown in Fig.3. A concentric duct is used to transport the secondary helium flow to and from the IHX to receive heat from the primary coolant. The turbo-generator set is on the floor level, below which are the heat exchangers, ducts, and auxiliary equipment for the gas turbine plant. The scale in Fig.3 clearly indicates the small dimensions of the closed-cycle helium turbine system.

Turbomachinery

Although the basic aerodynamic design techniques used for air turbines are applicable to helium machines, it is important to recognize that the drastically different thermodynamic properties of helium from those of air require some unique design considerations.

First, because of the larger specific heat of helium than air; the specific enthalpy change between two given cycle temperature limits is approximately five times larger for helium than for air, and therefore many more stages seem to be required for a helium rotor under a similar stage loading. However, because the pressure ratio of a recuperated helium cycle is relatively small, the number of helium rotor stages and thus the rotor length are comparable with air turbines. Second, the sonic speed of helium is about three times that of air, which completely eliminates the Mach number limitations that are often imposed on air turbine designs. Two benefits follow: (1) Much higher shaft rotational speed is permitted, resulting in considerably smaller rotor size. The highest possible rotational speed is only limited by allowable blade stresses; and (2) The helium flow benefits from low Mach Numbers (<0.35) and high Reynolds numbers (>2x10^6), both of which augment turbomachine efficiency.

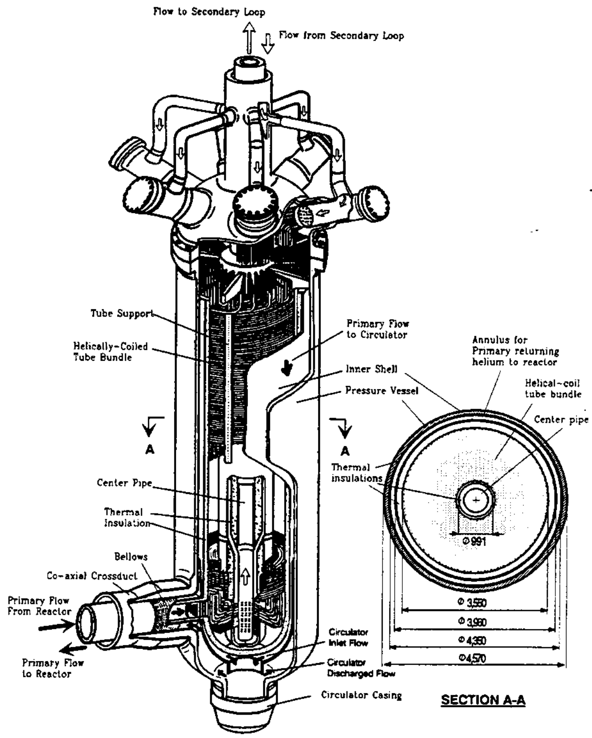


FIG. 6: IHX FOR THE INDIRECT CYCLE MGR-GTI

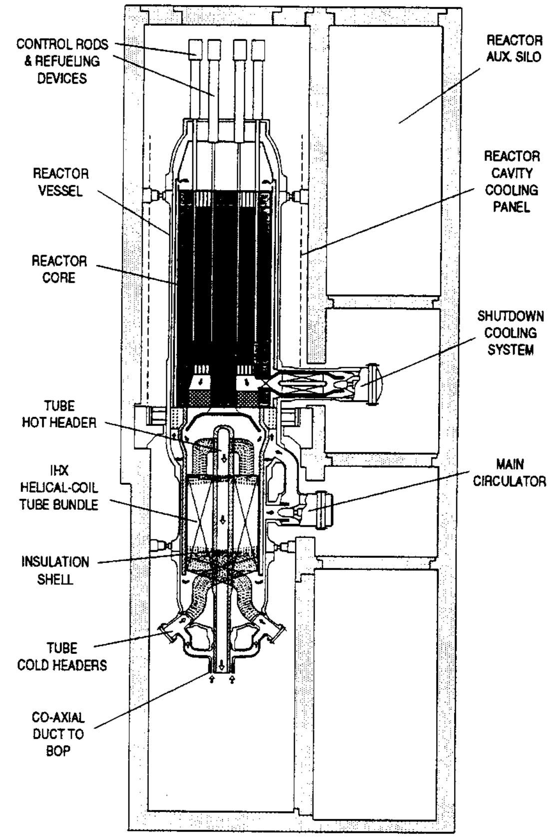


FIG. 7: SINGLE-VESSEL MGR MODULE DESIGN

The turbomachine for the MGR-GTI employs a twin-shaft rotor configuration consisting of a high-speed (HS) shaft with a high-pressure (HP) turbine driving mid-pressure (MP) and HP-compressors at 11,000 rpm, and a low-speed (LS) shaft with a low-pressure (LP) turbine driving an LP-compressor and a generator at synchronous speed. A schematic of the turbomachine rotor design is shown in Fig.8. The twin-shaft arrangement enables optimum aerodynamic blade design of the high speed machines while avoiding use of frequency conversion equipment for output power. The LP-compressor connected with the LP-turbine is intended for generator overspeed protection in case of loss of electric load, which is discussed later in the paper. In addition, the startup of the turbomachine can be provided by rotating the generator as a motor to drive the LP-compressor to gradually buildup the system pressure ratio and to power the HS-shaft machines with the aid of flow bypass control valves; no separate startup motor is necessary.

The compressors and turbines are all axial-flow machines. The total 3.0 cycle compression ratio is partitioned among the three compressors with pressure ratios of 1.24 for the LP-compressor and 1.56 for each of the MP- and HP-compressors. The pressure ratios of the HP- and LP-turbines are respectively 1.44 and 1.93. The total number of stages is 19 (7+5+7) for the three compressors and 13 (3+10) for the two turbines. The maximum blade tip diameters of the three rotors on the HS-shaft are all under one meter whereas they are 1.46 m and 1.72 m respectively for the LP-compressor and LP-turbine rotors on the LS-shaft. The overall rotor length is less than five meters for each of the HS- and LS-rotor groups. The maximum blade metal temperature is the highest at the first HP-turbine stage (765°C), for which conventional Nickel-base alloys such as IN 738 LC are adequate and no blade cooling is required.

The blade design for the turbines and compressors is based on a 50% mean stage reaction to enable the highest possible efficiency at the design point since the flow velocity conditions in this machine are likely to operate near the design point at all times. The hub-to-tip ratios of all rotor stages are controlled below 0.89. Small tip clearance can be used owing to the modest blade deformation as a result of the small rotor dimensions. These design measures, in combination with the high Reynolds numbers and low Mach numbers of the flow, allow for 90% polytropic efficiency for the compressors and 92% polytropic efficiency for the turbines.

Recuperator

The recuperator design is shown in Fig.9. The recuperator core uses a compact plate-fin strip-fin heat transfer surface, fabricated of 304 stainless steel, in a counterflow arrangement. The clean helium environment enables an identical heat transfer surface with compact geometry to be used for both sides of the recuperator to enhance the overall heat transfer coefficient. The core consists of two modules with a total volume of 22.1 m^3 and weighs approximately 57 tons.

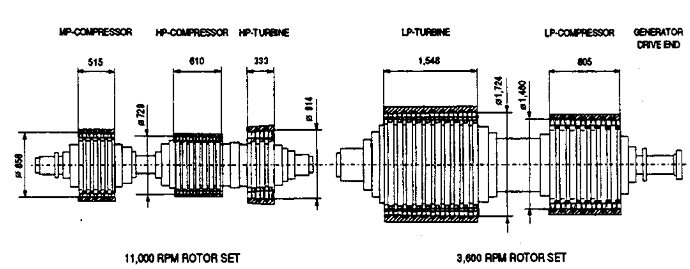


FIG. 8: 169 MWe HELIUM TURBOMACHINE ROTOR

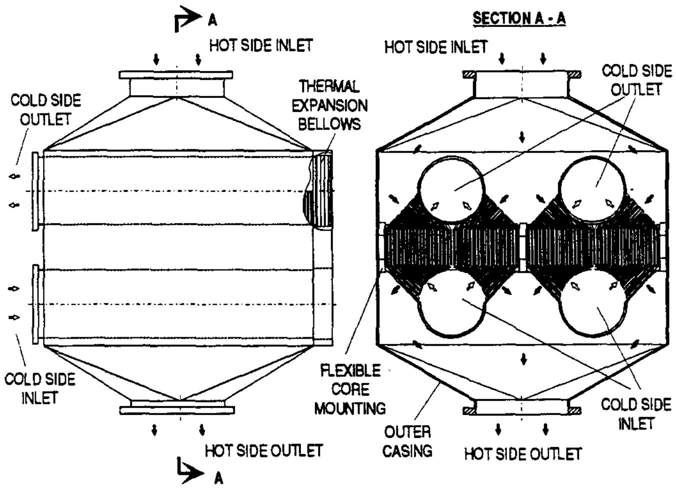


FIG. 9: COMPACT PLATE-FIN RECUPERATOR

The temperature effectiveness is 0.95 with a total pressure drop of 1.8%.

The cold flow at 96°C from the compressor enters the two inlet circular plena at the core bottom and is distributed into oblique inlet channels and then into the vertical counterflow section in which most of the heat transfer occurs. Leaving the core through the top oblique exiting channels, the cold helium at about 429°C enters the two outlet circular plena on the top of the core. The hot helium at about 446°C from the turbine exhaust is discarded into the recuperator through the inlet on the top shell and is distributed over the core frontal area through baffles. The hot helium then passes the core and leaves the recuperator at 114°C through the outlet on the bottom of the outer casing. Insulation on the inner wall of the casing may be used to isolate the hot helium from the steel casing.

The flow pressure in the core is internally balanced so that flexible core mounting can be used between the core and casing to allow independent thermal expansion, thus limiting thermal fatigue in the core. Similarly, expansion bellows are used to allow free thermal movement of the core in longitudinal direction relative to the casing.

The technology supporting the current recuperator design is mature. Recuperators of similar construction are being operated in many industrial gas turbine installations. However, compared to those applications, the current recuperator will present a simpler design challenge because of the clean working environment and enhanced heat transfer of helium.

Precooler and intercoolers

The precooler design shown in Fig.10 is a crossflow tube-and-shell heat exchanger. The helium flows in the shell passing a bank of tubes, inside which the cooling water flows, and is cooled from 114°C to 30°C with a pressure drop of 0.4%. The baffles may be used to help evenly distribute the flow over the core frontal area and to reduce pressure losses at the shell entrance and exit. The tube bundle consists of smooth circular tubes of 12.7 mm O.D. with a total volume of 8.5 m^3, manufactured of inexpensive carbon or low-alloy steels. A 15% design margin for water-side fouling has been added in the tube bundle.

The two intercoolers have a construction identical to the precooler, but with smaller core volumes because of their reduced heat duties and increasingly pressurized flow. The core volume is 4.2 m^3 for the first intercooler between the LP- and MP-compressors

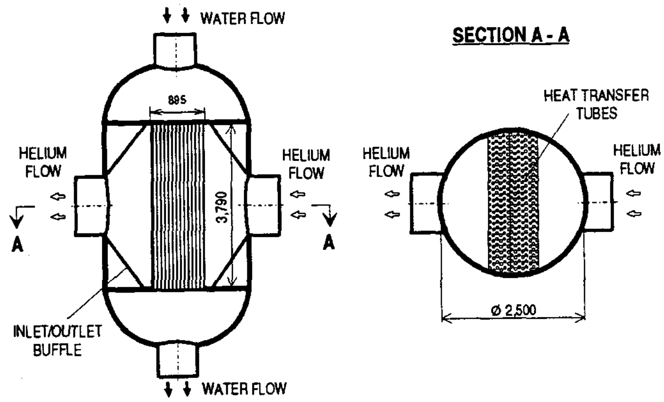


FIG. 10: TUBE-AND-SHELL CROSSFLOW PRECOOLER

and 6.4 m^3 for the second one between the MP- and HP-compressors. The helium flow is cooled from 60°C to 30°C in the first intercooler and from 96°C to 30°C in the second intercooler with a 0.3% pressure drop in each.

3.4 Plant Control

The inherent safety of the reactor greatly simplifies the safety requirements for the plant control system. As a result, the objective of plant control becomes achieving optimum control characteristics during normal operation and adequate investment protection in case of accidents. Since MGR gas turbine control has been extensively studied and well documented (Yan and Lidsky, 1990 and 1991a), no detailed description is to be given in this paper, except for a brief discussion of turbo-generator overspeed events which are relevant to helium turbine designs.

In the case of loss of generator load, two separate control valves, which bypass gas flow away from the two turbines, are quickly activated to protect the two turbine rotors from excessive overspeed. The control response to this event has been simulated with a full-scale plant model implemented by a computer code, GTSim (Yan, 1991). Fig.11(a) shows the simulation results. As the electric load is disconnected at time = 1 second, the two bypass valves of 28 cm diameter for the HP-turbine and 33 cm diameter for the LP-turbine are activated to control the overspeed of the two rotors. The rotors are quickly returned to steady idle running conditions at nominal speeds after a brief period of overspeed. The maximum overspeed is 4% for the HS-rotor and 10% for the LS-rotor, well within acceptable limits.

Control redundancy is incorporated to assure control reliability for the overspeed protection. However, should active control fail, passive protection is ultimately provided by the compressors on the respective turbine rotors. In the turbomachine design, most of the cycle compression duty is given to the two compressors on the HS-rotor because of the desire to design the compressors at the high rotational speed. However, the LP-compressor, which only provides a small fraction of the total compression work in design conditions, is purposely positioned on the LS-rotor for generator overspeed protection in case of loss of generator load. Such passive overspeed control is demonstrated in a simulation, as shown in Fig.11(b), in which the bypass valves are intentionally disabled. After the load is disconnected at 1 second, the LS-rotor experiences rapid overspeed; however, the seven stages of the LP-compressor provide increasingly large loading, due to increased Mach number and stage losses, as the overspeed increases. A self-sustained overspeed limit

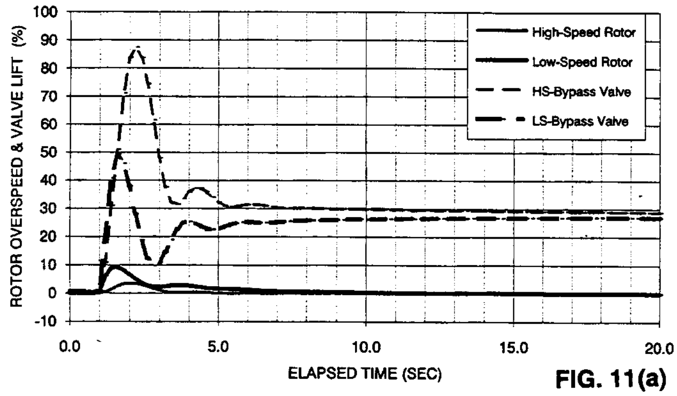


FIG. 11(a)

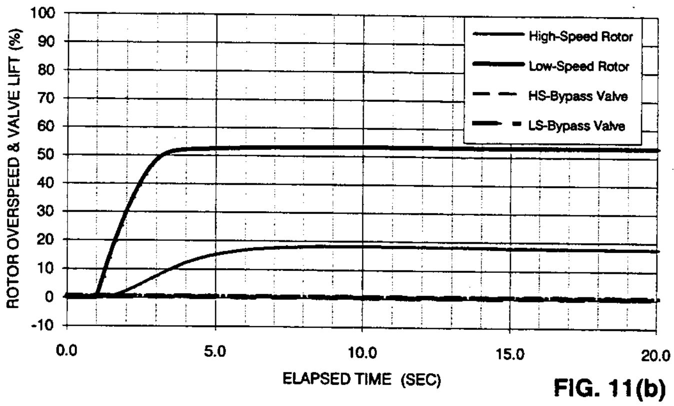


FIG. 11(b)

FIG.11: TURBINE RESPONSE TO LOSS OF ELECTRIC LOAD
(a) WITH NORMAL PROTECTIVE CONTROL
(b) WITHOUT ACTIVE CONTROL

of 53% is eventually reached within 3 seconds when the LP-compressor load has counterbalanced the LP-turbine power. The overspeed of the HS-rotor (18%) is much smaller and takes longer time to peak owing to the larger compressor loads on the rotor and to cycle thermodynamic damping. These results provide a design basis for the turbo-generator that is required to maintain the integrity of the rotors at the respective overspeed limits.

3.5 Plant Costs

Based on conventional cost forecasting methodology and existing data base for nuclear plant components (ORNL, 1990, GCRA, 1990, and ESEERCO, 1992), the costs (in 1990$) of series-production MGR-GTI plants have been estimated. Capital costs are projected to be approximately 1790$/KWe for a four reactor plant totaling 1400 MWth thermal power or 632 MWe electric output. The cost of electricity is estimated at 39 mills/KWh, including capital costs, operation and maintenance costs, fuel costs, as well as decommissioning costs. The capacity factor is assumed at 88% based on the predicted scheduled and forced outages. These cost data, although preliminary, indicate an economic potential that is competitive with other nuclear or fossil plant options.

4. DIRECT CYCLE PLANT DESIGN - MGR-GT

While the indirect cycle plant described above presents the fewest licensing difficulties for near-term deployment, an ultimate gas turbine power plant will be a direct cycle plant that offers the maximum design simplicity and economical potential (Yan and Lidsky, 1991b). Such a design is shown in Fig.12 with its major design data listed in Table 1.

In the direct cycle plant, the entire gas turbine power conversion system is integrated into the reactor primary system in a single vessel configuration, thus eliminating the IHX and its temperature drop, making a higher temperature available to the turbine. The compressor provides flow circulation in the circuit, obviating a separate circulator. Reactor vessel cooling is conveniently provided by the compressor exhaust flow, which flows upward in the annulus between the reactor pressure vessel and core barrel, so that the cycle can now be operated at the peak efficiency pressure ratio which is only 2.2. Intercooling is not recommended because the efficiency gain at the lower cycle pressure ratio is small and not worth the additional system complexity. In spite of such a simple cycle configuration, the plant net efficiency is 45.6% with 850°C core outlet temperature and 0.95 recuperator effectiveness.

The direct cycle plant takes advantage of the compactness of the closed-cycle helium turbine system to integrate the BOP into the reactor vessel. The elimination of the costly IHX and the simplified system packaging are expected to further improve the plant economics.

The direct cycle plant design does not appear to require any significant technology development before its commercial deployment. However, several design issues may require verification for licensing purposes. They include confirmation of satisfactory fission product behavior with regard to turbine contamination and the accumulation of successful helium turbine operating experience. While the problems do not vitiate the viability of the direct cycle design, immediate deployment is difficult because seeking persuasive answers to these issues is time consuming and costly. Although long-term developmental work might be undertaken to acquire the necessary data, a simple and economic solution is the deployment and operation of indirect cycle plants, which will naturally provide the technical answers and financial incentives for the direct cycle design.

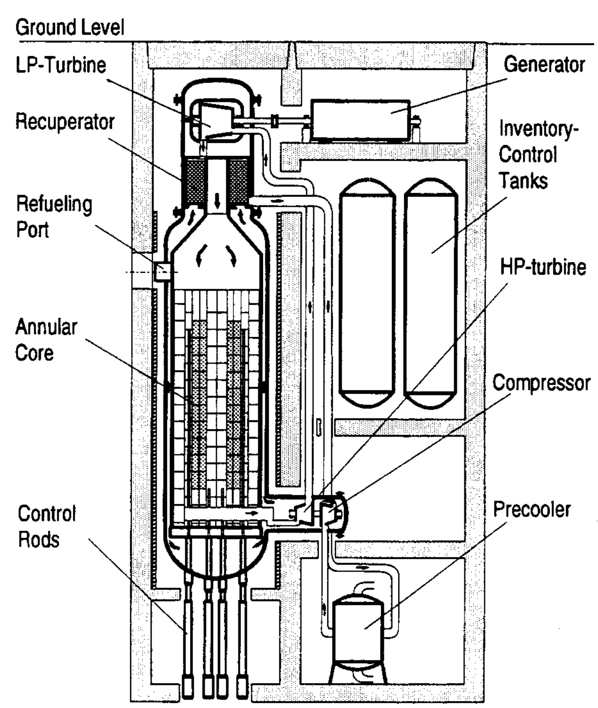


FIG.12: DIRECT CYCLE MGR-GT POWER PLANT

5. SUMMARY

The closed-cycle helium turbine combined with the MGR is an attractive, near-term power plant option from both technical and economic standpoints. The major system features are summarized as follows:

Inherent Safety: The MGR, relying on physical laws rather than engineering safety control systems, enables a desirable "Licensing by Tests" approach to nuclear regulation and offers the highest possibility of social and political acceptance for nuclear power.

Reliance on Today's Technology: Plant designs can be realized using present fuels and currently available materials and design codes. The helium gas turbine system is supported by established industrial gas turbine and heat exchanger technologies.

High Efficiency: A busbar efficiency in excess of 45% is achievable at 850°C core outlet temperature.

Potential for Technology Growth: Expected improvement of existing reactor fuels and the development of new high temperature materials will likely permit use of higher core outlet temperatures (950°C or higher) to enable plant efficiencies to exceed 50%.

Operating Flexibility: The plant is suitable to either base or isolated power operation without serious economic penalty and thermal cycling limitations. High temperature heat rejection allows economic dry cooling and siting freedom and makes waste heat recovery possible without impact on generating efficiency.

Promising Economics: The plant offers competitive economics through a combination of high performance, design simplicity, and low-cost O&M.

Evolving Commercialization Strategy: The two plant designs presented in this paper provide an economic incentive and a straightforward path for the timely incremental commercialization of the MGR gas turbine technology.

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