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EVALUATION OF THE GAS TURBINE MODULAR HELIUM REACTOR

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ABSTRACT

Recent advances in gas-turbine and heat exchanger technology have enhanced the potential for a Modular Helium Reactor (MHR) incorporating a direct gas turbine (Brayton) cycle for power conversion. The resulting "GT-MHR" power plant combines the high temperature capabilities of the MHR with the efficiency and reliability of modern gas turbines. While the passive safety features of the steam cycle MHR (SC-MHR) are retained, generation efficiencies are projected to be in the range of 48% and steam power conversion systems, with their attendant complexities, are eliminated. Power costs are projected to be reduced by about 20%, relative to the SC-MHR or coal. The GT-MHR will require additional technical development; however, no fundamental feasibility issues have been identified.

This report documents the second, and final, phase of a two-part evaluation that concluded with a unanimous recommendation that the direct cycle (DC) variant of the GT-MHR be established as the commercial objective of the U.S. Gas-Cooled Reactor Program. This recommendation has been endorsed by industrial and utility participants and accepted by the U.S. Department of Energy (DOE).

The Phase I evaluation was initiated with an International Workshop on the GT-MHR, which was held at the Massachusetts Institute of Technology (MIT) in June 1991. Following the workshop, an initial preconceptual design was developed for a direct cycle GT-MHR and subjected to a series of performance, O&M, safety and economic evaluations. An important conclusion of the initial evaluation was that the peak fuel temperatures in a highly recuperated direct cycle MHR-GT (operating at 850°) would be essentially the same as those for the steam cycle reactor. Also, advances since 1980 in turbomachinery and heat exchanger technology were found to provide the basis for significant efficiency improvements relative to the earlier designs. Sufficient technical and economic advantages were identified for the GT-MHR that a more detailed assessment was recommended, to include an comparative evaluation of the direct cycle (DC) and indirect gas turbine cycle (IDC) options.

The Phase II effort, documented herein, concluded that the DC GT-MHR offers substantial technical and economic advantages over both the IDC and SC systems. Both the DC and IDC were found to offer safety advantages, relative to the SC, due to elimination of the potential for water ingress during power operations. This is the dominant consequence event for the SC. The IDC was judged to require somewhat less development than the direct cycle, while the SC, which has the greatest technology base, incurs the least development cost and risk. While the technical and licensing requirements for the DC were more demanding, they were judged to be incremental and feasible. Moreover, the DC offers significant performance and cost improvements over the other two concepts. Overall, the latter were found to justify the additional development needs.

The Phase II evaluation reported herein was sponsored by the U.S. Department of Energy, (DOE), with additional support from the Empire State Electric Energy Research Corporation (ESEERCO), Oak Ridge National Laboratory (ORNL) and Gas-Cooled Reactor Associates (GCRA). ESEERCO was the principal sponsor of the Phase I study and the International Workshop on the GT-MHR.

ACKNOWLEDGMENTS

The evaluation of the Gas Turbine Modular Helium Reactor (GT-MHR), described in this report, has culminated in the redirection of the U.S. High Temperature Gas-Cooled Reactor Program to the development of the GT-MHR as its initial commercial objective. It is believed by participants that the successful development of the GT-MHR will lead to substantial improvements in the safety and economics of nuclear energy in the early 21st century. In this context, it is particularly appropriate to acknowledge the sponsors and major contributors to the GT-MHR evaluation.

SPONSORS

The U.S. Department of Energy (DOE) was the majority sponsor of the GT-MHR evaluation described in this report. DOE also provided significant support for the initial evaluation sponsored by the Empire State Electric Energy Research Corporation (ESEERCO). The leadership of HTGR Program Director Peter M. Williams was a crucial element in the success of both. Support for DOE's management of the evaluation was provided by Pete Karcz of the DOE staff along with Ray Mills, Larry Brey and Fred Swart of DOE's Plant Design Control Office.

ESEERCO was the principal cosponsor of the work described in this report and was the majority sponsor of the initial evaluation of the GT-MHR and the International Workshop on the GT-MHR. ESEERCO's support for the initial evaluation was instrumental in highlighting the potential of the GT-MHR and their foresight directly led to the current effort. The efforts of Project Manager Kevin Zugibe of Orange and Rockland Utilities and James M. Burger of the ESEERCO staff, along with the continuing support of the ESEERCO Nuclear Power Committee are particularly acknowledged.

Gas-Cooled Reactor Associates proposed the initial evaluation of the GT-MHR and provided planning and coordination support to both ESEERCO and DOE. Scott R. Penfield, Jr. of the GCRA staff led the Evaluation Team in both the initial and current evaluation. He was also responsible for integrating and editing the Proceedings of the International Workshop on the MHR and the final reports for both phases. The GCRA Utility Working Group on Operations was primarily responsible for the O&M evaluation, with notable support from Joseph Cooney (retired from Philadelphia Electric Company) and Harold Gotschall of the GCRA staff. Gregory R. Pause of the GCRA staff integrated the overall cost inputs and completed the competitive economic assessment.

Oak Ridge National Laboratory (ORNL) conducted an independent, internally funded assessment of the GT-MHR that was closely coordinated with both the initial and current evaluations. This effort was led by Howard T. Kerr. Additional support was provided through ORNL's participation in the DOE HTGR Program, under the direction of Phil Rittenhouse. ORNL both proposed and participated in the evaluation of design alternatives and were instrumental in the selection of 9Cr-1Mo-V as the material for the primary pressure vessels.

James C. Conklin provided valuable input regarding the characteristics of heat exchangers and Robert S. Holcomb provided similar input for turbomachines. Syd J. Ball performed independent analyses of core conduction cooldown events and Lowell Reed provided inputs to the cost assessments. Special mention is due to Paul R. Kasten (ORNL Consultant), who supported the current evaluation and was a major contributor to the Proceedings of the International Workshop on the MHR in the initial evaluation.

DOE CONTRACTORS

Within the context of the DOE Program, the following organizations contributed to the evaluation effort.

ABB Combustion Engineering Nuclear Power (ABB)

ABB/CENP developed the precooler and intercooler designs for the direct cycle (DC) and the intermediate heat exchanger (IHx) design for the indirect cycle (IDC). They also developed the vessel system design for both concepts. In addition, ABB/Power Generation provided important input to the designs, assessments and cost estimates of the turbomachinery for both the DC and IDC concepts. They were supported by Strategic Power Systems, which developed reliability estimates for the gas turbines. The ABB effort was led by Sten A. Caspersson. Gerald DiLauro, Hans Frutschi, John P. Cook, Devendra D. Mehta, Axel Lubkowitz and N.U. Karim were key contributors to the design and evaluation efforts.

Allied Signal

Allied Signal developed initial designs for both the DC and IDC recuperators. The Allied Signal effort was led by Karl Kretzinger.

Bechtel

Bechtel developed the architectural and engineering design of the nuclear island systems, structures and components for both the DC and IDC designs. They also developed the DC GT-MHR plant site plan. Bechtel's effort was led by David A. Dilling and Sunil Ghose.

General Atomics (GA)

GA was responsible for the overall integration and optimization of the DC and IDC systems designs. They developed the reactor and power conversion module designs for the DC, as well as the reactor design for the IDC. GA was also a major contributor to assessments of performance, safety and investment protection, operational characteristics, reliability and maintainability. They provided input to economic evaluations and were a major contributor to the writing of this report. The GA effort was led by George C. Bramblett and Fred Silady. Significant contributions were made by Guido M. Baccaglioni, Thomas D. Dunn, Klaus T. Etzel, David D. Kapich (Consultant to GA), Anthony J. Neylan, Robert W. Schleicher, Arnold A. Schwartz, and James B. Zgliczynski. Special mention is due to Colin F. McDonald, who proposed the GT-MHR concept in 1986 and contributed to the current evaluation.

General Electric Aircraft Engines (GE)

GE conducted an evaluation of the DC turbomachinery that provided added confidence in the feasibility of the design. The GE effort was led by Carol Russo and Clay Haubert.

Massachusetts Institute of Technology (MIT)

Earlier work by MIT (circa 1987) was an important precursor of the GT-MHR evaluations. As part of the initial evaluation, MIT hosted and was a primary contributor to the organization of the International Workshop on the GT-MHR. Notable contributions to the current effort included the initial layout of the IDC power conversion system and turbomachinery and the design and analysis of the IDC plant control system. MIT's participation was led by Professor Lawrence M. Lidsky, with major contributions by X. L. Yan.

Stone & Webster Engineering Corporation (SWEC)

SWEC developed the power conversion system design for the IDC, including the layouts of the major structures and the design of the secondary loop intercoolers and precooler. Walt Parker led the SWEC effort. John J. Elder was primarily responsible for the IDC design.

OTHER PARTICIPANTS

Several organizations and individuals, while not formal participants, none-the-less made significant contributions to the current effort by their inputs and advice. Included are Stephen L. Chapman of Magnetic Bearings, Inc., Michael LeComte of Framatome, Z. P. Tilliette, consultant to the Commissariat a l'Energie Atomique (CEA), and H. Reutler of KFA, Jülich.

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FORWARD

The evaluation documented in this report was undertaken by a small, but dedicated "Evaluation Team" comprising experts in gas cooled reactors and related power conversion technologies. Given the short schedule and intensive nature of the evaluation, it was necessary to develop many of the design details in parallel, without the opportunity for the normal iteration that would occur as part of the design process. Accordingly, the reader will note minor inconsistencies in design detail among the various sections of the report. These inconsistencies, which will be resolved in the furtherance of the design effort, are not considered important in terms of the evaluation process that was the focus of this effort.

It is also worth noting that the term "Modular Helium Reactor (MHR)" is introduced in this report as a replacement for the more descriptive, but cumbersome generic term "Modular High Temperature Gas-Cooled Reactor (MHTGR)". For the purposes of this report, these terms should be considered synonymous.

EXECUTIVE SUMMARY

The high temperature capability of the Modular Helium Reactor (MHR) coated particle fuel system leads to enhanced applications of nuclear energy that are not possible with contemporary technologies. Among these, a particularly attractive option is direct utilization of the helium primary coolant as the working fluid in a closed gas-turbine (Brayton) cycle. The resulting gas-turbine MHR (GT-MHR) power plant combines the high temperature capabilities of the MHR with the efficiency and reliability of modern gas turbines. It represents a significant advancement in the state of the art of nuclear power, offering generation efficiencies in the range of 48% and the elimination of the steam cycle with its attendant complexities.

This report documents the second, and final, phase of a two-part evaluation of the GT-MHR. The evaluation concluded with a unanimous recommendation that the direct cycle (DC) variant of the GT-MHR be established as the commercial objective of the U.S. Gas-Cooled Reactor Program. This recommendation has been endorsed by industrial and utility participants and accepted by the U.S. Department of Energy (DOE). It is believed by those participating in the evaluation that the successful development of the GT-MHR will lead to a significant improvement in the safety and competitive potential of nuclear energy in the early 21st century.

BACKGROUND

Interest in the potential for nuclear powered closed cycle gas turbines dates to the earliest days of commercial nuclear energy development. In the late 1970's, both the U.S. and European High Temperature Gas-Cooled Reactor (HTGR) programs explored large gas-turbine plant designs. However, the technology available at that time was inadequate and, in 1980, a decision was made to continue development of the steam cycle HTGR.

Consideration of gas-turbine applications was largely dormant until 1985, when the MHR replaced the large HTGR as the principal focus of the U.S. program. A paper presented by McDonald of General Atomics (GA) at the 1986 IECEC Conference (Ref. 1) was the first to suggest that the inherent and passive safety characteristics of the MHR, in combination with recent advancements in gas-turbine, magnetic bearing and heat exchanger technologies, would support a practical GT-MHR design. A thesis by Staudt, under the direction of Professor Lidsky at the Massachusetts Institute of Technology (MIT) subsequently outlined a pebble-bed GT-MHR design, based upon the German HTR Modul concept (Ref. 2). The MIT work confirmed the importance of heat exchanger and turbine technology improvements, and was particularly effective in communicating the potential for mid-40% plant efficiencies with a relatively modest increase in core outlet temperature to 850°C. Gas-Cooled Reactor Associates (GCRA) reviewed the GA and MIT results and recommended a more detailed evaluation to DOE and utility sponsors (Ref. 3).

The emergence of the GT-MHR began in earnest when the utilities comprising the Empire State Electric Energy Research Corporation (ESEERCO) agreed to serve as the majority sponsor

of an initial (Phase I) evaluation. A highlight of the Phase I evaluation was the International Workshop on the GT-MHR, which was attended by 94 invited experts, including 23 representatives from Europe, Japan and China. The proceedings of that workshop (Ref. 4) remains the most comprehensive reference on GT-MHR enabling technologies. The results of the Phase I evaluation (Ref. 5) identified compelling incentives for the GT-MHR, and led to the Phase II evaluation effort that is documented in this report. A key recommendation of the Phase I study was that a comparative evaluation be made of the DC and indirect cycle (IDC) versions of the GT-MHR. That comparison was a major focus of the Phase II evaluation.

TOP LEVEL REQUIREMENTS AND ASSUMPTIONS

The redirection of the U.S. Gas-Cooled Reactor Program to the Steam Cycle Modular Helium Reactor (SC-MHR) in August 1985 represented a consensus decision on the part of sponsors and participants to embrace a new approach to nuclear safety and economics. The essence of this approach may be summarized as follows:

- Apply the high temperature capabilities of the coated particle fuel system, in conjunction with the inherent and passive safety characteristics of smaller, modular gas-cooled reactor designs to eliminate the potential for accidents with offsite public impact.
 - In particular, the plant design shall preclude the need for off-site sheltering or evacuation as a means of protecting the public.
- Achieve competitive costs for electricity by:
 - A major simplification of plant systems, particularly by eliminating the need for the active safety systems found in large gas-cooled reactors and light water reactors.
 - Increased plant efficiency, made possible by the high temperature capability of the gas-cooled reactor's coated particle fuel system.
 - Achieving the benefits of factory fabrication and learning associated with standardized modular construction techniques.
- Provide additional flexibility for matching system growth and reduce the economic risk to owners by reducing the size of incremental deployments.

In developing the GT-MHR concept, a fundamental groundrule was that the inherent and passive safety concept embodied in the SC-MHR design would not be compromised. Other SC-MHR requirements (Ref. 6) were reviewed for applicability and modified, as appropriate, based on differences between the two technologies. Of these, three changes adopted for purposes of this initial evaluation are considered significant, and require further confirmation. These are:

- The site envelope wet bulb temperature value of 28°C (82°F) was retained for safety assessments, but the reference value recommended by EPRI [11°C (52°F)](Ref. 7) was used for economic assessments.

This change reflects the greater sensitivity of the Brayton cycle to ambient temperature conditions, compared to the SC-MHR. On cooler, dryer sites, the performance of the GT-

MHR would exceed the nominal value on an average basis. On warmer, humid sites, the nominal performance value would likely be attained on an average basis.

- The weekly load-following cycle specified for the SC-MHR was assumed to be an added cost option for the GT-MHR.

This change reflects the most likely operating mode of the GT-MHR. Although the evaluated design is capable of load following, maintaining efficient performance during load following operations would require additional equipment or a modification of the control scheme. As operation of the GT-MHR in a base-load mode is likely (due to its high efficiency and low projected operating costs), it was concluded that, while the plant design should accommodate the addition of load following capability, the associated equipment would be considered an added cost option for the purposes of this evaluation.

- Capacity factor enhancements were assumed to take precedence over extending the refueling interval in design tradeoffs.

The utility/user requirements for the SC-MHR specify a plant design capacity factor of 84%. The requirements also specify a minimum refueling interval of 18 months and a fuel design capacity factor of 88%. Core design evaluations to date have indicated that a refueling interval of 18 months is achievable with capacity factors of up to 88%. Beyond that level, a tradeoff is required between capacity factor and refueling interval.

SUMMARY TECHNICAL DESCRIPTION

The GT-MHR incorporates a closed gas-turbine cycle that is conceptually depicted in Figure 1. The gaseous working fluid, helium in this case, is first compressed and then heated in two steps. Initial heating takes place in a regenerative heat exchanger (the recuperator), which recovers energy from the low pressure part of the cycle. The nuclear reactor heat source provides energy input to the cycle and further raises the helium temperature to its maximum working level. The hot, high pressure helium is then expanded through the turbine, which drives the compressor and provides mechanical energy input to the generator. The lower pressure, but still hot helium from the turbine exhaust is routed to the recuperator, where it is used to preheat the high pressure helium from the compressor. Waste heat is then rejected in the precooler before the helium is returned to the compressor.

The relatively high energy content of the turbine exhaust is a barrier to high efficiency in gas turbine cycles. In conventional combined cycle gas-turbine (CCGT) systems, a steam bottoming cycle is utilized to recover part of this energy. However, the bottoming cycle adds complexity to the plant and the efficiency of the bottoming steam cycle is relatively low. In closed gas turbine cycles, the recuperator serves an analogous energy recovery function. With modern compact heat exchangers (e.g., plate fin), this function is accomplished with both high effectiveness and low pressure losses. Thus, for a given turbine inlet temperature, the efficiency of closed gas turbine cycles can be significantly higher than that of comparable combined cycles.

DIRECT CYCLE

INDIRECT CYCLE

Figure 1 GAS TURBINE MODULAR HELIUM REACTOR CONCEPT

Compressor intercooling can provide additional efficiency enhancement. This is accomplished by subdividing the compression step into two or more stages with a heat exchanger (intercooler) between the individual stages to remove part of the heat of compression. Less work is required to raise the pressure of the cooler helium in the higher stages of the compressor than would otherwise be the case.

Direct Cycle

The overall arrangement of an individual DC reactor and power conversion module is provided in Figure 2. The core and fuel of the GT-MHR are located in the reactor vessel and closely resemble those of the SC-MHR. The fuel is of the standard prismatic design (Figure 3); however, detailed differences will be required in fuel heavy metal loading, to accommodate the higher operating temperatures of the GT-MHR. As in the SC-MHR, the active core of the GT-MHR will consist of 84 columns of graphite fuel elements stacked 10 blocks high in an annular array (Figure 4). The active region of the core consists of three rings of fueled blocks. Graphite reflectors are provided both in the center region and outside of the active core. The annular configuration of the core is an important feature contributing to the passive safety concept of the MHR. The lower portion of the reactor vessel contains the shutdown cooling heat exchanger and the shutdown cooling circulator, both integral components of the shutdown cooling system.

The power conversion vessel (Figure 5) contains all equipment necessary to convert the thermal energy transported by the primary coolant helium into mechanical and, then, electrical energy. Included within the power conversion vessel are a gas turbine, recuperator, precooler, compressor, intercooler and a submerged generator. The cross-vessel provides the connection between the reactor and power conversion vessels and includes an internal hot duct.

In the reactor vessel, the helium flows downward through the core to a plenum located below the core. From the plenum, the heated helium is directed to the hot duct, which is contained within and concentric with the cross-vessel. The internal surface of the hot duct is insulated to restrict regenerative heat flow and to maintain the hot duct structure at an acceptable temperature level. Upon entering the power conversion vessel, the helium then expands upwards through the power turbine, which is vertically mounted on a central shaft. The central shaft also supports the compressor, and is directly coupled with the generator. At the power turbine outlet, the helium turns outward and flows downward through the recuperator modules, which are located in an annulus adjacent to the power conversion vessel shell. After leaving the recuperator, the helium flows downwards to and through the precooler. The cooled helium is then compressed in the compressor, which is mounted on the central shaft just below the power turbine. A single stage of intercooling is provided for improved cycle efficiency.

Leaving the compressor, the helium flows downward through an annulus between the precooler and the recuperator. The helium enters the recuperator at the bottom of the power conversion vessel and flows upwards, accepting heat from the downward flowing helium leaving the gas turbine. Shrouds and flow passages route the helium to the annulus of the cross-duct vessel, which is formed by the outside of the hot duct and the inside of the cross-duct vessel. The external surface of the cross vessel is also insulated to reduce heat losses. After re-entering the reactor vessel, the helium flows upward through ducts, attached to the outside of the core

Figure 2 DIRECT CYCLE GT-MHR

Figure 3 MHR FUEL SYSTEM

Figure 4 450 MWt MHR CORE CROSS SECTION

Figure 5 DIRECT CYCLE POWER CONVERSION MODULE

barrel, to the top of the core.

In the GT-MHR, the temperature of the helium returned to the reactor inlet is higher than that of the SC-MHR (due to recuperation). For this reason, 9Cr-1Mo-V has been selected as the material for the primary system vessels.

Indirect Cycle

The most fundamental difference between the DC and IDC GT-MHR concepts is the addition of an Intermediate Heat Exchanger (IHX) that separates the primary helium coolant from the helium working fluid of the power conversion cycle (Figure 1).

Each reactor module (Figure 6) incorporates side-by-side welded steel vessels located below grade, and connected by a cross vessel containing a concentric hot duct. One of the vessels contains the reactor and the shutdown cooling system and the other contains the IHX and main helium circulator.

The reactor is substantially the same as that described for the DC. The IHX is of a compact helical tube design. The main circulator is located at the top of the vessel to provide easy access for maintenance or replacement. All safety-related structures and components are located within the Nuclear Island, permitting procurement and construction of the remaining parts of the plant to conventional standards.

The gas-turbine-generator unit is located at grade level within the energy conversion area, and is connected to the IHX vessel through two large ducts, as conceptually shown in Figure 7. The Power Conversion System (PCS) comprises a twin-shaft turbine-compressor-generator unit, a precooler, two intercoolers, plate-fin recuperators, and the interconnecting ducting. These components are readily accessible for maintenance or replacement.

Enhanced Steam Cycle

An enhanced steam cycle design was also evaluated that provides the same margins for core related accidents as found in the two GT-MHR designs. The intent was to provide a "level playing field" for the SC-MHR in comparisons with the two gas turbine concepts. The enhanced steam cycle reactor power is 475 MWt and was achieved by increasing the power density by 5.6%. The existing 84 column core and reactor vessel configurations are utilized without change. The size of the circulator, steam generator, and possibly the shutdown cooling system components must be increased to accommodate the increased power and/or its effects. The larger steam generator may also result in a small increase in the silo diameter.

PERFORMANCE

Key performance parameters for the reference 450 MWt SC-MHR, the enhanced 475 MWt SC-MHR and the respective GT-MHR options are provided in Table 1. The enhanced steam cycle design offers an increase in power output relative to the reference 450 MWt SC-MHR, based on the higher thermal power generated in the core. Both GT-MHR concepts offer significant efficiency and power enhancements, without an increase in core thermal power.

Figure 6 INDIRECT CYCLE GAS TURBINE REACTOR MODULE

Figure 7 INDIRECT CYCLE REACTOR AND TURBOMACHINERY BUILDINGS

Table 1 KEY PERFORMANCE PARAMETERS

	STEAM CYCLE	ENHANCED STEAM CYCLE	DIRECT CYCLE	INDIRECT CYCLE
REACTOR POWER (MWt)	450	475	450	450
REACTOR OUTLET TEMP (°C)	700	700	850	850
TURBINE INLET TEMP (°C)	540	540	850	775
TURBINE INLET PRESSURE (MPa)	17.34	17.34	7.03	7.24
AMBIENT TEMP (°C)	28	28	18	18
CYCLE EFFICIENCY (%)	42.7	42.7	50.3	47.6
FREQUENCY CONVERTER EFFICIENCY (%)			98.5	
NET ELECTRICAL OUTPUT (MWe)	173	183	213	197
NET PLANT EFFICIENCY (%)	38.5	38.5	47.3	43.8
NET ELECTRICAL OUTPUT @ 11°C AMBIENT TEMP (MWe)			217	202
NET PLANT EFFICIENCY @ 11°C (%)			48.3	44.8

OPERATIONS ASSESSMENTS

There are significant operational differences among the DC, IDC and SC designs; however, all were found to be acceptable. The primary circuits of the IDC and SC are similar, in that helium flow through the core is provided by the main circulator, which is controlled in a manner that is independent of, but coordinated with the secondary power conversion cycle conditions. This capability provides a great deal of flexibility in operations, and core power output is varied simply by increasing or decreasing the helium flow through the core.

The principal differences between the IDC and SC are found in the secondary loop. In the case of the SC, the control strategy centers on coordination of the feedwater supply and the steam turbine admission valves. By contrast, the IDC uses turbine bypass valves as the primary method to regulate the speed of the turbomachinery.

The control approach for the DC power conversion system is similar to that for the IDC secondary loop. A significant difference in the DC, however, is that helium flow through the

core is provided by the turbomachine. This implies a tighter coupling between the reactor and power conversion system operations than is found in either the IDC or SC designs.

To explore the practical differences among the various concepts, the evaluation effort centered on potentially distinguishing issues. These included startup and rise to power, response to representative normal and upset transients, shutdown and refueling.

As is the case with other gas-turbine systems, startup of the DC and IDC plants requires rotation of the turbomachine to establish the flow and pressure differences required for net power output. The motive force for this startup rotation is provided by motoring the generator in both cases. In the IDC, the startup process is somewhat more complex, due to the twin shaft turbomachinery configuration that has been selected for that concept. The startup time for either gas-turbine cycle is expected to be shorter than that of the SC.

The transient response of the DC was a particular focus of the operational evaluation, due to concern with the tighter coupling of the reactor and power conversion systems. A real-time control system simulation was used to model both normal operating and upset transients. These included negative and positive load ramps at 5%/min over the range of 25%-100% of rated power, a negative load step change of 10% and loss of electrical grid load. In all cases, the plant remained controllable and within limits. The initial indication is that the plant can revert to house loads following loss of electrical grid load. Because the control strategy is still evolving, these conclusions must be considered preliminary.

The IDC was similarly evaluated for loss of electrical grid load. The plant response was readily controllable and within limits. Normal operating transients were not run for the IDC, since the operational characteristics of the primary system are closer to the SC.

Neither shutdown nor refueling posed unusual problems for the GT-MHR designs. However, in the DC design, the hot duct must be plugged when the turbomachine is removed for service to allow continued use of the shutdown cooling system. An interesting characteristic of the GT-MHR designs is the ability of the turbomachines to continue rotation without the need for external power (self-turbining) following a reactor shutdown. In the DC, this represents an additional possibility for continued helium circulation through the core when power is lost. A source of cooling water for the precoolers and intercoolers would still be needed, however, to retain the temperature differentials required for continued turbomachinery operation.

SAFETY AND INVESTMENT RISK ASSESSMENTS

The safety and investment risk characteristics of the DC, IDC and enhanced SC were assessed and compared with the reference 450 MWt SC plant. Key transients and failure modes were examined and, of these, the following three categories were judged to be of particular importance:

- Pressurized and depressurized conduction cooldowns, a major focus of prior SC safety assessments.
- Turbomachinery failures, which are a safety issue in the case of the DC, but largely an investment risk issue in the SC and IDC designs.

- Heat exchanger failures, which have been particularly important in SC safety assessments.

Conduction Cooldown Events

For all of the designs, there are three independent systems for removing the residual heat. These are the normal power conversion systems (Heat Transport System and/or Power Conversion System), the Shutdown Cooling System and the Reactor Cavity Cooling System. Differences arise from the higher thermal power level of the 475 MWt SC, the higher normal operating temperatures of both GT-MHR designs and the lower equilibrium pressure of the DC primary system when the capability for active cooling is lost.

For the pressurized conduction cooldown (PCC) event, peak fuel temperatures remain well below the 1600°C guideline in all cases. Metallic components are also within temperature limits, with one exception. In the IDC, the maximum core barrel temperature of 789°C (1453°F) exceeds the investment risk guideline of 760°C (1400°F). In the DC, the maximum core barrel temperature is lower [704°C (1300°F)], since the equilibrium pressure in the DC reactor is reduced when turbine rotation ceases (the peak fuel temperature is correspondingly higher, but well below limits, as noted above). In conclusion, a code case would be required for the IDC design, but this is considered feasible. The PCC is not limiting for the other designs.

For the depressurized conduction cooldown (DCC) event, the pressure is the same for all the concepts (atmospheric) and temperatures are driven by differences in reactor power (475 MWt SC) and initial operating temperatures (DC and IDC). The resulting maximum fuel and metallic component temperatures are acceptable in all of the concepts, although the control rods are at or near the limit in every case. It is interesting to note that the DC and IDC vessel temperatures are below their normal operating temperatures in the DCC event.

Radionuclide releases associated with DCC events are influenced by the initial operating conditions within the core and the mechanisms for transporting radionuclides from the vessel to the vented low pressure containment (VLPC). The higher initial core operating temperatures in the GT-MHR designs lead to radionuclide releases from the core early in the accident that are higher than the SC. When considering the mechanisms that control the rate of release from the vessel to the VLPC, the equilibrium pressure within the vessel following loss of the turbine or heat transport system is particularly important. Because the equilibrium pressure in the DC vessel is lower, analyses indicate that the SC and IDC have longer and higher initial depressurization rates to the VLPC than the DC. This results in higher releases. Since the temperature effects dominate, both GT-MHR plants will have releases higher than the SC plants. However, the lower equilibrium pressure in the DC will result in lower releases than projected for the IDC.

Turbomachinery Failures (DC)

Turbomachinery failure was assessed for the DC design, due to the location of the turbomachine in the primary system. A review of existing experience data for conventional turbomachines indicates that major failures have been relatively rare, on the order of 10^{-3} per year. The probability of failure should be even lower in the GT-MHR, since the operating

environment is relatively benign and failure modes related to fuel combustion systems and turbine hot section overheating, which represent the majority of failures for conventional turbomachines, are not a factor in the GT-MHR.

The containment of missiles generated by turbomachinery related failures is considered manageable. A containment design similar to that proposed for the DC was used in the Fort St. Vrain circulator. However, deblading events may lead to relatively large, rapid pressure changes in the reactor.

To explore this issue, a preliminary computer model was developed to assess the gas dynamics of the DC during accident conditions. A transient was analyzed in which a turbine deblading event is assumed to result in the loss of all blades, except for the first stage stator guide vanes, in a time period of 0.05 s. The results of the analysis indicate a maximum pressure differential across the reactor of 304 kPa (3 atm) at 2 s from the initiation of the transient. Structural analysis of the reactor core supports from previous studies indicates that core support integrity is maintained up to a core pressure differential of 1 MPa (10 atm). Thus, failure would not be expected.

In conclusion, while much additional work is required, the probability and/or consequences of turbomachinery failures do not appear to impact the inherent and passive safety features of the MHR design.

Heat Exchanger Failures

Steam/water ingress resulting from steam generator tubing failures during power production leads to the highest consequence events in the SC concepts. In the DC design, the steam generator is replaced by the precooler and intercooler, and the circulator cooler is replaced by the generator cooler. All of these heat exchangers, as well as the shutdown cooling heat exchanger common to both concepts, operate at a lower pressure on the water side than in the primary system. Thus, the potential for significant water ingress during power production is eliminated in the DC.

In the IDC, the intercoolers and precooler are located in the secondary circuit, thus further reducing the potential for water ingress during power production. Qualitatively, then, the risk of water ingress into the primary coolant during power operation is much less for the gas turbine designs, which are without a steam generator. However, the risk of water ingress during cold depressurized shutdowns, which relates to reactivity addition rather than fuel hydrolysis or graphite oxidation, is approximately the same for the DC and SC, but lower for the IDC.

Summary

Overall, elimination of the water ingress event during power production represents a major safety advantage for the GT-MHR concepts. It eliminates the highest consequence event associated with the SC concept. In addition, the lower equilibrium pressure in the DC following loss of helium circulation provides a safety advantage relative to the IDC in terms of radiological releases associated with DCC events. This advantage must be weighed against the uncertainties associated with turbomachinery failure events.

AVAILABILITY, RELIABILITY AND CAPACITY FACTOR

A review of operating experience with large gas-turbine generation and cogeneration systems was performed by Strategic Power Systems (SPS), utilizing their Operational Reliability Analysis Program (ORAP). The assessment included plants with power ratings in the range of 70-100 MWe and service factors (fired hours/total hours) exceeding 50%. The availability of the plants examined averaged 94% for simple cycle plants and 95.8% for cogeneration plants. When component level failure data are examined, it becomes evident that systems and components that are major contributors to unavailability are not found in the GT-MHR. For example, failures associated with fuel systems, combustors and NO_x injections would not be relevant. Less evident are failures that, while occurring in components common to the GT-MHR, have their root cause in systems not included. An example would be turbine damage caused by debris from a failed combustor. Based on the SPS assessment, a failure rate of 127.1 equivalent forced outage hours (EFOH) per year was assigned to the DC turbomachinery system (including the generator and frequency converter) and 45 EFOH per year to the IDC. This compares to 105.6 EFOH per year for the reference SC. The lower rate for the IDC reflects improved access for maintenance and the absence of the frequency conversion step.

Scheduled outages were assessed for the DC, IDC and enhanced SC designs and compared with the reference SC-MHR. In all cases, the critical path was found to be set by reactor refueling, along with inspection and maintenance activities in common with or comparable to the reference 450 MWt SC-MHR. Thus, it was concluded that the scheduled outage allocation of 391.4 hours should be applied to all of the concepts.

The forced outage assessment revealed some significant differences among the various concepts. While the projected forced outage rate for the DC Turbomachinery System is higher than that of either the IDC or SC (see above), that increase is more than offset by elimination of major components in the Heat Transport System [Main Circulator and Steam Generator (SC) or IHX (IDC)]. The vessel system forced outage rate is also somewhat higher for the DC, due to the greater number of penetrations. Overall, the forced outage rates of the DC and IDC GT-MHR designs were projected to be approximately equal (6.4% and 6.2%, respectively), and both were projected to be lower than the SC-MHR concepts (7.8%).

In conclusion, both the DC and IDC GT-MHR concepts are projected to have increased reliability and the potential for higher availability and capacity factors than the SC-MHR. The design capacity factor for the SC-MHR has been previously set at 84%, with a refueling interval of 18 months. The projected capacity factors of the GT-MHR concepts are as high as 89% in both cases. At that level, the fuel loading of the core is limiting, if the 18 month refueling interval is retained.

MAINTAINABILITY

Maintainability is influenced by the population of equipment that must be maintained, the accessibility to and ease of maintaining the equipment, the required frequency of maintenance activities and the environmental conditions that are encountered during maintenance.

Radiological doses to workers are a major consideration in the maintenance of nuclear facilities, and a projection of dose factors was made for the DC, IDC and enhanced 475 MWt SC, relative to the reference 450 MWt SC. The relative dose factors were estimated by scaling SC plateout estimates by the power ratio, a release factor, and a plateout distribution factor. Overall, the DC and IDC dose factors were estimated to be 10 times that of the 450 MWt SC, while the enhanced SC shows a 6% increase. Dose factors related to tasks performed in the turbine (DC) or IHX (IDC) and reactor cavities are estimated to be a factor of 40 and 10, respectively, higher than the doses in the 450 MWt SC. The results of the assessment imply that a greater degree of remote handling and maintenance will be required in the DC or IDC, relative to the SC, for equivalent occupational dose requirements.

With respect to specific maintenance requirements, activities supporting the reactor system, shutdown cooling system, and primary system vessels are essentially the same in all of the concepts. In the IDC, the steam generator is replaced by the IHX. The IHX is expected to have reduced maintenance requirements, since steam/water corrosion issues are eliminated and the pressure difference across the IHX is much lower than that in the steam generator. On the other hand, access to the IHX tubesheets for inspection and maintenance will be more difficult, due to their location. The IDC power conversion system has fewer components than the SC power conversion system, and maintenance requirements will be correspondingly reduced.

In the DC, the main circulator and steam generator/IHX are eliminated; however, the turbomachine and generator, recuperator, precoolers and intercoolers are moved into the primary helium loop. Provisions are made in the design to remove and replace the turbomachine, generator and recuperator modules from the top of the power conversion vessel, using remote handling techniques. A spare turbomachine has been included in the cost estimates to assure high availability. The precoolers and intercoolers are located in the lower part of the power conversion vessel and access is provided to individual tube circuits for inspection and/or maintenance. While removal and replacement of these heat exchangers would pose similar difficulty to the steam generator (SC) or IHX (IDC), the operating conditions of these heat exchangers are relatively mild and the intermediate cooling circuit allows tight chemistry control.

The turbomachines for both the DC and IDC should have reduced maintenance requirements, compared to the reference steam cycle. In the case of the DC, magnetic bearings have been incorporated into the design, thus, both eliminating bearing wear and incorporating an inherent diagnostic monitoring capability. Based on conventional gas-turbine experience, however, a 7.5 year interval has been selected for removal of the turbomachine for inspection and maintenance.

In summary, since the reactor system is essentially the same for all configurations, plant maintainability tradeoffs are focused on turbomachine, ducting, piping and related heat exchanger maintenance considerations. The two gas turbine concepts appear to be superior to the SC because of the elimination of a substantial number of steam and water systems that are known to require significant maintenance efforts in steam and water plants. The IDC does have the advantage of having the turbomachine and supporting heat exchangers outside of the primary coolant circuit where they are more easily accessible. However, it still has a primary coolant circulator and a large intermediate heat exchanger in the primary circuit.

In the final analysis, the direct and the IDC concepts are comparable. However, a case can probably be made that DC simplicity outweighs IDC accessibility, particularly if special accessibility and remote handling provisions are incorporated in the design of the DC.

PROSPECTS FOR FUTURE TECHNOLOGICAL GROWTH

The potential for technological growth was addressed for both electricity generation and process energy needs. All of the concepts were found to have the potential for increased power output. By increasing the number of fueled columns in the reactor from 84 to 102, thermal power outputs of up to 600 MWt appear feasible. In the case of the SC and DC, there are no obvious feasibility issues at that reactor power level. In the case of the IDC, the upper feasibility limit for circulator power appears to be in the range of 18 MWe. This would limit the maximum IDC power output to levels below 600 MWt.

The efficiency of the SC is near its practical limit at 38%, based on the fundamental characteristics of the non-reheat Rankine cycle, combined with materials limitations. Both the DC and IDC have potential for increased efficiency. In the case of the IDC, this potential will be limited by materials available for the IHX. Temperatures in the range of 950°C-1000°C have already been demonstrated for such heat exchangers. Even higher temperatures are used in conventional gas turbines. Overall, the DC concept has the highest efficiency potential, because it uses the direct Brayton Cycle.

The high temperature capabilities of the MHR offer a significant potential for cogeneration and non-electric applications. A majority of process energy inputs are provided in the form of steam. Thus, cogeneration of electricity and steam represents a major opportunity for displacing fossil fuels in the industrial sector. Where high quality steam is required, the SC would be the optimum choice. For low quality steam, including desalination applications, any of the concepts could be employed in a cogeneration mode.

High temperature process heat applications would require the use of an IHX, thus the IDC is the only practical configuration. For applications which have been widely discussed, such as coal gasification, methanol production and hydrogen production, it would be required that the nuclear heat source be uprated to a temperature level of at least 950°C. Demonstration of that capability is an objective of the HTTR reactor, which is currently under construction in Japan.

ECONOMIC ASSESSMENTS

Initial economic projections were developed for the 450 MWt DC and IDC GT-MHR designs and comparisons were made with the reference 450 MWt SC-MHR, as well as competing fossil and nuclear alternatives.

The GT-MHR cost estimates were integrated by GCRA, with inputs from participating design organizations. The estimates were developed, in terms of January 1992 dollars, in accordance with the Advanced Reactor Cost Estimate Guidelines (Ref. 7) developed by Oak Ridge National Laboratory for the U.S. Department of Energy (DOE) and are based on the EPRI reference site near Kenosha, WI. The MHR cost estimates are expected to be comparable with capital cost estimates provided in a recent USCEA report (Ref. 8). However,

inconsistencies related to the application of contingency, indirect costs and site labor rates, productivities and site material costs may remain. These inconsistencies, where known, are expected to yield more conservative GT-MHR cost estimates relative to the evaluated USCEA alternatives.

Total and unit capital costs for the two GT-MHR concepts and the reference SC-MHR are provided in Table 2. Each plant consists of four 450 MWt reactor modules with common facilities. The IDC GT-MHR was evaluated to have the highest capital cost (\$1,981 million) and unit capital cost (\$2,457/kWe) resulting primarily from the inclusion of the intermediate heat exchanger and secondary helium loop. The SC-MHR capital cost was \$350 million lower than the IDC GT-MHR but the lower plant electrical output pushed unit capital costs to \$2,349/kWe. The DC GT-MHR had slightly higher capital costs at \$1,659 million than the SC-MHR but the higher electrical output dramatically reduced unit capital cost to \$1,910/kWe. As shown in Table 2, the evaluated capital contributions to busbar costs are 30.2 mills/kW-hr for the SC-MHR, 31.6 mills/kW-hr for the IDC GT-MHR, and 24.6 mills/kW-hr for the DC GT-MHR.

Table 2 also presents the 30 year levelized busbar generation costs for the three MHR concepts. The breakdown of the busbar costs is presented graphically in Figure 8. The SC-MHR plant has the highest evaluated busbar costs at 50.3 mills/kW-hr. A larger O&M staff associated with the steam/feedwater and water treatment systems increased O&M costs relative to the GT-MHR concepts, and plant efficiency affects both the O&M and fuel cost components of busbar cost. The IDC GT-MHR has a higher capital cost component than the SC-MHR, but benefits from reduced staffing assumptions and higher net thermal efficiency reduce the busbar cost estimate to 48.4 mills/kW-hr, a 4% improvement over the steam cycle. The DC GT-MHR combines the lowest capital cost and O&M staff requirements with the highest plant electrical output leading to a busbar cost estimate of 39.7 mills/kW-hr. The DC GT-MHR is 21% lower than the SC-MHR and 18% lower than the IDC GT-MHR busbar cost estimates.

Table 2 and Figure 8 also provide a comparison of the three MHR concepts with the pulverized coal (PC) and natural gas combined cycle combustion turbine (CCCT) options prescribed in the Reference 7 guidelines and described in the USCEA study (Ref. 8). In addition, an Advanced Light Water Reactor (ALWR) and an integrated coal gasification combined cycle (IGCC) plant were added from the USCEA Study. All cases presented are for the guideline fuel cost scenarios as presented in Table 2. The ALWR and IGCC cases were also adjusted for AFUDC, to be consistent with the approach identified in Reference 7. For the 2016 startup, all options are in the competitive range, given the amount of uncertainty in estimating operating costs between the years 2016 and 2045.

The DC GT-MHR Target Plant (an equilibrium commercial plant that reflects the benefits of learning) provides an advantage of nearly 18% over the three fossil options presented. The IDC GT-MHR Target Plant is competitive with the fossil options, and the SC-MHR is marginally higher in evaluated busbar generation costs. Clearly the DC GT-MHR provides a significant economic incentive and offers the opportunity to pursue higher temperature applications of nuclear power. The ALWR plant cost estimate is presented as a point of discussion only, as updated estimates for the ALWR, consistent with the Reference 7 Advanced Reactor Cost Estimating Guidelines, were not available for a direct comparison. Based on GCRA's understanding of the labor rates, productivities, and plant contingencies applied to the

SUMMARY GENERATION COST COMPARISON ('92\$) 2016 STARTUP, PLANTS UNDER 900 MWe							
COST COMPONENTS	MHR TARGET PLANTS			ALWR	COAL		GAS
	STEAM	GT	GT	USCEA	PC	IGCC	CCCT
	CYCLE	IDC	DC	BASED	REF	REF	REF
o THERMAL RATING (MWt)	4x450	4x450	4x450	1828	1705	2x655	2x550
o NET RATING (MWe)	693	806	869	600	600	500	500
o NET EFFICIENCY (%)	38.5	44.8	48.3	32.8	35.2	38.1	45.4
o NET HEAT RATE (BTU/kWh)	8,868	7,620	7,070	10,400	9,700	8,950	7,514
o CAPACITY FACTOR (%)	84	84	84	80	80	84	84
o # OF TURBINES	4	4	4	1	1	2	2
o SCHEDULE (OVERALL)	60	63	63	60	42	42	24
o SCHEDULE (CONSTRUCTION)	32	36	36	42	30	30	18
o TOTAL CAPITAL (M\$)	1,627	1,981	1,659	1,140	836	862	282
o UNIT CAPITAL (\$/kWe)	2,349	2,457	1,910	1,900	1,394	1,723	565
o ANNUAL O&M (\$/kWe)	58.6	45.9	38.5	64.2	63.2	55.8	12.9
o FUEL COST (\$/MMBTU)	1.26	1.27	1.28	0.77	1.45	1.45	2.33
o REAL ESCALATION (%/YR)	0.0	0.0	0.0	0.0	1.0	1.0	2.2
BUSBAR COST (mills/kWh)							
o CAPITAL	30.2	31.6	24.6	26.0	19.4	22.9	7.5
o O&M	8.0	6.2	5.2	9.2	9.0	7.6	1.8
o FUEL CYCLE	11.1	9.7	9.1	8.0	20.3	18.7	39.3
o DECOMMISSIONING	1.0	0.9	0.8	1.0	0.1	0.1	0.0
TOTAL	50.3	48.4	39.7	44.2	48.8	49.3	48.6
OTHER FACTORS (mills/kWh)							
o ENVIR. EXTER. RANGE	~0-1	~0-1	~0-1	~0-2	~2-40	~1-20	~1-8

Table 2 SUMMARY GENERATION COST COMPARISON

BUSBAR GENERATION COST BREAKDOWN ('92\$) **2016 STARTUP, TARGET PLANTS UNDER 900 MWe**

Figure 8 BUSBAR GENERATION COST COMPARISONS

original ALWR cost estimate, an updated capital cost estimate consistent with Reference 7 groundrules and assumptions would be higher than that reported here, unless performance improvements or other cost reductions have been implemented.

Also included in Table 2 is a range of environmental externality costs which have been evolving throughout the U.S. on a state-by-state basis. Numerous studies have been completed and the range of values included in Table 2 are indicative of the diversity of results. Consistent with the USCEA Study, a \$500/ton Sulphur tax has been included in the O&M costs. This adds 1 mill/kW-hr to the reported busbar costs for the PC plant and 0.2 mills/kW-hr to the IGCC plant. The intent is to capture the full environmental cost of the respective options, at least in the decision process for option selection.

DEVELOPMENT REQUIREMENTS

Both the DC and IDC would require additional development efforts, relative to the SC. Of the two gas-turbine concepts, the DC poses the more difficult development challenge. Common to both GT-MHR concepts, as well as the SC, is the development of fuel meeting the stringent MHR safety and licensing requirements. Since the SC fuel requirements envelope those of the GT-MHR, no significant changes would be required in that most important development effort.

Most of the significant issues in the DC relate to systems integration, rather than the feasibility or performance of individual components. DC components are, for the most part, enveloped by the existing state of the art. Notable exceptions, however, are the turbomachinery and generator, which will require a significant development effort. The particular challenge of the DC, however, is the integration of the individual components into a compact, highly reliable, operable and maintainable system that consistently attains the high performance that has been projected for it.

A major requirement of the DC development program is the integrated testing of a prototypical power conversion system before operation in a nuclear environment. A non-nuclear heat source will be required, and the major goal will be to verify the performance and integrity of the overall system within the power conversion vessel. Recognizing that the helium turbo-machine is a key element of the test, the Evaluation Team has recommended that the facility have the following characteristics: (1) full speed operation, (2) design value of turbine inlet temperature [i.e., 950°C (1562°F)], and (3) system design for reduced helium pressure (say 20%) for cost effectiveness.

A full spectrum of plant steady state, transient, and upset conditions will be simulated. The test facility will also be used to demonstrate all of the handling equipment necessary to facilitate component installation and removal.

The IDC concept incorporates systems and components from the reactor to the power conversion system in an arrangement similar to that of the SC. The steam generator is replaced with a helium-to-helium IHX. The IDC turbomachinery is a conventional, horizontal layout of the compressor, turbine, and generator. The generator is a conventional, off-the-shelf unit. The turbomachinery relies on the extensive European operational experience with closed-cycle air

turbines using fossil-fueled heaters. Oberhausen, a fossil-fired prototype for the German/Swiss high-temperature helium turbine (HHT) gas-cooled reactor provides a base of experience for turbomachinery, using helium as the working fluid. This is supplemented with the design experience gained in the HHT project. Power plants using twin-shaft turbomachinery are also supported by operational data from aero-derivative fossil-fired plants currently in service.

Prototype testing would be required for a full scale main circulator and at least part of the IHX tube bundle. The main circulator proof testing could be performed in a vendor facility, which may need expansion to provide the appropriate capacity and power. The IHX would require a special test set-up.

It is not anticipated that the power conversion loop, in general, and the turbomachinery, in particular, would require integrated proof testing, such as that proposed for the DC. The turbomachinery vendor would include any required confirmations within their normal scope.

Development Cost Projections

Development cost projections for the SC, IDC and DC are provided in Table 3. The IDC technology development cost estimate is nearly 8% higher than the SC at \$321 million. Increases are found in the circulator (\$11 million), turbomachine (\$5 million), and international technology transfer (\$6 million) development costs. The IHX development cost replaces the steam generator development needs of the SC. The \$91 million increase in technology program costs for the DC relative to the IDC is dominated by the \$115 million cost for the turbomachine test facility. This is offset by elimination of the circulator testing (\$34 million). Additional costs are also estimated for metals technology, \$5 million, and reactor system DV&S, \$11 million, due to complexities related to placing the turbomachine in the primary system. The DC technology development cost is estimated to be \$422 million, which is 42% higher than the SC and 31% higher than the IDC. Design development costs are estimated at \$436, \$514, and \$549 million for the SC, IDC, and DC, respectively. Turbomachinery design costs account for a large fraction of the design development cost increase relative to the SC, with the IDC turbomachine design costs estimated at \$41 million and the DC estimated at \$90 million. Plant level design and analysis activities, reactor design, and licensing activities are also envisioned to be higher for the GT-MHR concepts.

EVALUATION

The DC and IDC gas turbine options were evaluated against the reference SC plant, using a variant of the Kepner-Tregoe methodology. An enhanced 475 MWt SC plant was also evaluated; however, the level of technical detail available was less than that for the other three concepts, and cost estimates were not developed for the 475 MWt SC. The essential criteria were deemed to be met by each of the four options. For some of the more subjective criteria, a definitive conclusion could not be reached, and so the concepts were given the "benefit of the doubt".

In the subsequent quantitative evaluation, a clear advantage was indicated for the two gas-turbine options, relative to the two steam cycle options. The differentiation of the two categories was unambiguous in that it was consistent throughout the most important categories addressing

TABLE 3
MHR DEVELOPMENT COST SUMMARY
(1992 M\$)

COST CATEGORIES	Steam Cycle		Gas Turbine – IDC		Gas Turbine – DC	
	50%/50%	90% Conf.	50%/50%	90% Conf.	50%/50%	90% Conf.
	Estimate	Factor	Estimate	Factor	Estimate	Factor
TECHNOLOGY DEVELOPMENT						
FUELS DEVELOPMENT; FP BEHAVIOR	100	145%	100	145%	100	145%
GRAPHITE DEVELOPMENT	22	109%	22	109%	22	109%
METALS DEVELOPMENT	26	110%	26	111%	31	110%
FUEL PROCESS DEVELOPMENT	58	150%	58	150%	58	150%
REACTOR SYSTEM DV&S	19	150%	19	150%	30	150%
CIRCULATOR DV&S	23	130%	34	130%	0	N/A
HEAT EXCHANGER DV&S	8	130%	6	150%	5	150%
TURBOMACHINERY DV&S	0	N/A	5	150%	125	170%
SHUTDOWN COOLING SYS. DV&S	7	130%	9	130%	9	130%
REACTOR CAVITY COOLING SYS. DV&S	5	141%	5	141%	5	141%
FUEL HANDLING & STORAGE SYS. DV&S	5	175%	5	175%	5	175%
PLANT CONTROL/PROTECTION SYS. DV&S	4	140%	4	140%	4	140%
PROGRAM MANAGEMENT	9	134%	10	121%	10	129%
MISC. (SAFETY, INT'L TECH TRANSFER)	12	161%	18	146%	18	146%
SUBTOTAL TECH. DEVELOPMENT	298	136%	321	139%	422	149%
DESIGN AND LICENSING						
PLANT LEVEL DESIGN & ANALYSIS	75	144%	84	147%	88	149%
REACTOR SYSTEM	80	145%	94	150%	94	150%
VESSEL SYSTEM	19	130%	19	150%	21	150%
CIRCULATOR	12	140%	18	140%	0	140%
HEAT EXCHANGERS	25	130%	25	150%	25	150%
TURBOMACHINERY	0	N/A	41	150%	90	150%
SHUTDOWN COOLING SYSTEM	15	137%	15	143%	15	143%
REACTOR CAVITY COOLING SYSTEM	3	140%	3	140%	3	140%
FUEL HANDLING & STORAGE SYSTEM	17	174%	17	174%	17	175%
OTHER NI SYSTEMS	15	127%	16	129%	14	134%
PLANT INSTRUM., CONTROL & PROTECT.	21	129%	18	133%	17	134%
TURBINE PLANT EQUIPMENT	7	106%	5	97%	1	60%
NI BUILDINGS & STRUCTURES	20	126%	20	129%	20	126%
ECA BUILDINGS & STRUCTURES	4	100%	3	121%	1	157%
PLANT AIR, WATER & SERVICE SYS.	11	109%	9	118%	8	122%
PLANT ELECTRICAL & SECURITY SYS.	11	121%	10	121%	9	124%
DESIGN MGT. & COST DEVELOPMENT	56	130%	66	138%	65	143%
REQ'TS, PLANNING & CONTROL, QA	25	125%	30	137%	28	147%
LICENSING MANAGEMENT	20	157%	21	157%	33	157%
SUBTOTAL DESIGN & LICENSING	436	137%	514	144%	549	147%
TOTAL DEVELOPMENT COST	734	136%	835	142%	971	148%
FACTOR RELATIVE TO SC	1.00	100%	1.14	104%	1.32	108%

safety, performance and economics. After further evaluation, it was concluded that the DC, while posing the greater development challenge, offers convincing economic incentives, along with a potential for future growth as the technology evolves. As a conclusion of the evaluation, it was the unanimous recommendation of the Evaluation Team that the DC GT-MHR be identified as the initial commercial objective of the HTGR Program. The Evaluation Team recommendation was formally endorsed by the participating contractors in the form of letters from corporate officers. The recommendation was also endorsed by the Gas-Cooled Reactor Associates utility membership and approved by the U.S. Department of Energy in June 1993.

CONCLUSIONS AND RECOMMENDATIONS

As a result of the evaluation, the following conclusions were reached:

1. The DC GT-MHR should be adopted as the commercial objective of the Gas-Cooled Reactor Program.
2. A limited effort should be maintained on the IDC, with emphasis on the development of a compact, low-cost intermediate heat exchanger (IHX) technology. A low-cost, reliable IHX is seen as a key element of practical process heat applications, as well as providing a technical backup for the DC GT-MHR.

In addition to the continued development of MHR base technology (e.g., fuel), the following work was recommended in support of the DC GT-MHR design:

1. A trade study should be undertaken to identify the optimum initial power level for the GT-MHR. Initial indications are that thermal power levels of up to 600 MWt are practical, given that the prior constraints of the SC-MHR (related to the steam generator) are eliminated in the DC GT-MHR.
2. Additional options should be explored for the development and integrated testing of the DC power conversion loop. An alternate approach, which uses a conventional gas-turbine and external combustor to provide pressurized heated air to an intermediate heat exchanger has been suggested as a possible heat source. Work is planned to further explore that option. In support of that effort, a better definition of design and development requirements is needed.
3. The design and development of the DC turbomachinery and generator should be continued with high priority.
4. The effort leading to a better understanding of the levels of radionuclide contamination in the primary system and the development of maintenance procedures appropriate to those levels should be continued with high priority.

As a summary conclusion, the evaluation of the GT-MHR has fully achieved its objective. The DC GT-MHR has been unanimously recommended by evaluators as the commercial objective of the Gas-Cooled Reactor Program, and that recommendation has been accepted by both participants and sponsors. The resultant redirection to the DC GT-MHR represents a major

turning point in the evolution of the gas-cooled reactor. While the development and demonstration of the GT-MHR represents a significant challenge, the inherent and passive safety characteristics, high efficiency and improved economics of the GT-MHR offer an attractive and competitive nuclear generation option for the early 21st Century.

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SECTION 1

INTRODUCTION

This report documents the final phase of a two-part evaluation of the Gas Turbine Modular Helium Reactor (GT-MHR). The evaluation concluded with a unanimous recommendation that the direct cycle (DC) variant of the GT-MHR be established as the commercial objective of the U.S. Gas-Cooled Reactor Program. This recommendation has been accepted by industrial and utility participants and the U.S. Department of Energy (DOE). It is believed by those participating in the evaluation that the successful development of the GT-MHR will lead to a significant improvement in the safety and competitive potential of nuclear energy in the early 21st century.

In this section, the background leading to the evaluation of the GT-MHR is presented, and the scope and approach of the evaluation are summarized. The organization of the report is also outlined in the final subsection.

1.1 EVOLUTION OF THE GT-MHR

Interest in the direct coupling of gas-cooled reactors with gas turbine power conversion cycles dates to the early development of nuclear energy (Ref. 1-1). For example, the Army's 3.3 MWt/330 kWe prototype Mobile Low-Power Reactor (ML-1), which used nitrogen as the coolant and a direct closed Brayton cycle for power conversion, was tested at the National Reactor Testing Station in Idaho in the early 1960's. In the same timeframe, the 3 MWt helium cooled, graphite moderated Ultra High Temperature Reactor Experiment (UHTREX) was operated at Los Alamos with core outlet temperatures of 1320°C.

The successful demonstrations at the Peach Bottom I and AVR plants, along with the early indications of success at the Fort St. Vrain and THTR plants, led to commercial sales of large steam cycle High Temperature Gas-Cooled Reactors (HTGR-SC) in the U.S. (later canceled or withdrawn), as well as interest in advanced concepts. The interest in advanced HTGR concepts was particularly focused in Europe, where the German/Swiss High-Temperature Helium Turbine (HHT) Project made significant advancements in gas turbine technology. The HHT Project included a gas turbine test loop at Jülich, and a fossil-fired closed cycle gas turbine of prototypical size at Oberhausen (Ref.1-2). Large gas turbine HTGRs (HTGR-GT) were also addressed in French studies during 1970-77 (Ref. 1-3).

During the HTGR commercialization period in the early-1970s, U.S. involvement in HTGR-GT development was largely limited to advanced design studies. With the mid-1970's, however, came an industry-wide retrenchment of nuclear deployment, that included cancellation or withdrawal of all ongoing HTGR steam cycle (HTGR-SC) commercialization efforts. In that context, interest in the HTGR-GT increased in the late 1970s, as a new focus for the U.S. development program. This led to comparison of a 2000 MWt/800 MWe HTGR-GT with the reference 2240 MWt/858 MWe HTGR-SC during 1979-81. On the basis of that comparison, the HTGR-SC was selected for continued development as the lead HTGR design for initial

deployment. On balance, the economic incentives for the HTGR-GT were found insufficient to justify the technical development and licensing risks that were foreseen (Ref. 1-4).

Consideration of gas-turbine applications in the U.S. was largely dormant until 1985, when the Modular Helium Reactor (MHR) became the principal focus of the U.S. program. A paper presented by McDonald at the 1986 IECEC Conference (Ref. 1-5) was the first to suggest that the inherent and passive safety characteristics of the MHR, in combination with recent advancements in gas-turbine, magnetic bearing and heat exchanger technologies, would support a practical GT-MHR design. A thesis by Staudt, under the direction of Professor Lidsky at the Massachusetts Institute of Technology (MIT), subsequently outlined a pebble-bed GT-MHR design, based upon the German HTR Modul concept (Ref. 1-6). The MIT work confirmed the importance of heat exchanger and turbine technology improvements, and was particularly effective in communicating the potential for mid-40% plant efficiencies at a relatively modest core outlet temperature of 850°C. Gas-Cooled Reactor Associates (GCRA) reviewed the GA and MIT results and recommended a more detailed evaluation to DOE and utility sponsors (Ref. 1-7).

The emergence of the GT-MHR began in earnest when the Empire State Electric Energy Research Corporation (ESEERCO) agreed to serve as the majority sponsor of an initial (Phase I) evaluation. A highlight of the Phase I evaluation was an International Workshop on the GT-MHR, which was attended by 94 invited experts, including 23 representatives from Europe, Japan and China. The proceedings of that workshop (Ref. 1-8) remains the most comprehensive reference on GT-MHR enabling technologies. The results of the Phase I evaluation (Ref. 1-9) identified compelling incentives for the GT-MHR, and led to the Phase II effort that is documented in this report. A key recommendation of the ESEERCO study was that a comparative evaluation be made of the direct and indirect cycle versions of the GT-MHR.

1.2 OBJECTIVES AND APPROACH

The top level objective of the GT-MHR evaluation was to identify the appropriate role of and priority for the GT-MHR in the overall context of gas-cooled reactor development in the U.S. Supporting objectives were as follows:

- Advance the designs of the direct cycle (DC) and indirect cycle (IDC) GT-MHR concepts to a level that supports comparative evaluations.
- Conduct assessments of the DC and IDC concepts, as required, to project their operability, maintainability, availability/reliability, safety/investment protection and economic characteristics.
- Identify the incremental development requirements for the GT-MHR concepts and outline proposed approaches for their development.
- Conduct a structured evaluation of the GT-MHR concepts in comparison with the reference SC-MHR, as well as an enhanced steam cycle concept that is representative of the maximum growth potential for that option.

The GT-MHR evaluation was conducted over about a one-year period, beginning in mid-1992 and concluding with a recommendation to DOE and utility sponsors in April 1993. Given the ambitious schedule and limited resources available for the evaluation, an "Evaluation Team" was assembled, that included specialists from each of the major participants in the Gas-Cooled Reactor Program, including ABB/Combustion Engineering, Bechtel, General Atomics, Massachusetts Institute of Technology, Oak Ridge National Laboratory and Stone & Webster Engineering Corporation. Overall coordination was provided by Gas Cooled Reactor Associates (GCRA). Meetings of the evaluation team were held on a frequent basis to exchange information on the evolving designs, to reach decisions on key design selections and to coordinate ongoing efforts. The effort concluded in late-April 1993 with a formal evaluation of the candidate concepts against a set of criteria that were earlier established. A consensus was reached by the Evaluation Team as a result of that meeting, and a recommendation to establish the DC GT-MHR as the commercial objective of the Gas-Cooled Reactor Program was made to DOE and utility sponsors. The Evaluation Team recommendation was reviewed by the GCRA Management Committee in early May and their concurrence obtained. DOE approved the recommendation in June 1993.

1.3 ORGANIZATION OF REPORT

The remainder of this report documents the further definition of the GT-MHR designs and the evaluation of those designs in comparison with the reference 450 MWt SC-MHR, as well as an enhanced 475 MWt SC-MHR. The definition of the designs begins in Section 2 with a summary of the Utility/User Requirements upon which the designs are based. In Sections 3 and 4, the overall designs of the DC and IDC variants are respectively summarized. Additional detail regarding major components and systems is provided in Section 5 for both the DC and IDC plants. Section 6 provides similar details regarding plant structures and the overall layouts of the DC and IDC plants. An enhanced SC design that was also evaluated is outlined in Section 7. The assessment of the various design options begins in Section 8 and continues through Section 14. Section 15 describes the evaluation methodology and results, and Section 16 provides conclusions and recommendations.

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SECTION 2

REQUIREMENTS FOR THE GT-MHR

The redirection of the U.S. Gas-Cooled Reactor Program to the Steam Cycle Modular Helium Reactor (SC-MHR) in August 1985 represented a consensus decision on the part of sponsors and participants to embrace a new approach to nuclear safety and economics. The essence of this approach may be summarized as follows:

- Apply the high temperature capabilities of the coated particle fuel system, in conjunction with inherent and passive design features to eliminate the potential for accidents with offsite public impact.
- Achieve competitive production costs for electricity by:
 - A major simplification of plant systems, particularly by eliminating the need for the active safety systems found in large gas-cooled reactors and light water reactors.
 - Increased plant efficiency, made possible by the high temperature capability of the gas-cooled reactor's coated particle fuel system.
 - Achieving the benefits of factory fabrication and learning associated with standardized modular construction techniques.
- Provide additional flexibility for matching system growth and reduce the economic risk to owners by reducing the size of incremental deployments.

2.1 KEY UTILITY/USER REQUIREMENTS

The philosophical approach to the MHR, outlined above, is communicated in the form of utility/user requirements, as input to the DOE Program. The most recent version of these requirements is found in Reference 2-1. The key requirements that drive the SC-MHR design are summarized as follows:

- Plant Capacity Factor
 - The design capacity factor for electrical generation averaged over the plant's lifetime shall be at least 84%.
 - The capacity factor loss due to planned outages averaged over the plant's lifetime shall not exceed 8%.
- Plant Investment Protection
 - The calculated capacity factor loss due to unplanned outages averaged over the plant's lifetime shall not exceed 8%.
 - Unplanned outages of six months or greater shall not contribute more than 10% of the capacity factor loss due to all unplanned outages.

- The mean likelihood of exceeding the design limits associated with safety related design conditions shall be less than 10^{-5} per plant-year.
- Safety and Licensing
 - The plant shall be designed in accordance with the "Top-Level Regulatory Criteria for the Standard HTGR." (Ref 2-2)
 - The plant shall meet the top level regulatory criteria without credit for sheltering or evacuating the public beyond the plant's exclusion area boundary.
 - The plant shall protect public health and safety without relying on the control room, its contents, the automated plant process control system, or auxiliary power supplies (other than batteries). In addition, no operator action shall be required to prevent fuel damage or mitigate accidents.
- Site Characteristics
 - The site characteristics enumerated in Table 2-1 are intended to envelope a majority of U.S. sites.

2.2 REQUIREMENTS FOR THE GT-MHR

In evaluating the potential of the GT-MHR, a fundamental assumption was that the key utility/user requirements pertaining to safety and investment protection would be the same as those specified for the SC-MHR. In practical terms, this meant that the inherent and passive safety characteristics of the SC-MHR would be retained as an essential requirement of the GT-MHR designs. Other requirements were reviewed for applicability; however, as a general rule, the SC-MHR requirements were retained unless found inappropriate to the GT-MHR designs.

As a result of the requirements review, adjustments to the requirements were assumed in a few areas. The assumed changes, which are summarized below, must be verified in the course of more detailed follow-on efforts.

2.2.1 Site Temperature Envelope

For the SC-MHR, the Utility/User Requirements document provides a site characteristics profile (Table 2-1) that is intended, in part, to provide a safety design basis envelope for conditions at a majority of prospective nuclear sites. The site characteristics profile includes specification of a maximum wet-bulb temperature which, for the SC-MHR, is 82°F (27.8°C). Referring to Figure 2-1, it can be seen that the SC-MHR design value envelopes the peak wet bulb temperature at a high temperature, high humidity site, such as Gulf States Utilities' River Bend Station in Francisville, LA.

In the reference SC-MHR design, the wet-bulb temperature specified in Table 2-1 was used both for safety and economic evaluations. For the latter, the important consideration is the impact on the normal heat rejection path through the condenser, circulating water system and, ultimately, the cooling tower. In the case of the steam cycle, the specified wet-bulb temperature, while high, does not lead to a large economic penalty, since the heat rejection train

Table 2-1 SITE CHARACTERISTICS

Site Conditions

SOIL CHARACTERISTICS

Shear Wave Velocity ⁽¹⁾	300—2,400 m/s (1,000—8,000 ft/s)
Allowable Static Bearing Capacity ^(1,2)	480 kPa (10,000 lb/ft ²)
Water Table	Normal Elevation: 2.4 m (8 ft) below grade.

METEOROLOGY⁽¹⁾

Precipitation (Rain)	Rainfall: 25.4 cm/hr (10 in/hr)		
Precipitation (Snow)	Snow Load: 2,400 Pa (50 lb/ft ²) per ASCE 7-88, Category III		
Wind Velocity	49.2 m/s (110 mi/hr) at 10 m (33 ft) above grade per ASCE 7-88, Category III		
Air Temperature		<u>Dry Bulb</u>	<u>Wet Bulb</u>
	Maximum:	43.3°C (110°F)	27.8°C (82°F)
	Minimum:	-42.8°C (-45°F)	N/A
Atmospheric Dispersion	X/Q: 2x10 ⁻⁵ sec/m ³ (5.66x10 ⁻⁷ sec/ft ³) (Annual average at EAB, accident evaluations per applicable NRC Regulatory Guides)		

DEMOGRAPHY

Population Density	Uniform 193 persons/km ² (500 persons/mi ²) to 48.3 km (30 mi) (NRC Regulatory Guide 4.7 (tentative))
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SITE BOUNDARIES⁽¹⁾

Exclusion Area Boundary (EAB)	425 m (1,390 ft) minimum from reactor release
Protected Area Boundary (PAB)	To be determined by site layout
Low Population Zone (LPZ)	425 m (1,390 ft) (same as EAB)
Emergency Planning Zone (EPZ)	425 m (1,390 ft) (same as EAB)

WATER SOURCE

Nonbrackish river water

ELEVATION

Nuclear Island ⁽¹⁾	0—1,830 m (0—6,000 ft) above mean sea level
Energy Conversion Area	30.5 m (100 ft) above mean sea level

Table 2-1 SITE CHARACTERISTICS (Continued)

Site Events

SEISMIC⁽¹⁾ (Maximum at Grade Ground Acceleration)

	<u>Horizontal</u>	<u>Vertical</u>
Safe Shutdown Earthquake	0.3g	0.3g
Operating Basis Earthquake	0.15g	0.15g

Non-safety-related systems, structures, and components shall be designed in accordance with ASCE 7-88, Zone 2, Category III.

TORNADO⁽¹⁾ (Not applicable to non-safety-related systems, structures, and components)

Designation	Region I (NRC Regulatory Guide 1.76)
Maximum Wind Speed	161 m/s (360 mi/hr)
Rotational Speed	130 m/s (290 mi/hr)
Translational Velocity	31 m/s (70 mi/hr)
External Pressure Drop	21 kPa @ 14 kPa/s (3.0 lb/in ² @ 2.0 psi/s)

FLOODING (No specific provisions. Maximum probable flood level assumed below plant grade.)

AIRCRAFT HARDENING (No specific provisions. Plant assumed outside of flight paths.)

- (1) These site characteristics are intended to envelope a majority of prospective U.S. sites. Other characteristics are reference values.
 (2) Value applicable to large mat foundations, not isolated footings.

can be sized to provide the required turbine back pressure. This is because the spread between the steam condensing temperature, 43°C (109°F), and the wet-bulb temperature provides an adequate thermal potential for efficient heat rejection. Lower wet-bulb temperature values, which would be expected at most sites, and even at high temperature, high humidity sites during much of the year, would have a positive impact on efficiency and economics if the reference plant heat rejection train were retained. Alternatively, the duty and, hence, capital cost of the heat rejection train could be reduced, consistent with more realistic site ambient temperature conditions. For the steam cycle, these options related to more realistic site temperatures have not yet been evaluated.

By contrast, gas-turbine cycles are, in general, more sensitive to ambient temperature. It was therefore concluded in Reference 2-4 that the 82°F (27.8°C) wet bulb value should be retained for safety evaluations (e.g., reactor cavity cooling system performance), but more realistic values should be selected for performance and economic assessments. Based on the

FIGURE 2-1 SITE TEMPERATURE PROFILES

analysis of Reference 2-4, a nominal value of 18.3°C (65°F) was used for the point designs documented in Sections 3 and 4. It was subsequently determined that the standard site conditions recommended by EPRI and ORNL (Ref. 2-5) for economic evaluations included a recommended wet bulb temperature of 11.1°C (52°F). This latter value was used as the basis for the economic evaluations summarized in Section 13. The sensitivity of plant efficiency to the assumed wet bulb temperature is indicated in Sections 3 and 4 for the respective GT-MHR designs.

2.2.2 Performance at Reduced Power

The utility/user requirements for the SC-MHR (Ref. 2-1) mandate that each module be capable of "... continuous operation under automatic control between 25% and 100% of nominal module feedwater flow". Amplifying requirements are provided to define rates of change in power level that must be accommodated within that range. In addition, the plant must be capable of operation in accordance with a weekly load cycle that is defined within the requirements document. The intent of such requirements is to maximize the operational flexibility afforded to utilities in meeting the needs of their customers. Operation at reduced power and load following capability become increasingly important as the size of the utility grid decreases (e.g., export to countries with small electrical grids) and as the modules become older (presumably their operating costs increase and are higher than those of newer, more modern plants).

In the case of the SC-MHR, the principal impacts arising from extended operation at reduced power relate to steam generator stability and maintaining the steam generator tubing bimetallic welds within their required operating environment. These issues are, in turn, reflected in higher feedwater pressure, steam generator orificing and increased steam generator analysis requirements. While sometimes technically challenging, these considerations do not impose a significant economic penalty on the capital or operating costs of the plant.

Operation at reduced power involves somewhat different considerations for the GT-MHR. Several approaches are available for operation of the GT-MHR at reduced power levels. These include:

1. Bypass control

Under this approach, a portion of the high-pressure helium exiting the compressor is by-passed to the low-pressure side of the cycle. This reduces flow through the reactor and turbine and, thus, reduces power output. While by-pass is a rapid and effective means of control, plant efficiency is significantly reduced while operating in this mode. Hence, long term operation at reduced power operation in the by-pass mode is not desirable.

2. Inventory Control

In the inventory control scheme, the power output of the power conversion system is adjusted by increasing or decreasing the inventory of helium in the primary system. In effect, this varies the mass flow of the helium in the cycle, while maintaining a relatively constant volumetric flow. An advantage of this approach is that plant efficiency remains

high at reduced power levels. Two design approaches have been suggested for varying the helium inventory. One approach is simply to use the already included helium storage, transfer and clean-up systems. While potentially feasible, such an approach would allow only slow inventory/power changes unless a significant system upgrade is implemented.

A second approach involves the use of separate helium storage vessels for inventory control, along with associated valves and piping. The latter involves additional equipment, enlarges the primary coolant pressure boundary and may have a significant impact on the layout and cost of the nuclear island structures.

3. Turbomachinery Speed Control

In the case of non-synchronous turbomachinery designs, which incorporate frequency conversion between the generator and the grid, the speed of the turbomachine may be reduced at part load, thus matching the mass flow through the turbine with the desired power output. Using this approach, the need for helium bypass flow at part load may be reduced or eliminated. This would result in a significant improvement in plant efficiency and a significant reduction in precooler steady state heat loads during part load operation.

4. Turbine Inlet Temperature Control

Lowering the turbine inlet temperature reduces both power output and cycle efficiency. Due to the resulting efficiency losses and relatively slow response times in a GT-MHR, turbine inlet temperature control is more likely to be used as an adjunct to one of the other control strategies, above.

In theory, bypass control could be used to meet reduced power operating requirements equivalent to those specified for the SC-MHR. Plant efficiency would be low, however, and fuel costs significantly higher. In practical terms, if extended operation at reduced power is required for the GT-MHR (e.g., a weekly load cycle), some form of inventory control, turbomachinery speed control or their equivalent, would likely be employed. This would potentially involve significant additional cost and must be weighed against the alternative of shutting down individual modules for reduced power operation, while operating the remaining modules at high power. Further, the reduced maintenance requirements and high efficiency of the GT-MHR, combined with the relatively low cost of nuclear fuel should result in very competitive generation costs. This would indicate a high likelihood of base-load operations for such plants.

For the purposes of this evaluation, the reference GT-MHR was assumed to be a base load facility. It was further assumed that provisions would be made for the addition of inventory control, turbomachinery speed control or their equivalent, as an optional plant feature, at additional cost, but without major impacts on the plant layout or major systems, structures or components. Such provisions would enhance the potential of the GT-MHR for application in countries or by utilities with small electric grids. It would also provide additional flexibility for large utilities that are willing to pay the added cost.

2.2.3 Capacity Factor and Refueling Interval

The utility/user requirements for the SC-MHR specify a plant design capacity factor of 84%. The requirements also specify a minimum refueling interval of 18 months and a fuel design capacity factor of 88%. The latter requirement is intended to provide additional flexibility for plant operators that achieve better than design capacity factors. Core design evaluations to date have indicated that a refueling interval of 18 months is achievable with capacity factors of up to 88%. Beyond that level, a tradeoff is required between capacity factor and refueling interval. For the purposes of this evaluation, it was assumed that capacity factor enhancement would take precedence over refueling interval in design tradeoffs.

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SECTION 3

PLANT LEVEL DESIGN AND INTEGRATION - DIRECT CYCLE

3.1 PLANT LEVEL DESIGN DESCRIPTION

The direct cycle gas turbine plant comprises four standard high-temperature, gas-cooled reactor modules and the systems and service facilities required to support the reactor modules. Each reactor module includes a reactor core and power conversion equipment. The reactor core and power conversion equipment are housed in separate welded steel vessels which are connected by a cross vessel. The same helium which flows through the reactor is used as the working fluid in the power conversion portion of the module. A plot plan is provided in Figure 3-1.

A simplified schematic flow diagram of one reactor module is shown in Figure 3-2. The single standard reactor module, which is the building block of the modular HTGR, contains all the power conversion equipment required to generate electricity within the primary pressure boundary. This equipment includes the turbine-compressor-generator set, plate-fin recuperator modules, precooler, intercooler and the interconnecting gas ducting.

Figures 3-3 and 3-4 depict the helium flow path inside the reactor module. Pressurized helium enters the reactor vessel and is directed to a plenum located above the reactor core. After flowing downwards through the cooling channels, the heated helium is collected in an outlet plenum and then directed to the power conversion vessel through the hot duct within the cross vessel. The helium enters the turbine at 850°C (1562°F) and 7.03 MPa (1019 psia). Upon expanding through the turbine, helium at 510°C (950°F) and 2.63 MPa (382 psia) flows through the hot side of the six parallel recuperator modules, rejecting heat to the helium returning to the core on the cold side. Cooled to 132°C (269°F), the helium then flows downwards to and through the precooler where it is further cooled to approximately 33°C (91°F).

The cooling of the helium from 510°C (950°F) at the turbine outlet to 33°C (91°F) at the precooler outlet minimizes the work required to compress the helium from 2.59 MPa (375 psia) at the compressor inlet to 7.07 MPa (1025 psia) at the core inlet. However, 79% of the thermal energy which is removed from the helium in this cooling process is recovered in the recuperator prior to precooling and, hence, not lost from the power cycle.

The cold helium is compressed in two steps in the low and the high-pressure compressor. In the low-pressure compressor, the helium is compressed from 2.59 MPa (375 psia) to 4.34 MPa (630 psia). The helium then flows through an intercooler where it is cooled to 33°C (91°F) to further optimize the compressor work in the high-pressure compressor. There, the helium is compressed from 4.29 MPa (622 psia) to 7.21 MPa (1045 psia) and heated by compression work to 112°C (233°F). Upon leaving the high-pressure compressor, the helium is channeled through the cold side of the recuperator modules. To achieve 95% recuperator heat-transfer effectiveness, the cold side helium flows in a U-shaped flow-pattern through this component. Consequently, cold side inlet and outlet manifolds are both located at the bottom

Figure 3-1 PLAN VIEW OF FOUR-MODULE DIRECT CYCLE MHTGR-GT POWER STATION

Figure 3-2 SIMPLIFIED SCHEMATIC FLOW DIAGRAM - DIRECT CYCLE

Figure 3-3 HELIUM FLOW PATH IN REACTOR MODULE

Figure 3-4 HELIUM FLOW PATH IN POWER CONVERSION VESSEL

of the recuperator modules. The helium at 112°C (233°F) at the cold side inlet is heated to 490°C (914°F) at the recuperator outlet. Here, the helium is collected in a ring-shaped outlet plenum from which it is routed through the annulus between the hot duct and the cross vessel back to the reactor vessel. The helium then flows upwards in riser channels which are attached to the outside of the core barrel to the plenum located at the top of the reactor core.

The plant includes the following key systems:

- Reactor System
- Vessel System
- Power Conversion System
- Shutdown Cooling System
- Shutdown Cooling Water System
- Reactor Cavity Cooling System
- Fuel Handling & Storage System
- Helium Services System
- Radwaste and Decontamination System
- Reactor Protection System
- Investment Protection System
- Plant Control, Data and Instrumentation System
- Power Conversion Cooling Water System
- Circulating Water System

The steam cycle Heat Transport System (notably including the steam generator and main circulator) and Turbine-Generator System are replaced by the Power Conversion System in the direct cycle gas turbine design. This system includes the turbomachine, the recuperator, precooler and intercooler and interconnecting ducts. While most of the other listed systems are shared by the steam cycle plant, they are somewhat different in the direct cycle gas turbine. The differences are mainly in size and complexity. A typical example is the Radwaste and Decontamination System. Whereas this system has to handle decontamination of the main circulator in the case of the steam cycle, it has to accommodate decontamination of the relatively large and heavy turbomachine in the case of the direct cycle gas turbine. Another example is the Circulating Water System, which can be smaller in the case of the direct cycle gas turbine, since waste heat is reduced to about half that of its steam cycle counterpart.

Several key energy conversion area (ECA) systems, structures and components found in the steam cycle plant are not needed in the direct cycle gas turbine plant. Those systems include:

- Steam and Water Dump System
- Main and Auxiliary Steam System
- Feedwater and Condensate System
- Turbine Complex and associated auxiliary systems

3.2 SELECTION OF POINT DESIGN PARAMETERS

Table 3-1 provides the point design parameters for the direct cycle. This set of parameters provides for optimum plant economics, considering the limits imposed by the current and

Table 3-1
POINT DESIGN PARAMETERS - DIRECT CYCLE

REACTOR

Core power, MWt	450
Reactor vessel parasitic losses, MWt	2.7
Core inlet/outlet temperatures, °C/°C (°F/°F)	490/850 (914/1562)
Core upper plenum inlet pressure, MPa (psia)	7.07 (1025)
Helium mass flow rate, kg/s (1000 lb/h)	239 (1897)
Active core pressure drop, MPa (psi)	0.036 (5.26)

VESSELS

Material	9 Cr - 1 Mo-V
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TURBOMACHINERY

Turbine mass flow rate, kg/s (1000 lb/h)	239 (1897)
Turbine inlet/outlet temperatures, °C/°C (°F/°F)	850/510 (1562/950)
Turbine inlet/outlet pressures, MPa/MPa (psi/psi)	7.03/2.63 (1019/382)
Turbine overall expansion ratio	2.67
Turbine polytropic efficiency, %	92
Turbine control bypass flow fraction, %	0.5
Compressor inlet/outlet temperatures, °C/°C (°F/°F)	33/112 (91/233)
Compressor inlet/outlet pressures, MPa/MPa (psi/psi)	2.59/7.21 (375/1045)
Compressor overall pressure ratio	2.79
Compressor polytropic efficiency, %	90

RECUPERATOR

Mass flow rate, kg/s (1000 lb/h)	240.3 (1907)
Hot side inlet/outlet temperatures, °C/°C (°F/°F)	510/132 (950/269)
Hot side inlet/outlet pressures, MPa/MPa (psi/psi)	2.63/2.61 (382/378)
Cold side inlet/outlet temperatures, °C/°C (°F/°F)	112/490 (233/914)
Cold side inlet/outlet pressures, MPa/MPa (psi/psi)	7.19/7.12 (1042/1032)
Effectiveness, %	95
Overall design $\Delta p/p$ (incl. cold side manifold losses), %	2.0

PRECOOLER

Mass flow rate, kg/s (1000 lb/h)	240.3 (1907)
Inlet/outlet temperatures, °C/°C (°F/°F)	132/33 (269/91)
Inlet/outlet pressures, MPa/MPa (psi/psi)	2.61/2.59 (378/375)
Design $\Delta p/p$ (incl. inlet/outlet losses), %	0.75

Table 3-1 (Continued)

INTERCOOLER

Mass flow rate, kg/s (1000 lb/h)	240.3 (1907)
Inlet/outlet temperatures, °C/°C (°F/°F)	112/33 (233/91)
Inlet/outlet pressures, MPa/MPa (psi/psi)	4.33/4.31 (628/625)
Design $\Delta p/p$ (incl. inlet/outlet losses), %	0.47

POWER PLANT

Parasitic losses, MWt	2.6
Thermodynamic efficiency, %	50.3
Frequency converter efficiency, %	98.5
Generator efficiency (excl. windage loss), %	99.0
Generator windage loss, MWe	0.7
Output at the generator terminals, MWe	222
Estimated House Loads ^(a) , MWe	5.6
True pressure loss ratio ^(b) , %	5.4
Net electrical output, MWe	213
Net plant efficiency	47.3

^(a)The house load includes the following contributors [in MWe]:

Nuclear Island (NI) load center	2.2
Intermediate loop water system	1.2
Liquid N ₂ recondenser	0.3
Circulating water system	0.7
Service water	0.3
Cooling tower makeup water	0.3
Fire pump	0.3
Water chiller	0.3

^(b)The true pressure loss ratio includes the following contributors ($\Delta p/p$ in %):

Active core	0.51
Recuperator hot side	1.00
Precooler	0.75
Intercooler	0.47
Recuperator cold side	1.00
Flow ducts	1.69

expected state-of-the-art in key technologies. The optimization approach, the bases for selected parameters, and the sensitivity of key point design parameters are presented in the following paragraphs.

Point design parameters were optimized with the goal of maximum thermal efficiency. Maximum thermal efficiency, in concert with high availability, results in the lowest power cost, which in turn is the main driver for plant economics. In the course of the optimization, a number of thermal parameters with significant impact on thermal efficiency were identified. Initial values for those parameters were then varied until the highest thermal efficiency was reached. Constraints on this optimization process, however, were imposed by hardware limitations and by the assumed ambient temperature. Hardware limitations affecting the maximum thermal efficiency are:

- Recuperator effectiveness: 95%
- Reactor vessel material: Modified 9Cr-1Mo (core inlet temperature is limited to 490°C (914°F) to stay within the time independent region during normal operation)
- Turbine polytropic efficiency: 92%
- Compressor polytropic efficiency: 90%
- Heat exchanger approach temperatures: 5.6°C (10°F)

The hardware limitations represent relatively soft limits which are subject to change as key technologies continue to advance. Industry experience and projections were used when available to determine acceptable design values. In the absence of available industry experience, engineering judgement was used to determine the design parameter.

An ambient wet bulb temperature of 18.3°C (65°F) was used for the point design calculations. This temperature represents a mean yearly averaged value of ambient temperatures at U.S. nuclear power plant sites. The thermal parameters which provided some degree of freedom for optimization of thermal efficiency within the envelope of hardware limits were:

- Primary system pressure
- Core outlet temperature
- Pressure ratio

Primary System Pressure

The selection of the primary system pressure has been based on a trade study involving the wall thickness of the reactor and the power conversion vessels, the primary coolant system pressure drop and the size of the components located within the two vessels. Higher helium pressure results in smaller component sizes for the same primary coolant pressure drop by reducing the volumetric flow rate, but increases the vessel wall thickness. Lower helium pressure results in thinner reactor and power conversion vessel walls, which reduces the cost of the vessels and makes them easier to handle and ship. Conversely, the volumetric flow rate will increase, requiring larger components (larger cross-sectional flow areas) for a fixed primary coolant loop pressure drop.

A preliminary selection of the pressure has been made for the 100% power point design. This pressure is 7.07 MPa (1025 psia) at the core inlet. Further optimization of the primary system pressure will take place as the plant design progresses.

Core Outlet Temperature/Turbine Inlet Temperature

Increasing the core outlet temperature/turbine inlet temperature increases the turbine output and hence the cycle thermal efficiency; reducing it causes the opposite effect. Constraints in increasing core outlet temperature are metallic temperatures of the hot duct and the core support structure. Also, increasing the core outlet temperature increases core ΔT and increases peak fuel temperatures.

A core outlet temperature of 850°C (1562°F) was selected. This value reflects a prudent choice with respect to the current state-of-the-art in high-temperature materials and demonstrated experience in gas-cooled high-temperature reactors.

Pressure Ratio

Increasing the pressure ratio complicates the turbomachine design (the number of stages and hub-to-tip ratio have to be increased), reduces the cycle thermal efficiency and reduces the core inlet temperature. Reducing the pressure ratio simplifies the turbomachine design (optimum for helium at a pressure ratio of about 2.4), increases thermal efficiency and increases core inlet temperature. The pressure ratio was fixed at 2.79, because this value corresponds to a core inlet temperature of 490°C (914°F), the value which represents the acceptable limit on vessel temperature during normal operation.

Thermal Efficiency Sensitivity Study

A sensitivity study was performed to determine the impact of uncertainties in key design parameters on thermal efficiency. The imposed uncertainty considers the potential range of off-point-design performance for a particular design parameter. This off-point-design performance may be caused by too optimistic/conservative point design assumptions, or because components actually perform other than expected in the as-built plant. The lower 5% and upper 95% values have been selected based on engineering judgement and industry experience. In some cases, estimates and calculations from previous gas turbine studies are reflected as well. The considered design parameters are:

- Wet bulb temperature
- Heat exchanger approach temperature
- Compressor polytropic efficiency
- Turbine polytropic efficiency
- Recuperator effectiveness
- Turbine bypass flow fraction
- True pressure loss ratio
- Recuperator leakage flow fraction

These parameters, and the core outlet temperature, are the constituents in the equation for thermal efficiency. The core outlet temperature was not varied in the sensitivity study, because its value will be controlled and maintained by the plant control system. Measurement uncertainties in the core outlet temperature have not been considered.

Each of these eight parameters was independently varied around the nominal value used for the point design calculation. Table 3-2 shows the parameters, the nominal value used for the point design and the lower 5% and the higher 95% value used for the sensitivity study.

The sensitivity study revealed that thermal efficiency is most sensitive to compressor polytropic efficiency, turbine polytropic efficiency and recuperator effectiveness, in that order. These parameters directly define the quality of the Brayton cycle in the gas turbine plant, as compared to the ideal but only theoretically achievable cycle. Their impact on thermal efficiency is exponential, which magnifies their weight.

Thermal efficiency is less sensitive to the remaining five key design parameters. However, the impact of a parameter on thermal efficiency is not only a function of the sensitivity to this parameter, but also the range of uncertainty in this parameter. If one of the insensitive parameters is perturbed far enough, or if its uncertainty band is large, then its impact on thermal efficiency can become significant. An example is turbine bypass, which has been considered in the sensitivity study with an upper 95% value of 3.5%. This upper uncertainty band, which was set high as a result of low confidence in the point design value, envelopes about 700% of the point design turbine bypass value of 0.5%.

Confidence in the point design values for compressor and turbine polytropic efficiencies and the recuperator effectiveness is high. Those parameters are well studied and industry experience is available to accurately predict the performance values, before the plant is built. Consequently their associated uncertainty range is small. Therefore, despite the high sensitivity to these three parameters, the net impact on thermal efficiency is within acceptable limits.

The results of the sensitivity study for thermal efficiency are presented in Figure 3-5. The high-lighted horizontal line at 50.3% thermal efficiency represents the point design. Placed on top of the point design line and on the right side of the uncertainty bars are the point design values which correspond to the eight parameters given on the ordinate axis.

A separate evaluation of the statistical combination of all uncertainties concluded that the mean thermal efficiency will be close to 49.7%. This result shows that the thermal efficiency of the direct-cycle can be very high even when design and performance uncertainties are considered. The statistical work assumed that the uncertainty band considered for each parameter covers the 2 sigma range. Most distributions were normal while others, like the recuperator leakage flow, were characterized by asymmetrical distributions.

A similar sensitivity study was done for the net plant efficiency, and those results are shown in Figure 3-6. A mean net plant efficiency of 46.8% was computed compared to the point design estimate of 47.3%.

DC THERMAL EFFICIENCY

Figure 3-5
RESULTS OF DIRECT CYCLE SENSITIVITY STUDY (THERMAL EFFICIENCY)

Table 3-2
PARAMETER RANGE USED FOR THE SENSITIVITY STUDY

Parameter	Lower 5% ^(a)	Nominal Value	Upper 95% ^(a)
Wet bulb temperature	10°C (50°F)	18.3°C (65°F)	26.7°C (80°F)
Approach temperature	83°C (15°F)	14.4°C (26°F)	20.6°C (37°F)
Turbine bypass	0.3%	0.5%	3.5%
Compressor efficiency	89%	90%	92%
Turbine efficiency	90%	92%	93%
True pressure loss ratio	5.0%	5.4%	8.0%
Recuperator effectiveness	93%	95%	96%
Recuperator leakage, from cold side to hot side	0.0%	0.0%	1.0%

^(a) These values were used in the sensitivity study as low and high boundaries. For the purpose of the statistical evaluation, which combined all low and high values, it was reasonably assumed that these boundary values also represent 2 sigma limits.

The high efficiency of the direct cycle GT-MHR can largely be attributed to use of two enabling technologies: the high effectiveness plate-fin recuperator and the helium working fluid, which allows high turbine and compressor polytropic efficiencies.

3.3 KEY PLANT LEVEL DESIGN SELECTIONS AND BASES

Design selections were made on the following key plant level design issues:

- Reactor vessel material
- Power conversion loop packaging
- Vertical versus horizontal turbomachine
- Submerged versus external generator
- Intercooling
- Intermediate cooling water loop

The design objective, alternatives considered, the selection made and key assumptions which require further confirmation are given for each of the above listed design issues.

Reactor Vessel Material

Objective: Contain helium at temperatures up to 510°C (950°F) (up to 621°C (1150°F) during loss of forced cooling) and pressure up to 7.93 MPa (1150 psia) within the reactor vessel.

Figure 3-6 RESULTS OF SENSITIVITY STUDY (NET PLANT EFFICIENCY)

Alternatives:

- SA533/508 vessel material
- Modified 9Cr-1Mo vessel material

Discussion: The advantage of SA533/508 is that this material is well proven and documented from light-water reactor experience. The disadvantage is that a vessel wall cooling system is required for the above specified service conditions. This vessel wall cooling system would conceptually be accomplished as follows: a small fraction (~2%) of cold, high pressure helium is branched off at the compressor outlet and piped to a header located at the bottom of the reactor vessel. From there, the helium flows upward in the annulus formed by the reactor vessel wall, the cooling channels and the core barrel. At a location above the core, through a penetration in the upper plenum shroud, the heated helium is introduced back into the main flow. Initial studies of such a cooling system concluded that vessel temperatures could be maintained below the limit of 700°F for SA533/508 during normal operation, but that the complexity of such a system could be challenging. Modified 9Cr-1Mo material can operate at the service conditions without the need for a cooling system. This material, however, is yet to be qualified for high temperature service under Code Case N-47 of the ASME Boiler and Pressure Vessel Code and additional irradiation data at the specific service conditions of interest must be generated. Also, the vessel made of Modified 9Cr-1Mo would have to be slightly heavier than the SA533/508 vessel to achieve similar stress allowables.

Selection: Modified 9Cr-1Mo

Key assumptions that were made in support of the selection and which need confirmation, are:

- Irradiation data will confirm at least equivalent performance as for SA533/508.
- Reactor vessel weight will be approximately the same as with SA533/508.
- Modified 9Cr-1Mo forgings are available in the required sizes or weldments are acceptable.

Power Conversion Loop Packaging

Objective: Provide acceptable access for maintenance of power conversion components.

Alternatives:

- Multiple vessels
- Single vessel

Discussion: The concept of multiple vessels is based on the idea of physically separating the heat exchangers from the turbomachine and housing them in individual vessels. The single vessel concept integrates all power conversion equipment, including the heat exchangers and turbomachine, within one single vessel. The multiple vessel concept provides more access to components for maintenance and in-service-inspection, but a complex system of multiple interconnecting ducts is required to connect the vessels. Another disadvantage is that the vessels

and their interconnecting ducting have to be supported seismically. Finally, the size of the silo has to be increased significantly to accommodate the multiple vessels. The key advantage of the single vessel concept is that capital costs are reduced due to the highly integrated layout, and that no interconnecting ducting between vessels is required. Component access is more challenging, which makes maintenance and in-service-inspection more difficult.

Selection: Single vessel

This selection assumes that further evaluation will confirm methods for seal inspection and leak detection, and that a full scale, part power, part pressure test facility for development testing of the integrated Power Conversion System will be available.

Vertical Versus Horizontal Turbomachine

Objective: Orientation of the turbomachine has to allow for maintainable and economic layout of power conversion components.

Alternatives:

- Horizontal turbomachine
- Vertical turbomachine

Discussion: Horizontal layout of the turbomachine is the traditional approach in the power industry. There is extensive field experience with horizontal open-cycle combustion turbomachines. However, there is no experience with horizontal turbines enclosed in pressure vessels. When positioned horizontally within a single pressure vessel, removal of the power conversion train requires rails or similar internal fixtures within the vessel. Anchorage of the turbine supports to the vessel walls is challenging due to the uneven load distribution. Silo size and complexity for a horizontal layout are significantly increased to accommodate the long turbomachine, and a maintenance cavity at least as long as the turbomachine is required to provide space for removal. Alignment of the radial bearings may be difficult, and cooling of the magnetic bearing stator coils will probably be required due to the large weight and relatively high rotating speed of the turbomachine. The design of the circumferential seals appears difficult because of the uneven loading associated with a horizontal machine. By contrast, a vertical orientation allows removal and replacement of individual power conversion components, using an overhead crane. While loads on the radial bearings are reduced significantly, the thrust bearing must bear the static load of the machine at rest. The machine flow path design is such that, during normal operation, the thrust loads react in the opposite direction to gravity, thus effectively unloading the thrust bearings.

The vertical orientation of turbomachines is not a new concept. Small to very large hydro turbine-generators and small steam turbines have been successfully operated in this mode. Nuclear experience with large vertical machines includes LWR feed pumps and gas-cooled reactor circulators. There is no experience to date, however, with heavy, high speed machinery, such as the direct-cycle turbomachine.

Selection: Vertical turbomachine

The selection of a vertical turbomachine is based on the assumption that a machine of the required size, with magnetic bearings, is achievable. It is also assumed that the weight of the components and the shielded casks, used for handling during maintenance, will not result in unreasonably large crane capacity requirements.

Submerged Versus External Generator

Objective: Locate generator to meet maintenance and safety requirements.

Alternatives:

- External generator
- Submerged generator

Discussion: Locating the generator outside the power conversion vessel provides maximum access for inspection and maintenance, and allows immediate hands-on work. In the extremely unlikely event of catastrophic failure, such as generator field disintegration, the primary pressure boundary is not affected. The key disadvantage of the externally located generator is the required high-pressure rotating shaft seal. This seal is challenging with respect to design, as well as licensing. The submerged generator eliminates the need for a large rotating shaft penetration through the vessel, and primary coolant leakage is less likely. Offsetting these advantages are several issues introduced by a submerged generator. Cooling of the generator will require a water to helium heat exchanger and, possibly, direct cooling of the stator. A rotating seal will be required between the turbine and generator cavities, both to prevent radionuclide contamination of the generator cavity and to avoid water ingress into the reactor core in the event of a cooling water pressure boundary failure when the primary system is depressurized. This seal is not considered to be a major issue. In going to a submerged generator, a further disadvantage is the large number of vessel penetrations required for generator power and instrumentation lead-ins and lead-outs. Most of these penetrations are relatively small in size, are well within the current state of the art and will not contribute significantly to the risk of primary coolant leakage. On the other hand, the generator power output penetrations, which must pass large currents through the pressure boundary at a voltage level of 25 kV, potentially represent a major development challenge.

Selection: Submerged generator

The selection of a submerged generator assumes that a machine of the required size and configuration is achievable. Initial discussions with industry experts have indicated that, while a new design will be required, the design of such a generator is felt to be within the current state-of-the-art. Prior design and operating experience with submerged gas-cooled reactor circulators provides some relevant experience, albeit in much smaller sizes. A further assumption is that the generator power lead penetrations can be successfully developed. In this case also, initial discussions with a vendor specializing in pressure boundary penetrations give encouragement that such a design is feasible (See Section 5.4 for additional information). On balance, the static penetrations are preferred by designers over the rotating seal, although both

approaches are likely to be feasible. A key differential is the perception of higher reliability in service for static components.

Intercooling

Objective: Increase thermal efficiency by intercooling between compressor sections.

Alternatives:

- One intercooler
- Two intercoolers

Discussion: Intercooling between compressor sections improves the cycle thermal efficiency, because the required compressor work is effectively reduced. As the number of intercoolers increases, the thermal efficiency increases, but the benefit is less significant with more than two intercoolers. Increasing the number of intercoolers further does not result in significant thermal efficiency gains but, in the case of the highly integrated direct-cycle configuration, results in significantly increased packaging problems. Since space is at a premium in the power conversion vessel, the concept of a single intercooler was adopted.

Selection: One intercooler

Intermediate Cooling Water Loop

Objective: Reduce risk of downtime associated with secondary water chemistry and primary coolant leaks.

Alternatives:

- Secondary cooling water heat rejection to the cooling tower loop
- Secondary cooling water heat rejection to an intermediate cooling water loop

Discussion: Direct heat rejection from the secondary cooling water loop to the cooling tower cooling loop provides the simplest and cheapest (in terms of initial cost) way to remove heat from the precooler and intercooler. In this case, the secondary cooling water loop is rejecting heat directly to the environment in the cooling tower. However, very stringent water chemistry control requirements are required to assure that the heat exchangers, which are relatively more difficult to access and maintain, do not degrade in performance, or even fail over time. Also, a method of tube leak detection and isolation has to be developed to positively avoid primary coolant leaks to the secondary cooling water loop. In case of secondary cooling water loop heat rejection to an intermediate water loop, the secondary cooling water loop is effectively sealed from the environment. The probability of heat exchanger corrosion and fouling is significantly reduced. Primary coolant leaks are contained within the intermediate cooling water loop. The disadvantages of adding a separate cooling water loop are the increased capital, operations and maintenance costs associated with the additional loop. The latter may be offset all or in part by the improved chemistry control possible in the secondary cooling loop.

Selection: Intermediate cooling water loop

SECTION 4

PLANT LEVEL DESIGN AND INTEGRATION - INDIRECT CYCLE

4.1 PLANT LEVEL DESIGN DESCRIPTION

The indirect-cycle gas turbine plant is divided into two major areas, a Nuclear Island containing four high-temperature, gas-cooled reactor modules and a energy conversion area (ECA) containing four recuperated gas-turbine power conversion units, one for each reactor, and the systems and service facilities required to support the reactor modules and gas-turbine units, as shown in Figure 4-1.

Each reactor module is located within the Nuclear Island and is comprised of two side-by-side welded steel vessels located underground and connected, as shown in Figure 4-2, by a cross vessel with a concentric hot duct. One of the vessels contains the reactor and the shutdown cooling system and the other the intermediate heat exchanger (IHX) and the main circulator. The IHX is of a compact helical tube design. The main circulator is located at the top of the vessel to provide easy access for maintenance or replacement. All safety-related structures and components are located within the Nuclear Island, permitting procurement and construction of the remaining parts of the plant to conventional standards.

The gas-turbine-generator unit is located at ground level within the energy conversion area, and is connected to the IHX vessel through two large ducts, as shown in Figure 4-3. The Power Conversion System comprises a twin-shaft turbine-compressor-generator unit, a precooler, two intercoolers, a plate-fin recuperator, and the interconnecting ducting. These components are readily accessible for maintenance or replacement.

A simplified schematic flow diagram of a reactor module is shown in Figure 4-4. The reactor module employs a 450 MWt core. There are two helium cooling loops. The primary cooling loop removes the heat generated by the core and transports it to the shell side of the IHX. The primary helium flow path is as follows: the helium discharged from the main circulator flows through the annular section, formed by the OD of the hot duct and the ID of the cross vessel, to the top of the core, as shown in Figure 4-2. After flowing downwards through the core, the helium is collected in an outlet plenum and then routed through the hot duct to the inlet of the IHX. From there the helium flows downward through the shell side of the IHX tube bundle and back to the main circulator inlet. The helium temperature and pressure at the IHX bundle inlet are 850°C (1562°F) and 7.01 MPa (1016 psia) and at the outlet are 485°C (905°F) and 6.96 MPa (1010 psia).

The secondary cooling loop removes the heat from the tube side of the IHX and transports it to the gas-turbine power conversion unit. Because of the high helium pressure, the secondary cooling loop components are quite compact. The secondary helium leaves the IHX at 775°C (1427°F) and 7.25 MPa (1052 psia) and is expanded through the high and low pressure turbine stages. A twin-shaft configuration has been selected for the turbomachinery. The medium and high pressure compressor stages are driven by the high pressure turbine. The low pressure

Figure 4-1 PLAN VIEW OF FOUR-MODULE INDIRECT CYCLE MHTGR-GT POWER STATION

Figure 4-2 INDIRECT CYCLE GAS TURBINE REACTOR MODULE LAYOUT

Figure 4-3 INDIRECT CYCLE REACTOR AND TURBOMACHINERY BUILDINGS

Figure 4-4 INDIRECT CYCLE SIMPLIFIED FLOW DIAGRAM

compressor and the generator are driven by the low pressure turbine. The low pressure shaft runs at 3600 rpm.

After expansion through the turbines, the helium enters the hot side of the recuperator at 442°C (828°F) and 2.55 MPa (370 psia), as shown in Figure 4-4, where it is cooled to 109°C (228°F). The recuperator improves the cycle efficiency by recovering heat from the helium at the turbine outlet and using it to preheat the helium before it returns to the IHX. From the recuperator the helium is further cooled through a tube and shell water cooled precooler to 30°C (86°F). This further cooling is necessary to minimize the work required to compress the helium from 2.48 MPa (360 psia) to 7.53 MPa (1092 psia).

Compression occurs in three steps in the low, medium, and high-pressure compressor stages. The pressure ratios for these three stages are 1.34, 1.52, and 1.52, respectively, for a total of 3.08. Due to the lower synchronous speed (3600 rpm), a lower pressure ratio has been selected for the low-pressure compressor stage to minimize the number of blades. From the low-pressure compressor, the helium flows through the first intercooler where it is cooled to 30°C (86°F) to further optimize the compressor work. Then the helium is compressed and cooled again by a second intercooler to 30°C (86°F). From there, the helium goes through the final compression stage and is then channeled through the cold side of the recuperator back to the IHX.

The plant includes the following key systems:

- Reactor System
- Vessel System
- Heat Transport System
- Power Conversion System
- Shutdown Cooling System
- Shutdown Cooling Water System
- Reactor Cavity Cooling System
- Fuel Handling and Storage System
- Helium Services System
- Radwaste and Decontamination System
- Reactor and Investment Protection Systems
- Plant Control, Data, and Instrumentation System
- Circulating Water System

While comparable systems are also found in the steam-cycle, the systems differ in many respects. For example, the Heat Transport System for the gas turbine has an IHX instead of a steam generator. Another example is the Circulating Water System, which in the case of the gas turbine can be significantly smaller due to the reduced amount of waste heat compared to its counterpart in the steam-cycle.

Some of the steam-cycle plant key systems within the ECA are not needed or used in the indirect-cycle gas turbine power conversion system. These systems are:

- Steam and Water Dump System
- Main and Auxiliary Steam System
- Feedwater and Condensate System

4.2 SELECTION OF POINT DESIGN PARAMETERS

Table 4-1 summarizes the plant design parameters and plant performance for the indirect cycle point design. This set of parameters provides the best achievable plant economics, considering the limits imposed by the current and expected state-of-the-art in key technologies. The design parameters optimization approach, the bases for the parameters selection and the sensitivity of key parameters are presented in the following paragraphs.

The point design parameters were optimized with the goal of maximum thermal efficiency. Maximum thermal efficiency in concert with high availability results in the lowest power cost, which, in turn, is the main driver for plant economics. In the course of the optimization, a number of thermal parameters with significant impact on thermal efficiency were identified. Initial values of these parameters were then varied until the highest thermal efficiency was obtained. Constraints on this optimization process, however, were imposed by hardware limitations and by the assumed ambient temperature. Hardware limitations are:

- IHX hot/cold ends approach temperatures: 75°C/60°C (135°F/108°F)
- Recuperator effectiveness: 95%
- Reactor vessel material: Modified 9 Cr - 1 Mo [Core inlet temperature is limited to 490°C (914°F) during normal operation to stay within the time independent range.]
- Turbine polytropic efficiency: 92%
- Compressor polytropic efficiency: 90%
- Precooler/intercoolers approach temperatures: 5.6°C (10°F)

The hardware limitations represent relatively soft limits which are subject to change as key technologies continue to advance. Industry experience and projections, when available, were used to determine acceptable design values. In the absence of available industry experience, engineering judgment was used.

As noted in Section 2.2, the wet bulb temperature used for the point design calculations was 18.3°C (65°F). This temperature represents a mean yearly ambient temperature value averaged over typical U.S. nuclear power plant sites.

The thermal parameters which provided some degree of freedom for optimization of the thermal efficiency within the envelope of hardware limitations are:

- Primary and secondary cooling loops pressure.
- Core outlet temperature.
- Turbine/compressor pressure ratio.

Table 4-1
POINT DESIGN PARAMETERS FOR GT INDIRECT CYCLE

<u>DESIGN CONCEPT</u>	Helium-helium indirect-cycle, twin-shaft, synchronized generator, two intercoolers, external generator
<u>PRIMARY COOLANT SYSTEM</u>	
REACTOR	
Core power, MWt	450
Reactor vessel parasitic losses, MWt	2.7
Heat added by the main circulator, MWt	7.0
Core inlet/outlet temperatures, °C/°C (°F/°F)	490/850 (914/1562)
Core upper plenum inlet pressure, MPa (psia)	7.05 (1022)
Primary coolant mass flow rate, kg/s (1000 lb/h)	239.4 (1900)
Active core pressure drop, MPa (psi)	0.036 (5.27)
VESSELS	
Material	9Cr-1Mo-V
MAIN CIRCULATOR	
Mass flow rate, kg/s (1000 lb/h)	239.4 (1900)
Pressure rise across circulator, MPa (psi) ^(a)	0.104 (15.1)
Primary coolant temperature rise, °C (°F)	5.7 (10.2)
Electric power required at 100% flow, MWe	8.1
Motor power required at 100% flow, MWe	7.5
Adiabatic efficiency, %	80.0
Mechanical efficiency, %	99.9
Motor efficiency, %	94.0
Two transformers (2 @ 96%) efficiency, %	92.0
INTERMEDIATE HEAT EXCHANGER (IHx)	
Heat duty, MWt	454
Hot side mass flow rate, kg/s (1000 lb/h)	239.4 (1900)
Hot side inlet/outlet temperatures, °C/°C (°F/°F)	850/485 (1562/905)
Hot side inlet/outlet pressures, MPa/MPa (psia/psia)	7.01/6.96 (1016/1010)
Cold side mass flow rate, kg/s (1000 lb/h)	250 (1984)
Cold side inlet/outlet temperatures, °C/°C (°F/°F)	425/775 (797/1427)
Cold side inlet/outlet pressures, MPa/MPa (psia/psia)	7.46/7.25 (1082/1052)
Effectiveness, %	86.0
Hot end approach temperature, °C (°F)	75 (135)
Cold end approach temperature, °C (°F)	60 (108)

Table 4-1 (Continued)

SECONDARY COOLANT SYSTEM**TURBO MACHINERY**

Turbine mass flow rate, kg/s (1000 lb/h)	248.8 (1975)
Turbine inlet/outlet temperatures, °C/°C (°F/°F)	775/442 (1427/828)
Turbine inlet/outlet pressures, MPa/MPa (psi/psi)	7.24/2.56 (1050/372)
Turbine overall expansion ratio	2.82
Turbine polytropic efficiency, %	92
Turbine control bypass flow fraction, %	0.5
Compressor #1, #2, #3 inlet/outlet temperatures, °C/°C (°F/°F)	30/72 (86/162) 30/91 (86/196) 30/91 (86/196)
Compressor inlet/outlet pressures, MPa/MPa (psi/psi)	2.48/7.53 (360/1092)
Compressor #1, #2, #3 overall pressure ratio	3.08 (1.340, 1.516, 1.516)
Compressor polytropic efficiency, %	90
True pressure loss ratio, %	8.24

RECUPERATOR

Mass flow rate, kg/s (1000 lb/h)	250 (1984)
Hot side inlet/outlet temperatures, °C/°C (°F/°F)	442/109 (828/228)
Hot side inlet/outlet pressures, MPa/MPa (psi/psi)	2.55/2.51 (370/364)
Cold side inlet/outlet temperatures, °C/°C (°F/°F)	91/425 (196/797)
Cold side inlet/outlet pressures, MPa/MPa (psi/psi)	7.52/7.49 (1091/1086)
Recuperator effectiveness, %	95
Overall design $\Delta p/p$ (include manifold losses), %	0.75

PRECOOLER

Mass flow rate, kg/s (1000 lb/h)	250 (1984)
Inlet/outlet temperatures, °C/°C (°F/°F)	109/30 (228/86)
Inlet/outlet pressures, MPa/MPa (psi/psi)	2.51/2.49 (364/361.2)
Design $\Delta p/p$ (include inlet/outlet losses), %	0.75

INTERCOOLERS (2)

Mass flow rate, kg/s (1000 lb/h)	250 (1984)
Inlet/outlet temperatures, °C/°C (°F/°F) (Intercooler #1)	72/30 (162/86)
Inlet/outlet pressures, MPa/MPa (psi/psi) (Intercooler #1)	3.32/3.30 (481/478.9)
Design $\Delta p/p$ (include inlet/outlet losses), % (Intercooler #1)	0.40
Inlet/outlet temperatures, °C/°C (°F/°F) (Intercooler #2)	91/30 (196/86)
Inlet/output pressures, MPa/MPa (psi/psi) (Intercooler #2)	4.99/4.97 (723.5/721)
Design $\Delta p/p$ (include inlet/outlet losses), % (Intercooler #2)	0.33

Table 4-1 (Continued)

POWER PLANT

Parasitic losses, MWt	2.6
Thermodynamic efficiency, %	47.6
Generator efficiency (excl. windage loss), %	99.0
Output at the generator terminals, MWe	210.5
House loads, ^(b) MWe	13.6
True pressure loss ratio, ^(c) %	8.24
Net electrical output, MWe	196.9
Net plant efficiency	43.8

^(a)The primary loop pressure drop at 100% power is based on the following expected pressure drops distribution:

Main circulator outlet to core upper plenum	0.017 MPa (2.5 psi)
Core upper plenum to core lower plenum	0.036 MPa (5.3 psi)
Core lower plenum to IHX inlet	0.006 MPa (0.8 psi)
Hot side IHX	0.041 MPa (6.0 psi)
Hot side IHX exit to main circulator inlet	0.003 MPa (0.5 psi)
TOTAL	0.104 MPa (15.1 psi)

^(b)The house load includes the following contributors [in MWe]:

Nuclear Island (NI) load center	2.2
Energy Conversion Area (ECA) Load Center	1.0
Liquid N ₂ recondenser	0.3
Circulating water system	0.7
Service water	0.4
Cooling tower makeup water	0.3
Fire pump	0.3
Water chiller	0.3
Primary loop circulator	8.1
TOTAL	13.6

^(c)The secondary loop true pressure loss ratio includes the following contributors ($\Delta p/p$ in %):

IHX cold side	2.90%
Recuperator (hot + cold sides)	2.00%
Precooler	0.75%
First intercooler	0.40%
Second intercooler	0.33%
Flow ducts	1.78%

The bases for selecting an optimized value for these parameters are the following:

The secondary cooling loop pressure at the IHX has been set slightly higher than the primary cooling loop pressure during normal power production. This was done to reduce the likelihood of contaminating the secondary cooling loop in case of an IHX tube leak. (Under some conditions other than normal power production, the primary system pressure may be higher; however, this would be a small fraction of the total time.) Increasing the secondary coolant pressure also increases the thermal efficiency of the plant, because the work of compression is reduced due to lower volumetric flow rates. Reducing the pressure has the opposite effect.

Higher secondary loop pressure requires thicker walls for the secondary ducts and pressure vessels. As the secondary loop pressure is increased, the primary coolant pressure must also be increased, in order to minimize the loads (pressure difference) across the IHX tubes. A higher primary coolant pressure means that the wall thickness of the primary system pressure vessels must also be increased, which results in a higher capital cost for those vessels. The added cost of transporting the heavier vessels to the site will also be a factor.

A primary system pressure of 7.07 MPa (1025 psia) at the main circulator outlet has been selected, as it results in good thermal efficiency, reasonable component sizes and acceptable capital cost. The secondary side maximum design pressure was selected by imposing a minimum differential pressure of 248 kPa (36 psi) between the secondary and the primary loop pressures during power operations. This minimum pressure differential between the two loops occurs between the IHX hot side inlet and the cold side outlet.

The core outlet temperature (or turbine inlet temperature) has a major effect on plant efficiency. Increasing the core outlet temperature increases the turbine output and, hence, the cycle thermal efficiency; reducing it causes the opposite effect. The core outlet temperature is limited by the maximum temperature of the metallic structures (e.g., the hot duct and the IHX) and by the core support structure. Higher core outlet temperature also results in a higher temperature rise across the core, which results in higher fuel temperatures.

A core outlet temperature of 850°C (1562°F) has been selected. This value reflects a prudent choice with respect to the current state-of-the-art in high-temperature materials and fuel design.

The compressor/turbine pressure ratio affects the turbomachinery design and the plant efficiency. Increasing the pressure ratio complicates the turbomachine design (larger number of stages and hub-to-tip ratio), reduces the cycle thermal efficiency and reduces the core inlet temperature. Reducing the pressure ratio simplifies the turbomachine design (optimum for helium at a pressure ratio of about 2.4), increases thermal efficiency and increases core inlet temperature.

A turbine pressure ratio of 2.82 was selected (corresponding to a core inlet temperature of 490°C (914°F) which represents an acceptable compromise between the limit on vessel temperature during normal operation, and the turbomachinery complexity.

Thermal Efficiency Sensitivity Study

A sensitivity study was performed to determine the impact of key design parameters uncertainties on thermal efficiency. The selection of the uncertainty range for each parameter was based on the potential for off-design plant performance. This off-point-design performance may be caused by too optimistic/conservative point design assumptions, or because components actually perform differently than expected in the as-built plant. The lower 5% and upper 95% values have been selected based on engineering judgment and industry experience. In some cases, estimates and calculations from previous gas turbine studies are reflected as well.

The following plant parameters were selected for the sensitivity study:

- Cooling tower wet bulb temperature.
- Precooler/intercoolers approach temperature.
- Compressor polytropic efficiency.
- Turbine polytropic efficiency.
- Recuperator effectiveness.
- Turbine bypass flow fraction.
- Primary cooling loop pressure losses.
- Secondary cooling loop pressure losses.
- Recuperator leakage flow fraction.

These parameters, and the core outlet temperature, are the only constituents in the equation for thermal efficiency. The core outlet temperature is not a parameter in the sensitivity study, because its value is kept constant by means of IHX secondary side outlet temperature control. Measurement uncertainties in secondary side outlet temperature have not been considered.

Each of these nine parameters was independently varied around the nominal value used for the point design calculation. Table 4-2 shows the nominal values used for the point design and the lower 5% and the higher 95% value used for the sensitivity study.

The sensitivity study revealed that thermal efficiency is most sensitive to compressor polytropic efficiency, turbine polytropic efficiency, and recuperator effectiveness, in that order. These parameters directly define the quality of the Brayton cycle in the gas turbine plant, compared to the ideal but only theoretically achievable cycle. Their impact on thermal efficiency is exponential, which magnifies their weight.

The remaining six key design parameters contribute less and about equally to thermal efficiency. However, the impact of a parameter on thermal efficiency is not only a function of the sensitivity to a parameter, but also the range of uncertainty in that parameter. If one of the insensitive parameters is perturbed far enough, or if its uncertainty band is large, then its impact on thermal efficiency can become significant. An example is the turbine bypass, which has been considered in the sensitivity study with an upper 95% value of 3.5%. This upper uncertainty band which was set high as a result of low confidence in the point design value, envelopes about 700% of the point design turbine bypass value of 0.5%.

Table 4-2
PARAMETER RANGE USED FOR THE SENSITIVITY STUDY

Parameter	Low 5% ^(a)	Nominal Value	Upper 95% ^(a)
Wet bulb temperature	10°C (50°F)	18.3°C (65°F)	26.7°C (80°F)
Approach temperature	6.1°C (11°F)	11.7°C (21°F)	17.2°C (31°F)
Turbine bypass	0.3%	0.5%	3.5%
Compressor efficiency	89%	90%	92%
Turbine efficiency	90%	92%	93%
Primary pressure loss ratio	1.3%	1.5%	1.7%
Secondary pressure loss ratio	6.1%	8.2%	10.2%
Recuperator effectiveness	93%	95%	96%
Recuperator leakage, from cold side to hot side	0.0%	0.0%	1.0%

^(a) These values were used in the sensitivity study as low and high boundaries. For the purpose of the statistical evaluation which combined all low and high values, it was reasonably assumed that these boundary values also represent 2 sigma limits.

Confidence in the point design values for the compressor and turbine polytropic efficiencies and the recuperator effectiveness is high. Those parameters are well studied and industry experience is available to accurately predict the performance values, before the plant is built. Consequently, their associated range of uncertainty is small. Therefore, despite the high sensitivity to these three parameters, their net impact on thermal efficiency is within acceptable limits.

The results of the sensitivity study for thermal efficiency are presented in Figure 4-5. The high-lighted horizontal line at 47.6% thermal efficiency represents the point design. Placed on top of the point design line and on the right side of the uncertainty bars are the point design values which correspond to the eight parameters given on the ordinate axis.

A separate evaluation of the statistical combination of all uncertainties concluded that the mean thermal efficiency will be close to 47.0%. This result shows that the thermal efficiency of the indirect-cycle can be very high, even when design and performance uncertainties are considered. The statistical work assumed that the uncertainty band considered for each parameter covers the 2 sigma range. Most distributions are normal while others which had to encompass design evolution margin, like the recuperator leakage flow, received asymmetrical distributions.

Figure 4-5 RESULTS OF INDIRECT-CYCLE SENSITIVITY STUDY (THERMAL EFFICIENCY)

A similar sensitivity study was done for the net plant efficiency, and those results are shown in Figure 4-6. A mean net plant efficiency of 43.2% was computed, compared to the point design estimate of 43.8%.

4.3 KEY NUCLEAR ISLAND DESIGN SELECTIONS AND BASES

Reactor Vessel Temperature Control

Objective: Contain helium at temperatures up to 950°F (up to 1150°F during loss of forced cooling) and pressure up to 1150 psi within vessel.

Alternatives: SA533/508
Modified 9Cr-1Mo

Discussion: SA533/508 is a well proven material used extensively in light water reactors. The application of this material, however, is limited by the maximum temperature allowable of 700°F for steady-state operation. For the required service conditions of the GT, a complex cooling system is required to keep the vessel wall temperatures below 700°F. This system would use cold, high pressure helium from the compressor discharge. Because the compressor is part of the secondary loop, the primary pressure boundary has to be penetrated twice by the piping which carries the cooling helium to and from the reactor vessel. This very complex cooling system is avoided by using Modified 9Cr-1Mo material, which is rated at significantly higher temperatures than SA533/508. Modified 9Cr-1Mo, however, is not yet code qualified and irradiation data at the GT-MHR service conditions has to be generated.

Selection: Modified 9Cr-1Mo

The selection of Modified 9Cr-1Mo is based on the assumption that performance at least equivalent to that of SA533/508 will be confirmed when irradiation data is available. In addition, it is assumed that the weight of the Modified 9Cr-1Mo vessel will be approximately that of the SA533/ 508 vessel, and that forgings of the required size are available or that weldments will be acceptable for the Modified 9Cr-1Mo vessel.

IHX Configuration and Layout

Objective: Configure IHX to meet safety and maintenance requirements.

Alternatives: Side-by-side above cross-vessel (circulator at bottom)
Side-by-side below cross-duct (circulator at top)
Vertical in-line

Discussion: Figures 4-7 through 4-9 show the principal layout of each of the three alternatives. The key discriminator between the side-by-side versions is the relative location of the circulator, which is the component most likely to require maintenance. Figure 4-7 depicts the layout with the circulator at the bottom of the IHX vessel. A fixture would be required to extract the circulator from the IHX vessel and to move it to a location where access to the overhead crane is provided. In this configuration, the IHX bundle could potentially be removed from the top,

Figure 4-6 RESULTS OF INDIRECT-CYCLE SENSITIVITY STUDY (NET PLANT EFFICIENCY)

MIT MGR-GTI PROGRAM

Figure 4-7 SIDE-BY-SIDE CONFIGURATION (IHX ABOVE CROSS VESSEL)

MIT INGR-GTI PROGRAM

Figure 4-8 SIDE-BY-SIDE CONFIGURATION (IHX BELOW CROSS VESSEL)

2 of 4

Figure 4-9 IN-LINE CONFIGURATION

although a cask may be required. In the design of Figure 4-8, the circulator is located at the top of the IHX vessel. Thus, circulator maintenance and replacement can be accomplished with standard procedures, using the overhead crane. Removal of the IHX bundle would be done by cutting the cross vessel and removing/replacing the entire IHX vessel. The vessel would serve as a cask in this case. The advantage of the in-line configuration shown in Figure 4-9 is to reduce the capital cost by eliminating the reactor vessel bottom head, the top head of the IHX vessel and the cross vessel. In general, however, maintenance access is much more difficult in this configuration.

Selection: Side-by-side vessel configuration with the IHX located below cross-vessel elevation.

4.4 KEY POWER CONVERSION DESIGN SELECTIONS AND BASES

Secondary Cooling Loop Working Fluid

Objective: Select a working fluid which provides high efficiency and an acceptable turbomachinery design.

Alternatives: Helium
Nitrogen

Discussion: The larger specific heat coefficient of helium, as compared to nitrogen, in concert with the differences in thermal conductivity and density, allows for lower mass flow rates and more compact heat exchangers due to better heat transfer characteristics. Leakage of helium, however, is more likely due to the smaller atom diameter of helium. Also, helium as the cycle working fluid requires more compressor stages for the same core inlet temperature, compared to nitrogen, due to required compression work. Closed-cycle helium turbines have been built and operated successfully, but more field experience, almost directly applicable to nitrogen turbines, has been accumulated during the operation of 19 closed-cycle air turbine plants. The use of nitrogen as the working fluid, however, would not provide any directly applicable experience that could support a helium direct-cycle plant, which is the ultimate configuration that achieves the full potential of the MHR high temperature capability.

Selection: Helium

The selection of helium as the working fluid is contingent upon the confirmation of Modified 9Cr-1Mo as an acceptable vessel material to accommodate a core inlet temperature of 490°C. If the core inlet temperature would have to be reduced significantly, the size and number of stages required for the helium turbine would become unacceptable. Another basic assumption with respect to the selection of helium is that a large output closed-cycle helium turbomachine is obtainable.

Twin Shaft Versus Single Shaft Turbomachinery

Objective: Choose a representative turbomachine configuration that meets economic and performance requirements.

Alternatives: Twin shaft
Single shaft

Discussion: Figure 4-10 depicts the single and twin shaft layouts side by side. The twin-shaft configuration provides flexibility in turbomachine design and performance optimization (such as pressure ratio distribution and shaft speed), because the high- and low-pressure rotors are allowed to run at different speeds. This results in a small size high-performance turbomachine design on the high pressure side, the ability to employ a synchronous generator without a frequency converter on the low pressure side, and the reduction of loss of generator load event severity, by adding a low pressure compressor section on the low-speed turbine-generator shaft.

The advantages of the single shaft layout are simplified startup and shutdown operation and simplified plant control. Also, the overspeed margin of the single shaft machine is higher than that of the twin shaft version. The disadvantage of the single shaft machine is that a frequency converter is required, with an attendant loss in plant efficiency.

Selection: Twin shaft

The selection of the twin shaft layout assumes that such a high output closed-cycle helium turbine is achievable.

Intercooling

Objective: Increase thermal efficiency by intercooling between compressor sections.

Alternatives: No intercooler
One intercooler
Two intercoolers

Discussion: Increasing the numbers of intercoolers increases the thermal efficiency of the plant because the required compressor work is reduced. Beyond two intercoolers, however, the gain in efficiency with further addition of intercoolers is only marginal and not economically practical. Because the power conversion portion of the indirect-cycle is not constrained by space requirements, the number of intercoolers is not a very sensitive parameter with respect to packaging complexity.

Selection: Two intercoolers

SECTION 5

MAJOR SYSTEMS AND COMPONENTS

5.1 REACTOR SYSTEM (DC AND IDC)

The Reactor System configuration for both the direct cycle (DC) and indirect cycle (IDC) designs is virtually identical to that of the 450 MWt steam cycle module, as shown in Figures 5-1 and 5-2. The 84 column core initially proposed during the Cost Reduction Study (Ref. 5-1) and more recently confirmed in core configuration trade studies is utilized, as is the 0-12-24 control rod configuration which results from trade studies on the neutron control arrangement. Both designs will operate at a nominal helium coolant pressure of 7.07 MPa (1025 psia) and produce 450 MW of thermal power, using 19.9% enriched uranium fuel (without thorium) in elements which have a core residence time of 3 years, with refueling at 1.5 year intervals. The major differences between the steam cycle design and the gas turbine designs arise primarily because of the following differences in core inlet and outlet temperatures:

	<u>Steam Cycle</u>	<u>Gas Turbine (DC and IDC)</u>
Core inlet temperature °C (°F)	288 (550)	490 (914)
Core outlet temperature °C (°F)	704 (1300)	850 (1562)

The information contained within this section is limited to the effects of these temperature differences, and to any assumptions regarding their effects. Furthermore, because both the DC and IDC designs have the same operating temperatures, as well as the same configuration, the differences in the Reactor System between the two gas turbine designs are relatively small.

5.1.1 Key Design Requirements

The requirements relevant to the evaluation are as follows:

- Reactor System materials shall be compatible with primary helium coolant with specified chemical impurities.
- The Reactor System shall accommodate design temperature, pressure, and flow conditions.
- The Reactor System shall be designed for the maximum flow induced forces resulting from Power Conversion System (PCS) (DC only)/Heat Transport System (HTS) (IDC only) or Shutdown Cooling System (SCS) operation.
- The Reactor System shall be designed for the maximum acoustic pressures resulting from PCS (DC only)/HTS (IDC only) or SCS operation.
- Fuel elements shall limit fractional releases of key radionuclides from the core during normal operation to [TBD].

Figure 5-1 REACTOR CORE CROSS SECTION

Figure 5-2 REACTOR SYSTEM ARRANGEMENT

- Fuel elements shall limit fractional releases of key radionuclides from the core during design basis events to [TBD].
- The core shall be capable of maintaining a minimum shutdown margin with $K_{eff} \leq 0.99$ under the following conditions:
 - At room temperature with only the operating control rods for at least one day.
 - At refueling temperature with all control rods inserted except those withdrawn for refueling, plus the maximum worth rod.
 - At any temperature with all control rods and reserve shutdown control elements (RSCE) inserted during water ingress events.
- The core shall have a negative temperature coefficient under all conditions.
- Reactor System connections to the Vessel System shall accommodate thermal and seismic deflections.
- The Reactor System shall be designed to accommodate the operating basis earthquake (OBE).
- The Reactor System shall be designed to remain functional during and after a safe shutdown earthquake (SSE) to the extent that the plant may be shut down safely and maintained in a safe shutdown condition.
- The Reactor System shall limit neutron fluence to the reactor vessel to [TBD n/cm²].
- The metallic core support shall be designed in accordance with the ASME Boiler and Pressure Vessel Code, Section III, Division 1, Subsection NG, and proposed revision to Code Case N-201 to allow operation at temperatures up to 760°C (1400°F).

5.1.2 Fuel and Core Design

The largest difference in the core between the gas turbine designs (both DC and IDC) and the steam cycle design is the temperature of the helium coolant gas and the core components. The inlet and outlet coolant temperatures are given above in the introduction to Section 5.1. The resulting fuel temperatures are shown in Figures 5-3 and 5-4. It can be seen that the peak fuel temperatures are reduced by introducing a new half column arrangement of fresh fuel at each refueling. In the steam cycle design, which utilized whole column refueling, fresh fuel was installed in alternate full height columns at each 1.5 year refueling. Therefore, the entire fuel column remained in location for 3 years, containing fresh fuel at the beginning of this time and old fuel at the end of this time. In the gas turbine design, fresh fuel will be installed in half the height of every column, alternating between the top and bottom halves of adjacent columns. Therefore, the top half of a column will contain fresh fuel when the bottom half contains old fuel, and the top half of the adjacent column will contain old fuel when the bottom half contains fresh fuel. This fuel zoning arrangement is shown diagrammatically in Figure 5-5. The net

Figure 5-3 HALF-COLUMN REFUELING REDUCES PEAK CORE FUEL TEMPERATURES

Figure 5-4 HALF-COLUMN REFUELING REDUCES PEAK CORE FUEL TEMPERATURES

Figure 5-5 FUEL ZONING SELECTED TO MINIMIZE PEAK TEMPERATURES

effect is that the peak fuel temperature is not increased relative to the steam cycle, even though the nominal coolant temperatures are much higher. (This fuel placement arrangement could have been used in the steam cycle design as well to reduce the peak fuel temperatures of that design). However, the average fuel temperature is not affected by the new fuel placement arrangement, and will increase approximately 170°C relative to the steam cycle design. This will result in an increase in the release of radionuclides from the core during normal operation, and will be a factor in designing for plant maintainability (see Section 11 for additional discussion of this issue). During the limiting depressurized conduction cooldown design basis event, the peak fuel temperature remains below 1600°C, so no significant change in fission product release is expected during accident conditions. However because the actual fuel temperature will increase from 1540°C for the steam cycle to 1584°C for the gas turbine, the margin is reduced and more release is predicted from initially failed fuel particles. (See Section 8.2 for more discussion of this topic.) No changes in the fuel design are judged necessary to accommodate the higher temperatures.

As a consequence of the higher coolant temperatures, and the smaller difference between inlet and outlet temperatures relative to the steam cycle design, the coolant flow rate will increase by approximately 30%. This in turn will increase the pressure drop through the core. After adjusting for increased flow in the gaps between fuel elements, the pressure drop is expected to increase from 27.6 kPa (4.0 psi) in the steam cycle design to approximately 39.3 kPa (5.7 psi) in the GT design. It is assumed that the increased flow rate and pressure drop will not significantly affect the design of the core to resist flow induced forces. This assumption relies on the New Production Reactor (NPR) design which had larger flow rates and pressure drops through fuel element coolant channels.

The higher temperatures may affect the stresses in the fuel elements. Increased thermal expansion of metallic supports will lead to greater gap flow, which will cause element stresses to increase. However the increased creep resulting from the higher temperatures, and a smaller coolant temperature gradient across the core would lead to a reduction in element stresses. It is assumed that, overall, the change in stresses will prove acceptable.

5.1.3 Reactivity Control Design

Reactivity control for the GT designs will be similar to that of the steam cycle design because the core arrangement, power density, total core power, and fuel element configuration are substantially identical. The only differences again arise from temperature differences between the two core designs. For the same fuel loading, the increased temperature for the GT increases the required worth of the control rods to bring the reactor from operation at full power temperature to cold shutdown. The increased temperature also causes a decrease in the maximum capacity factor that can be supported by the fuel cycle. The potential to add more fuel to the core, which could bring the core availability back to that of the steam cycle design, is limited because it results in a reduced negative temperature coefficient. The reduction in negative temperature coefficient will reduce the required worth of the control rods, but it is unlikely that the required worth can be reduced to that of the steam cycle design. The problem is exacerbated because there will also be a reduction in the relative worth of operating control rods (those in the outer reflector) if the fuel loading is increased. Thus, an increase in required worth coincident with a decrease in relative worth will occur. Hence a loss of shutdown margin

is expected for a GT design, compared to the reference steam cycle design, whether or not the core availability is maintained. Adding fuel would also adversely affect the shutdown margin during a postulated worst case water ingress event. The possibility of adding rhodium or erbium to the GT core to overcome these problems is discussed in Section 5.1.5. An optimum reactivity control design for the GT must find a balance among these considerations.

For the steam cycle design, the fuel cycle availability factor has not yet been determined. However, it is presently estimated to be approximately 86% for a refueling interval of 18 months. This is below the value of 88% desired by utilities for the 18 month refueling interval. (See Section 9.2.3 for more discussion on the capacity factor assessment.) Other reactivity control requirements for the steam cycle core which have not yet been completely demonstrated are $K_{\text{eff}} \leq 0.99$ during refueling with all rods inserted except those in one sector that are removed for refueling, plus the maximum worth rod, and a $K_{\text{eff}} \leq 0.99$ during a worst case water ingress event. Demonstration of an adequate reactivity control design in these areas will be slightly more difficult for the GT than for the steam cycle. Overall, it is assumed that an adequate design can be developed; however, a tradeoff may be required between the fuel cycle design capacity factor and the refueling interval.

5.1.4 Reactor Internals Design

The reactor internals designs for the GT and steam cycle are very similar. Again, the major differences result from the increased temperatures of the GT design. As described in Section 5.1.2, there is an increased mass flow rate, due to the smaller core ΔT , and a further increase in volumetric flow as a consequence of the higher average core temperature. To accommodate the increased flow, the hot duct diameter must increase from 1.19 m (47 in) to 1.37 m (54 in). The lower plenum of the Reactor System and the graphite core support post height must increase by the same amount to adequately channel the flow into the hot duct. The core support posts will have to be designed to have adequate resistance to flow induced vibrations, even though they have increased height and are exposed to increased flow velocities. The effects of the increased core outlet temperature on oxidation and corrosion of the posts will be considered in the design.

Most of the metallic reactor internals design is not expected to undergo significant change, since the structural components are constructed of Alloy 800H and exposed to core inlet temperatures during normal operation. These inlet temperatures are well within those that the long duration life of the metals can accommodate at the stresses that are expected. However, the hot duct insulation retainers, which are essentially nonstructural, will be exposed to core outlet temperatures of 850°C (1562°F), with periodic hot streaks up to approximately 900°C (1650°F). It is assumed that the retainer material can remain Alloy 800H, since this material is approved for use in pressure vessels up to temperatures of 900°C (1650°F) in the ASME nonnuclear code (although they are currently only approved for use in nuclear pressure vessels to 760°C (1400°F)), and the insulation retainers are not subjected to long term primary stresses. Should Alloy 800H not prove acceptable, alternative metals are available. These include high purity Hastelloy, which is being used in the Japanese HTTR reactor, and many nickel based alloys which are used in gas turbines.

One difference between the GT-DC and GT-IDC designs is the temperature experienced by the core barrel and upper plenum shroud during pressurized conduction cooldown events. Because the reactor pressure equilibrates at approximately 5.17 MPa (750 psi) in the DC design, the top of the reactor is cooler than in the IDC, where the pressure remains at 7.07 MPa (1025 psi). Consequently the IDC core barrel temperature rises above 760°C (1400°F), while the DC core barrel temperature remains below 760°C (1400°F). This introduces the need for a new or modified code case for the IDC core barrel, since on-going code case activities will not result in the existing 760°C (1400°F) temperature limit of alloy 800H being increased.

5.1.5 Incremental Development Requirements (Relative to Steam Cycle)

The incremental development requirements for the GT, relative to the steam cycle, are small for the Reactor System. It will be necessary to revise design data needs (DDN's) and related Technology Development Plans to reflect the higher temperatures. Furthermore, since the range of the temperatures is increased for the reactor internals, it may be necessary to introduce additional test conditions into the existing (or planned) test matrices.

New testing may be needed if erbium or rhodium is deemed necessary for the core to meet availability and shutdown margin requirements. These tests would be used to demonstrate the stability of the material, and, if coated, the coating process.

For the GT-IDC only, a code case (or code case revision) must be prepared to allow the use of Alloy 800H, or an alternate material, at temperatures above 760°C (1400°F).

No other incremental development activities are anticipated.

5.1.6 Summary of DC/IDC Differences

The only difference of some significance between the GT-IDC and the GT -DC that is anticipated is the need for additional ASME code work to permit Alloy 800H to be used at temperatures above 760°C (1400°F) for the IDC. The need for such a code case is not anticipated for the DC at a core power level of 450 MWt.

5.2 MAIN CIRCULATOR (IDC)

The main circulator is a key component of the IDC GT-MHR that provides for circulation of the primary circuit helium under normal operating and various transient and shutdown conditions. It is not required in the DC option.

5.2.1 Key Design Requirements

The major requirements for the IDC helium circulator closely follow those established for the steam cycle plant, namely that, (1) the main circulator be vertically oriented and installed in the top head of the heat exchanger vessel, (2) the machine utilize magnetic bearings, and (3) design margins be incorporated at the system level (i.e., the circulator must be capable of operating within a defined envelope). A summary of the requirements is given in Table 5-1.

5.2.2 Design Overview

A specific circulator design concept was not established for the IDC, but rather advantage was taken of previous work done by Howden, as described in the following.

5.2.2.1 Steam Cycle Plant Circulator

The design concept established by Howden for the steam cycle (Figure 5-6) has the following features: (1) power input: 5 MWe, (2) active magnetic bearings, (3) two stage axial flow impeller, (4) rotational speed: 4234 rpm, and (5) submerged induction motor drive.

5.2.2.2 Circulator Features for IDC-GT

Howden provided the following input based on a machine concept that would be similar to that shown in Figure 5-6, namely, (1) electric power required at point design conditions: 8.1 MWe, (2) electric power including margins and uncertainties: 13 to 14 MWe, (3) active magnetic bearings, (4) two stage axial flow impeller, (5) rotational speed: 5800 rpm, (6) submerged induction motor drive (configuration yet to be established).

5.2.3 Technical Issues/Limitations

The helium inlet temperature is 490°C (914°F) at steady state conditions, and higher during transients and heat soak. This is of concern, particularly in the magnetic bearing insulation, perhaps requiring a ceramic system. Also the motor windings must be thoroughly protected.

Rotor dynamic stability is a concern based on the following: (1) significant impeller overhang, (2) large bearing span (resulting from long motor rotor/stator), (3) perhaps need for multiple motors and bearings.

Limitations were noted in the following areas: (1) current AGR circulators are rated at 6 MWe, and based on proven technology (112 units in operation), (2) the single motor configuration could be extended to 9 MWe (firm), (3) a dual motor arrangement would be needed for a rating of 12 MWe based on current projections, (4) the potential for a single motor

Table 5-1
HELIUM CIRCULATOR COMPARISON

Circulator Requirements	Steam Cycle Plant	IDC-Gas Turbine Plant
Helium flow rate, kg/sec (lb/sec)	212 (467)	240 (529)
Inlet temperature, °C (°F)	288 (550)	490 (914)
Inlet pressure, MPa (psia)	7.07 (1025)	7.17 (1040)
Pressure rise, kPa (psi)	91.0 (13.2)	~ 145 (~ 21)
Circulator power, MWe	5.0	~ 13.2
Impeller type	2 stage axial	2 stage axial
Rotational speed, rpm	4234	5800
Induction motor drive	Single motor	Dual motors*
Magnetic bearings	3	5 or 6
Technology status	State-of-the-art	Difficult* (but doable)

* An alternative concept with a single motor drive would be more attractive (not studied to date).

12 MWe machine would require study, and (5) dual motor potential for ratings of 18 MWe, or higher, would require a design investigation.

5.2.4 Incremental Development Requirements

Development requirements were not investigated, it being felt prudent to address these only after a concept had been established. The following assumptions requiring future verification are noted:

1. Preliminary design of a magnetic bearing machine has been limited to a 5 MWe variant for the 350 MWt steam cycle plant.
2. Analysis/design must be undertaken for machines to meet much higher power levels.
3. Optimum configuration for a machine rated at 12 MWe or above remains to be determined.
4. Design approaches to accommodate very high helium temperatures must be investigated.

Figure 5-6 MAIN HELIUM CIRCULATOR ARRANGEMENT FOR SC-MHR PLANT

5.3 INTERMEDIATE HEAT EXCHANGER (IDC)

The Intermediate Heat Exchanger (IHX) is the Indirect Cycle (IDC) component which transfers the energy produced in the reactor from the primary helium circuit to the secondary helium circuit which contains the turbomachinery and other power conversion components. The principal functions of the IHX are to isolate the power conversion loop (balance of plant) equipment from the radioactive primary system coolant and to transfer the heat energy from the primary system coolant to the secondary coolant as efficiently as possible.

The IHX is part of the primary coolant pressure boundary and must be designed to maintain the primary boundary integrity. As part of the primary boundary, the IHX shares the radionuclide control functions of the Vessel System by preventing air ingress and consequent core oxidation.

5.3.1 Key Design Requirements

In addition to performing the functions described above, the IHX also must satisfy the following key design requirements:

- The IHX shall withstand the operational environment given in Table 5-2.
- The IHX shall accommodate all plant design duty cycle events within limits specified by the ASME Boiler & Pressure Vessel Code.
- The IHX shall be designed for a [40-year] plant life.
- Primary and secondary coolants are helium/helium.

5.3.2 IHX Design Overview

The design selections chosen to satisfy the key design requirements are:

- The IHX is located side-by-side with the reactor and at a lower elevation, to suppress natural convection flow and provide thermal isolation of the IHX from the high temperature core during conduction cooldown events.
- The IHX will be designed and fabricated in accordance with the requirements of the ASME Code, Section III, to ensure high reliability throughout the [40-year] plant life.
- The IHX tubing and central downcomer materials and sizing are chosen to withstand full external primary pressure at a tube temperature of 816°C (1500°F).

The IHX, shown in Figure 5-7, is a once-through, counter-flow, cross-flow, helium/helium heat exchanger. It has primary helium on the shell side and secondary helium on the tube side of a helical coil tube bundle. The return secondary helium from the power conversion loop enters the secondary inlet plenum at the bottom of the IHX vessel, where it is distributed to the

Table 5-2
IHX Operational Parameters at 100% Load

<u>Primary Helium</u>			
Mass Flow Rate	kg/s (10 ³ lb/hr)	239.4	(1900)
Inlet He Temperature	°C (°F)	850	(1562)
Outlet He Temperature	°C (°F)	485	(905)
Inlet Pressure	MPa (psia)	7.00	(1016)
Maximum Pressure Drop	MPa (psid)	0.04	(6)
<u>Secondary Helium</u>			
Mass Flow Rate	kg/s (10 ³ lb/hr)	250.0	(1984)
Inlet He Temperature	°C (°F)	425	(797)
Outlet He Temperature	°C (°F)	775	(1427)
Inlet Pressure	MPa (psia)	7.46	(1082)
Maximum Pressure Drop	MPa (psid)	0.21	(30)

individual tubes at the secondary inlet tubesheet. It then flows upward through the helical tube bundle, picking up heat from the primary helium. The tube bundle terminates in a cylindrical tubesheet at the top of the central downcomer. The hot secondary helium enters and flows downward through the central downcomer and into the power conversion loop piping.

The primary helium flow from the reactor enters the IHX through the hot duct and the inlet diffuser, flows downward through the tube bundle, giving up its heat to the secondary helium. The colder primary helium, exiting the tube bundle, returns to the main circulator at the top of the IHX vessel through the annulus between the vessel wall and the tube bundle outer shroud. The main circulator outlet helium returns to the reactor through the cross vessel/hot duct annulus.

The IHX tube bundle is made up of 1120 smooth tubes, 44.5 mm (1.75 in) O.D. and 48.4 m (158.8 ft) long, not counting the lead-ins and lead-outs. The tubes are arranged in 27 tube columns with a transverse pitch of 55.6 mm (2.19 in). Table 5-3 lists the IHX sizing results based on using Inconel 617 tubing.

The tube bundle support concept is shown in Figures 5-8 through 5-10. Figure 5-8 shows the scalloped vertical strips and wear rings that make up the tube supports. Each vertical support is made up of three strips which fit together as shown in Figure 5-9 to completely capture the helical tubes and wear rings. The arrangement results in tongue-and-groove tube

SECTIONAL ELEVATION

Figure 5-7 IHX ELEVATION

Table 5-3 IHX SIZING DATA

Heat Duty, MWt	454.0
LMTD, °C [°F]	49.4 [121]
HX Surface Area, m ² [ft ²]	7572 [81,500]
Tube Length, m [ft]	48.4 [158.8]
Bundle Depth, m [ft]	11 [36.2]
Bundle O.D., m [ft]	4.84 [15.88]
Bundle I.D., m [ft]	1.94 [6.38]
Tube Material	Inconel 617
Tube O.D., mm [in]	44.45 [1.75]
Tube Thickness, mm [in]	5.156 [0.203]
Helix Angle, Degrees	13.18
Number of Tubes	1120
Number of Columns	27
Transverse Pitch, mm [in]	55.63 [2.19]
Longitudinal Pitch, mm [in]	59.69 [2.35]
Shellside Pressure Drop, kPa [psi]	40.9 [5.93]
Tubeside Pressure Drop, kPa [psi]	204.8 [29.71]

support plates that rest on a "wagon wheel" support as depicted in Figure 5-10. The concept should provide adequate structural support to withstand design duty cycle events and sufficient flexibility to accommodate differential thermal expansions within the tube bundle.

The IHX inlet diffuser is a flow distribution device which is designed to redirect the hot helium flow from horizontal to vertical and to provide good flow distribution at the tube bundle entrance. It is believed that its purpose can be achieved with minimum pressure loss (without the use of flow screens) and within a minimum vertical height.

Figure 5-8 TUBE SUPPORT COMPONENTS

Figure 5-9 TUBE SUPPORT COMPONENTS ASSEMBLY

Figure 5-10 TUBE BUNDLE ASSEMBLY

5.3.3 Technical Issues/Limitations

Technical issues for the IHX are related to the selection of materials for high temperature operation. The primary coolant temperature, entering the IHX, is 850°C (1562°F). However, the maximum temperature, for which allowable stresses are listed in ASME Code Case N-47-29, is 816°C (1500°F). Therefore, whatever material is proposed for the IHX inlet diffuser, tube bundle and central downcomer will require submittal of an ASME Code Inquiry, with all of the attendant supporting high-temperature properties test data.

The currently proposed material for the high-temperature, primary pressure boundary applications is Inconel 617. It has been approved, in Code Case 1956-2, for use in Section I and in Section VIII, Division 1, at temperatures up to and including 899°C (1650°F). Another Code Case, 1982, extends the useful temperature range of Inconel 617 alloy in Section VIII, Division 1, up to and including 982°C (1800°F). The effects of irradiation on the material properties will need to be evaluated. However, radiation levels in the IHX will be much lower than for the reactor vessel.

The availability of Inconel 617 tubing for the IHX is not in question. However, the central downcomer is 1448 mm (57 in) inside diameter and 203 mm (8 in) thick. It may have to be fabricated from rolled and welded plate rather than forged shells, but this is common practice in the fabrication of large diameter, heavy-wall pipe.

There will have to be one or more bimetallic welds between the Inconel 617 heat exchanger and the 9Cr-1Mo-V IHX vessel, depending on which of the two materials is chosen for the secondary coolant inlet tubesheet. Inconel to ferritic welds are commonplace in pressurized water reactor (PWR) fabrication and do not cause concern. However, because the GT-MHR operating temperatures are much higher than those of the PWR, a thorough analysis will be required.

Details of the IHX inlet diffuser were not determined during the GT-MHR evaluation period. However, it should be possible to design and fabricate such a device from relatively thin shells, and with thermal insulation to prevent regenerative heat transfer between the inlet and outlet regions of the IHX. Figure 5-11 shows a possible arrangement of a diffuser, fabricated in such a manner. The inlet diffuser, like the hot duct, can be categorized as a flow baffle or shroud; an internal part which separates the high-temperature, incoming primary coolant from the lower-temperature, outgoing primary coolant. It is not part of the primary pressure boundary.

In-service inspection (ISI) is not a requirement for the steam generator tubing in the SC, and a similar case could be made for the IHX tubing, based on the provision of secondary piping isolation valves. Since the IHX design incorporates individual tubing circuits, inspection of the tubing is considered feasible, although the methods and equipment would have to be developed.

The current IHX design does not incorporate heat transfer enhancements, and it is recognized that more compact heat exchanger designs could be achieved using finned tube arrangements. These, and other methods of compact heat exchanger design should be evaluated further to determine the potential for reducing the IHX size.

PRIMARY FLOW DISTRIBUTOR
(DIFFUSER)

Figure 5-11 IHX INLET DIFFUSER CONCEPT

5.3.4 Development Requirements

Development requirements for the IHX fall into two categories:

- **Material Properties** - Because Inconel 617 is not approved for use in Section III, a materials database will be required to support an ASME Code Inquiry. Since the material is approved for use in Sections I and VIII at temperatures up to 982°C (1800°F), a significant database of high temperature properties must be available. However, the effects of long term irradiation on the material properties must also be evaluated.
- **Design Development/Confirmation** - The IHX design features have been selected based on extrapolations of, and evolutions from existing heat exchanger designs, some of which have not been constructed and operated. Therefore, test programs will be required to develop some design features and to verify the adequacy of others. Areas where design development or confirmation will be required include:
 - Inlet Flow Distribution Tests
 - Acoustic Response Tests
 - Large Helical Coil Fabrication/Assembly Development
 - Insulation Design Development
 - Shroud Seal Development
 - Lead-in/Lead-out Expansion Tests
 - Flow-Induced Vibration Tests
 - Tube Bundle Flow Resistance and Heat Transfer Tests
 - Tube ISI Methods and Equipment Development

Detailed requirements for each of the areas, listed above, are yet to be determined. And development requirements in addition to those listed above may be identified in the course of IHX design activities.

5.4 PRIMARY SYSTEM VESSELS (DC and IDC)

The principal functions of the primary system vessels, or Vessel System (VS), are to contain the primary coolant inventory and to maintain primary coolant boundary integrity. In addition, the VS provides structural support and alignment for the Reactor System components that are housed within the reactor vessel (RV) and, for the DC, all power conversion components that are housed within the power conversion vessel (PCV), or for the IDC, all components housed within the IHX vessel (IH XV).

The radionuclide control functions of the VS are to transfer decay heat from the reactor core to the reactor cavity cooling system (RCCS) during conduction cooldown events, to maintain the geometry of the reactor core with respect to the neutron control assemblies (NCAs) to control heat generation, and to prevent air ingress and consequent core oxidation.

5.4.1 Key Design Requirements

In addition to performing the functions described above, the VS also must satisfy the following key design requirements:

- The VS shall withstand the operational environment, including a design pressure of [TBD] and design temperature of [TBD]. [Design pressure and temperature have not yet been specified. However, the parameters that have been used to determine vessel shell thicknesses are listed in Table 5-4].
- The VS design shall accommodate all plant design duty cycle events, which may cause more severe operating conditions than the design conditions, within ASME Boiler & Pressure Vessel Code, Section III requirements.
- The VS shall be designed for a [40-year] plant life.
- The VS design shall limit primary coolant leakage to a minimum (Steam Cycle MHR requirements limit leakage to 10% of the VS-contained inventory per year).
- The VS design shall minimize the costs of capital equipment, operations and maintenance.

5.4.2 Vessel System Design

The design selections chosen to satisfy the VS key design requirements are:

- The reactor vessel (RV) will be uninsulated to permit core residual and decay heat removal during conduction cooldown events by radiation heat transfer to the RCCS.
- To enhance the radiation heat transfer, both the inner and outer surfaces of the RV will be processed, as required, to provide high emissivity (values > 0.8 are required).

Table 5-4 VESSEL SYSTEM PRINCIPAL DESIGN PARAMETERS

VESSEL	REGION OF VESSEL	DESIGN PRES-SURE	DESIGN TEMPERATURE/DURATION
Power Conversion Vessel (DC Only)	Closure Head Flange and Generator Housing	7.98 MPa, gage (1158 psig)	(50°C) 122°F
	Vessel Flange and Upper Shell	7.98 MPa, gage (1158 psig)	510°C for 3x10 ⁵ Hours (950°F)
	Lower Shell and Bottom Head	7.98 MPa, gage (1158 psig)	127°C (260°F)
IHX Vessel (IDC Only)	Primary Pressure Boundary Shells	7.93 MPa, gage (1150 psig)	490°C for 3x10 ⁵ Hours (914°F)
	Secondary Pressure Boundary Shells	8.41 MPa, gage (1220 psig)	425°C for 3x10 ⁵ Hours (797°F)
Reactor Vessel and Cross Vessel (DC and IDC)	Entire Vessel	7.93 MPa, gage (1150 psig)	495°C for 3x10 ⁵ Hours (923°F)
			538°C for 1x10 ³ Hours (1000°F)

- The VS and its components will be designed and fabricated per the requirements of the ASME Boiler & Pressure Vessel Code, Section III, to ensure high reliability throughout the 40-year plant life.
- The vessel components will be fabricated from 9Cr-1Mo-V ferritic steel forgings per SA-336 Grade F91 and plates per SA-387 Grade 91, Class 2 (see Section 5.4.3 Selection of Vessel Materials).
- The VS in-service inspection program will comply with the ASME Code, Section XI, requirements.

The general arrangement of the VS for the DC GT-MHR is illustrated in Figure 5-12. The VS components for the DC concept include:

Figure 5-12 DC-GT-MHR GENERAL ARRANGEMENT

- The reactor vessel (RV)
- The cross vessel (CV)
- The power conversion vessel (PCV)
- The vessel supports and lateral restraints
- The primary pressure relief trains.

The DC VS is located below grade level, enclosed in a reinforced concrete silo. The RV and PCV are placed side-by-side with the PCV at a lower elevation than the RV. This arrangement provides thermal isolation and protection of the power conversion components from the high temperature core during conduction cooldown events.

The general arrangement of the VS for the IDC-GT-MHR is shown in Figure 5-13. The list of components comprising the IDC is the same as for the DC except that the PCV is replaced by an IHXV. The IDC VS also is located below grade elevation in a reinforced concrete silo, placed similarly to the DC components, and for the same reason.

Principal operating and design parameters for the VS components are listed in Table 5-4.

5.4.2.1 Reactor Vessel (RV)

The RV design, shown in Figure 5-14, is the same for both GT-MHR concepts. It is 7.226 m (23 ft - 8.5 in) inside diameter, with a nominal shell thickness of 203 mm (8 in). The lower cylindrical shell thickness is 248 mm (9.75 in) to accommodate the RV support loads and cross vessel nozzle loads. The 165 mm (6.5 in) thick, hemispherical bottom head is welded to the lower cylindrical shell at the core support ring, and contains the shutdown cooling heat exchanger (SCHE) penetration.

The SCHE housing and shutdown cooling circulator housing designs are included in the RV design scope because they are part of the primary pressure boundary and are mounted on the RV.

The hemispherical RV closure head is bolted to the RV flange. The closure head is 203 mm (8 in) thick for increased stiffness because it contains the neutron control assembly (NCA) housings and the in-core flux mapping unit (IFMU) housing.

Overall dimensions of the RV are 8.395 m (27 ft - 6.5 in) flange outside diameter by 31.37 m (102 ft - 11.0 in) length. Estimated weight of the RV is 1252 metric tons (1380 tons).

5.4.2.2 Cross Vessel (CV)

The CV design, shown in Figure 5-15, is a one-piece forged cylinder, 2.16 m (85 in) inside diameter by 76.2 mm (3.0 in) thick. Its length is approximately 3 m (10 ft) for the DC concept: 3.66 m (12 ft) for the IDC concept. Actual length of the CV will be determined later, after detail design of the CV nozzles. The CV is designed according to the ASME Code, Section III rules for vessels.

Figure 5-13 IDC-GT-MHR GENERAL ARRANGEMENT

Figure 5-14 GT-MHR REACTOR VESSEL

Figure 5-15 GT-MHR CROSS VESSEL

5.4.2.3 Power Conversion Vessel (PCV)

The PCV configuration is shown in Figure 5-16. The PCV is 6.71 m (22 ft) inside diameter. The upper cylindrical shell operates at approximately 510°C (950°F) and is 216 mm (8.50 in) thick. It is locally increased in thickness to 241 mm (9.50 in) to accommodate the CV nozzle loads and the vessel support loads. The lower cylindrical shell operates at less than 127°C (260°F) and is 171 mm (6.75 in) thick. It contains the cooling water inlet and outlet penetrations for the intercooler modules. The hemispherical bottom head is 108 mm (4.25 in) thick; it contains the cooling water inlet and outlet penetrations for the precooler modules. The PCV closure head is 4.115 m (162.0 in) inside diameter. The closure head thickness is 108 mm (4.25 in) thick.

Overall dimensions of the PCV are 7.899 m (25 ft - 11.0 in) flange diameter by 27.00 m (88 ft - 7.0 in) long. Estimated weight of the PCV is 726 metric tons (800 tons).

5.4.2.4 Intermediate Heat Exchanger Vessel (IHXV)

The IHXV, shown in Figure 5-17, is 5.506 m (216.8 in) inside diameter. The main vessel operates at near core inlet temperature, 490°C (914°F), and is 178 mm (7.0 in) thick. The IHXV ellipsoidal closure head contains the main circulator penetration and also is 178 mm (7.0 in) thick.

The secondary inlet plenum, located at the bottom of the IHXV, operates at the secondary cold leg temperature, 425°C (797°F). It is 4.19 m (165 in) inside diameter, by 114.3 mm (4.5 in) thick. The central downcomer is 1.45 m (57 in) inside diameter, by 203.2 mm (8 in) thick. The proposed downcomer material is Inconel 617.

The horizontal, annular ring, that separates the secondary inlet plenum from the main vessel, is a 356 mm (14 in) thick tubesheet for the intermediate heat exchanger (IHX). The tubesheet material is the same as the vessel material, 9Cr-1Mo-V ferritic steel.

Overall dimensions of the IHXV are 6.55 m (21 ft - 6.0 in) flange diameter by 34.40 m (112 ft - 10.2 in) to the centerline of the secondary outlet elbow at the bottom of the IHXV.

5.4.3 Selection of Vessel Materials

As noted in Reference 5-2, the initial approach in designing the GT-MHR VS was to apply the standard light water reactor (LWR) technology vessel materials, SA-533 Grade B, Class 1 plate and SA-508 Class 3 forgings of Manganese-Molybdenum steel. Many reactor years of experience and an extensive database exist for these materials, and they were the vessel materials of choice for the SC-MHR. However, if these materials are used, the vessel primary coolant pressure boundary would have to be maintained at or below 371°C (700°F) during normal operation and during most transient events. Since the RV must be uninsulated, the suggested approach for controlling the reactor vessel temperature was to bypass a small fraction of the cold helium flow from the compressor outlet and to circulate it in the annular space adjacent to the

Figure 5-16 DC-GT-MHR POWER CONVERSION VESSEL

RV shell. The use of internal insulation was suggested to control the temperatures of the other vessels.

An alternative to the use of cold bypass flow was to design and fabricate the vessels from a high temperature ferritic steel, such as 9Cr-1Mo-V. The 9Cr-1Mo steel, used primarily for heat exchanger tubes in the past, has been stabilized and strengthened by the additions of Columbium, Vanadium and Nitrogen. The "Modified 9Cr-1Mo" alloy was developed by Oak Ridge National Laboratory and Combustion Engineering, Inc. during the 1970s and 1980s to extend and optimize its high temperature properties. Although the material has been primarily used as tubing and piping in fossil applications, the developers of the steel envisioned its use in nuclear steam generators and vessels in liquid metal reactor applications. The use of 9Cr-1Mo-V in the construction of heavy-wall vessels for refinery applications has been proposed by the French. Section thicknesses up to 300 mm (12 in) have been produced.

There are some issues related to the use of 9Cr-1Mo-V materials, particularly in heavy sections:

- An issue related to heavy sections, being addressed by ASTM, is concerned with acceptable cooling methods for pipes (and presumably plates) produced to A335 (and presumably A387). Heavy sections, when air cooled, may contain ferrite, which appears to reduce the creep strength. Manufacturers need to water quench to produce through-thickness martensite. There is some concern, however, that if the hardness of the martensite, due to water quenching, is too high, as in 12 Cr steels, creep strength may again be lost. Currently, the forging specifications, A 182 and A 336, permit water quenching, and the European and Japanese producers water quench thick-wall pipe and plate.
- Stress-rupture data obtained by ORNL from surface, quarter thickness and center locations of 200 mm (8 in) plate indicate slightly lower strength at the center location. Stress-rupture data produced by the Japanese for quarter thickness and center locations in 270 mm (10 in) plate indicated no difference in strength. A 550 mm (21.65 in) forging, produced in Japan, was found to have lower strength than a 25 mm (1 in) plate; however, this was later traced to low nitrogen content in the forging. Overall, there appears to be no trend with regard to strength versus product thickness. However, if ferrite forms in the thicker sections, some loss of strength will occur. The topic of thickness effects needs further study.
- Toughness data are available for thick section 9Cr-1Mo-V, but it will take some time to collect and evaluate the data.
- Welds have been produced and tested at ORNL on 200 mm (8 in) plate. However, the results of stress rupture tests on weldments depend on sample size and location, filler metal composition, weld process, plate thickness and post-weld heat treatment. Much remains to be learned about weldments of 9Cr-1Mo-V. Meanwhile, tables of weld strength reduction factors are available.

- A significant data base has been established characterizing the effects of fast neutron ($E > 0.1$ MeV) irradiation on the mechanical properties of 9Cr-1Mo-V in support of liquid metal reactor development (Ref. 5-3). Samples were irradiated in the EBR-II, FFTF and HFIR reactors, and data were obtained on tensile, creep and fracture toughness, as well as changes in microstructure and swelling. Both the neutron spectra and neutron fluences were far more severe than those which would be encountered in the GT-MHR. Additional data will be required to qualify 9Cr-1Mo-V for the specific operating conditions of the GT-MHR. However, every indication is that 9Cr-1Mo-V will be at least as good as SA-533/SA-508 low alloy steel with respect to irradiation properties.
- There is a need to develop an emittance data base for 9Cr-1Mo-V for use in the design and evaluation of the passive Reactor Cavity Cooling System.

9Cr-1Mo-V ferritic steel has been approved by the ASME Code Committee for use in Section I boilers and in Section VIII, Division 1 unfired pressure vessels. ASME Code Case N-466-1 allows its use in Section III, Division 1, Class 1, 2 and 3 components at temperatures up to 371°C (700°F), and Code Case N-253-6 allows its use up to 649°C (1200°F) for Class 2 and Class 3 components. A Code Inquiry is being pursued by Oak Ridge National Laboratory and ABB-CE for inclusion of 9Cr-1Mo-V alloy in the Section III, high temperature Code Case N-47.

Other material options include 2¼Cr-1Mo-¼V and 3Cr-1Mo-¼V which have short term strength properties slightly higher, and long term strength properties slightly lower than 9Cr-1Mo-V. 2¼Cr-1Mo-¼V is approved for use in Section VIII, Divisions 1 and 2 per Code Case 2098-1. However, a Code Inquiry would be required for its inclusion in Code Case N-47. The 2¼Cr-1Mo-¼V is lower cost material than 9Cr-1Mo-V, but lacks the higher temperature capabilities (e.g., transients above 900°F). Applicability of the 2¼Cr-1Mo-¼V should be reassessed as the design evolves, for use as a backup material. A database would have to be developed on the effects of irradiation on the strength and ductility of the 2¼Cr-1Mo-¼V alloy. 3Cr-1Mo-¼V, likewise, is approved for use in Section VIII, Divisions 1 and 2 per Code Case 1961-1. It also would require a Code Inquiry for inclusion in Code Case N-47, and an irradiation database is not available for the alloy.

Based on the foregoing discussion, the logical material choice for the GT-MHR primary system vessels is the 9Cr-1Mo-V alloy. However, it may be advisable to develop an irradiation database for one, or both, of the other material options, and to prepare and submit an ASME Code Inquiry for inclusion of the materials in Code Case N-47, so that alternative material choices may be available, if needed.

5.4.4 Technical Issues/Limitations

The key technical issues for the primary system vessels include materials selection for high temperature operation, as discussed in paragraph 5.4.3, above, the availability of large ring forgings, sealing of the vessel flanges and penetrations, and transportation of the completed major components to remote sites (i.e., sites located away from navigable waterways).

5.4.4.1 Availability of Large Ring Forgings

The availability of large ring forgings of 9Cr-1Mo-V is a concern. The ASME "Specification for Steel Forgings, General Requirements", SA-788, states in paragraph 5.1 that "The primary melting may incorporate separate degassing or refining and may be followed by secondary melting, using electroslag remelting (ESR) or vacuum arc remelting (VAR)." This additional refining process is used by some forging suppliers to ensure clean and homogeneous material. Japan Steel Works, a leading supplier of large forgings, uses the electroslag remelting (ESR) process for refinement of 9Cr-1Mo-V. Their ESR furnace has an ingot capacity of 100 metric tons, which they estimate will yield a 30 to 40 metric ton finished forging weight. The heaviest ring forging in the RV is the closure head flange, at 112.0 metric tons, finished weight. The closure head dome forging weight is 69.0 metric tons. The vessel flange weight is 98.1 metric tons, and the core support ring is 69.0 metric tons.

Creusot-Marrel melting and refining processes produce clean and homogeneous materials without using electroslag remelting. Their forging capability is up to 120 metric tons, finished weight, 6.10 m (20 ft) outside diameter and 508 to 635 mm (20 to 25 inches) thickness for shells. C-M plate capacity is up to 100 metric tons, finished weight, 21.84 m (860 in) length, 4.32 m (170 in) width, and up to 889 mm (35 in) thickness. They have produced industrial heats of 9Cr-1Mo-V in thicknesses up to 300 mm (12 in).

The maximum U.S. forging capability is at Ladish. Their size limit is 9.14 m (30 ft) outside diameter (as forged), 3.05 m (10 ft) length and 159 metric tons (175 tons).

In correspondence with Creusot-Marrel, it was suggested that it may be possible for them to produce 9Cr-1Mo-V hollow ingots and perform the initial forging steps at C-M, then have Ladish complete the forgings that exceed the C-M outside diameter limit. C-M was extremely interested and expressed plans to contact Ladish to explore that possibility further.

An alternative to using large ring forgings is to fabricate the flanges and core support ring in sections from heavy forged bars that are formed to the required radius and welded. ABB-CE has welded heavy sections up to 559 mm (22 in) in thickness with good results. The major disadvantage of fabricating the rings with this method is that additional in-service inspection will be required during plant life. Limits to the use of this method are overall diameter (vertical boring mill turning radius limit) and weight (handling ease).

5.4.4.2 Vessel Seals

Seal design, for containing high pressure helium, is always a concern. And, because of the high vessel operating temperatures that occur with the GT-MHR concepts, the importance of the seal design is amplified.

Helicoflex seals were selected for use on the Steam Cycle primary vessels. As shown in Figure 5-18, the Helicoflex seal is a metallic O-ring, containing a close-wound spiral Inconel spring inside of an Inconel jacket, with a silver jacket on the outside. The silver jacket will flow under the sealing load and fill any minor scratches or irregularities in the sealing surfaces. The Helicoflex seal has been used extensively in the French nuclear program and in many other

Figure 5-18 HELICOFLEX SEAL CROSS SECTION

applications. It has been used to replace other seal types in problem situations and appears to be the state-of-the-art in sealing. The Helicoflex seal has had extensive testing, by the Helicoflex Company and by others. A report on "Metallic Seal Testing" performed under DOE Contract PE-AC04-760P00789 by Sandia National Laboratory, published the results of tests on four different metal seal configurations. The report concluded that the Helicoflex seal performed best by a large margin. In fact, it was the only seal recommended out of the four seals tested. Another report on vacuum seal tests for the Tokamak Fusion Test Reactor at Princeton Plasma Physics Laboratory, found the Helicoflex seal satisfied the sealing requirements for that program.

According to the Helicoflex Company, the seals can be readily manufactured in the required diameters for the GT-MHR vessels. The Helicoflex seals are rated for use at temperatures up to 750°C (1382°F). The primary system vessels operate at temperatures up to 495°C (923°F).

An alternative sealing concept, which may be used on flanges that do not require frequent opening, is the welded omega seal, such as is shown in Figures 5-19 and 5-20.

5.4.4.3 Transportation Limits

Limits to barge transportation from ABB-CE, Chattanooga, should not constrain the shipment of completed major components for the GT-MHR to plant sites located on or near navigable waterways. The clearance limit is set by a fixed-span bridge over the Tennessee River at Scottsboro, Alabama. The clearance is 17.2 m (56.4 ft) above normal pool elevation. The maximum clearance requirement for the RV, secured to a shipping rig, mounted on a barge deck, is approximately 11m (35 ft).

For plant sites located away from navigable waterways, route surveys by experienced, heavy-hauling companies will be required to determine the numbers and types of obstacles to transportation and the potential means of removal or circumvention for each obstacle. Limits exist in the form of "per axle" or "per wheel" loads on highways, bridges and rails, and permits must be obtained from state and local governments.

Once on site, 1090 metric ton (1200 ton) translift cranes are available and can be used in pairs, if necessary, to lift and set in place the major components. Also, 544 metric ton (600 ton) crawler/transporters can be used in multiples to move the components any appreciable off-road distances.

5.4.5 Incremental Development Requirements

In paragraph 5.4.3, the recommendation was made to use 9Cr-1Mo-V materials for the primary system vessels and to allow the vessels to operate at temperatures up to 495°C (923°F). If that recommendation is accepted, then any developmental work that has been completed, or is in progress, on the Steam Cycle vessel materials will not be applicable to the GT concepts. New Design Data Needs (DDNs) will be required for the 9Cr-1Mo-V materials at the higher operating temperatures. The new DDNs will include:

Figure 5-19 WELDED OMEGA SEAL INSTALLATION

Figure 5-20 WELDED OMEGA SEAL DETAIL

- A data base for determining the neutron-induced, nil-ductility transition temperature (NDTT) shifts for SA-336 Grade F91 forgings, SA-387 Grade 91 Class 2 plate, weldments and heat-affected zones;
- Data to demonstrate that the reactor vessel material surfaces can be processed to achieve a thermal emissivity of at least 0.8 for the vessel temperatures that occur during conduction cooldown events.

There are two DDNs, for the Steam Cycle vessel materials, that will not be required for the GT vessel materials:

- Material properties at elevated temperatures were required to provide a data base for ASME Code Case N-499. That DDN has been completed and the Code Case has been approved. As mentioned in Section 5.4.3, a Code Inquiry is being pursued by ORNL and ABB-CE for inclusion of 9Cr-1Mo-V alloy in Code Case N-47. Approval of that Inquiry is anticipated, and it is unlikely that additional data will be required.
- Tests were required to define the creep relaxation properties of SA-240 B24 Class 3 bolting material for the Steam Cycle reactor vessel, for duty cycle events in which the bolting temperature would exceed 316°C (600°F). The GT vessel bolting will be fabricated from Inconel 718, a high temperature bolting alloy that is already included in Code Case N-47. Therefore, additional data will not be required.

5.4.6 Summary of DC/IDC Differences

With respect to the primary system vessels, the difference between the two GT concepts is that the DC includes the PCV and the IDC includes the IHXV. The RV is the same for both vessel system concepts, the CV is similar for both concepts and both concepts serve the same functions, as stated in Section 5.4.1. However, the PCV and IHXV house very different internal components. Both vessel system concepts are feasible, with no concerns that are unresolvable.

5.5 RECUPERATOR (DC AND IDC)

The recuperator is a gas-to-gas heat exchanger that recovers energy from the helium exiting the turbine and utilizes that energy to preheat the compressed helium before it enters the reactor. This energy recovery step is extremely important in attaining high cycle efficiency, due to the high temperature and, thus, energy content of the turbine exhaust stream. In conventional open combined cycle gas turbine (CCGT) plants, the bottoming steam cycle serves a similar function. Since the helium on both sides of the recuperator is common to a single circuit, the recuperator is not a helium pressure boundary component. There is, however, a significant pressure difference between the high and low pressure portions of the cycle and significant leakage would result in loss of cycle efficiency. Given the competing requirements of high effectiveness, low pressure losses and minimal volume, the design of the recuperator is both important and demanding.

5.5.1 Key Design Requirements

Table 5-5 presents the nominal steady state performance requirements for both the DC and IDC recuperators. These performance requirements result from plant level trade studies that concluded with the point designs summarized in Sections 3 and 4 for the DC and IDC, respectively.

The temperatures given in this table represent a thermal effectiveness for the recuperator of 0.95 for both the DC and IDC. This is the major key design requirement for the recuperator. Almost as important is the pressure drop specification of 0.02 (as the sum of the two streams' $\Delta P/P$ values), which was also selected for both the DC and IDC. These two specifications, in conjunction, have a tremendous effect on the recuperator design.

Effectiveness specifications for past and current gas-turbine applications have ranged from 0.75 to 0.88, with current practice typically in the range 0.82 to 0.88. Using Professor London's (Stanford University) rule for heat exchanger difficulty or size: the size of a heat exchanger is proportional to the reciprocal of one minus the effectiveness: $1/(1 - \epsilon)$ (see Ref. 5-4), i.e., the DC or IDC recuperator at .95 effectiveness would be some three times larger than an 0.85 effectiveness unit. In order to achieve this high effectiveness, the design must carefully address the uniformity of the helium velocity distribution at the hot and cold inlets.

The pressure drop specification is also about three times more demanding than for current gas-turbine practice, where a value for $\Delta P/P$ of 0.06 is typical, albeit at lower pressure levels.

The temperature levels for the GT-MHR do not represent a significant challenge, as gas-turbine units operating in aircraft and various ground power service applications typically see temperatures well above this range.

A design consideration is the approximately 5.55 MPa (660 psi) pressure differential between the two sides of the recuperator. Very few, if any, plate-fin heat exchangers have been built for such duty. Typically the high side of the pressure differential between the two sides of a recuperator is around 1.4 MPa (200 psig). However, calculations and the judgment of experienced heat-exchanger designers indicate that this is not a major impediment to producing

Table 5-5 RECUPERATOR REQUIREMENTS

Fluid Stream	Flow Rate kg/s (10 ³ lbm/hr)	Inlet Temperature °C (°F)	Outlet Temperature °C (°F)	Inlet Pressure MPa (psia)	Pressure Drop Sum (ΔP/P)
<u>Direct Cycle</u>					
Hot, Lower Pressure From Turbine	240.3 (1907)	510 (950)	132 (269)	2.63 (382)	0.01
Cold, Higher Pressure from Compressor	240.3 (1907)	112 (233)	490 (915)	7.18 (1042)	0.01
<u>Indirect Cycle</u>					
Hot, Lower Pressure From Turbine	250 (1984)	442 (828)	109 (228)	2.55 (370)	0.01
Cold, Higher Pressure from Compressor	250 (1984)	91 (196)	425 (797)	7.52 (1091)	0.01

the recuperator. The approach to containing this large pressure differential is to use somewhat thicker gauge materials than in current practice.

While transient performance requirements have not yet been addressed for the GT-MHR, it is worthwhile noting that the transient thermal environment for the recuperator is not nearly as severe as has been encountered in several previous programs.

The relatively small volume allotted to the DC recuperator array is a factor in the design of the units, but it is not a threat to the design goals. For the IDC, which has a distributed pressure boundary architecture, the volume of the recuperator is not as significant.

5.5.2 Basic Recuperator Design Characteristics

The recuperators for both the DC and IDC options use modern offset plate-fin surfaces, with the entire heat-exchanger structure (fins, plates, side bars, and manifolds) being brazed to form a very strong, monolithic unit. Figure 5-21 shows the basic construction of a plate fin heat exchanger, in comparison with other heat exchanger types. The major life limiting phenomenon in such heat exchangers is low-cycle fatigue, brought about by thermal stresses in the heat exchanger. These plate-fin recuperators have been designed to minimize such stresses and to offer a very long life.

The techniques used in their design follow those already proven in several programs which have produced working recuperators for propulsion and pipeline gas-turbine engines. In these applications (especially with propulsion engines), the recuperators see many more start/stop and low-load/high-load cycles than will be seen in the MHR application, and the propulsion-system transients are much more severe than in this power generation application.

tube-and-shell

plate-fin

prime-surface

- Choice dependent on meeting design criteria at minimum cost

Figure 5-21 PLATE-FIN HEAT EXCHANGER BASIC CONCEPT

The plate-fin type of construction is essentially immune to acoustic vibrations because the amplitudes of the sound waves are expected to be much larger than the plate-fin flow passages. Therefore, it is expected that the MHR acoustic environment will not be a factor in achieving long life times.

The recuperators for both the DC and IDC are comprised of several individual heat exchangers acting in parallel. This approach is necessary in the DC in order to meet the strict envelope constraints. Parallel units are also used in the IDC to help in the packaging of the recuperator.

Considering the key design requirements, as outlined in Section 5.5.1, a number of design aspects were adopted at the outset of the work. These are discussed in the next several paragraphs.

In the effectiveness range of most gas-turbine recuperators, and certainly those for the GT-MHR, a counterflow configuration is required; the crossflow configuration simply is out of the question at these effectiveness levels. For the purpose of introducing and withdrawing fluids, all *counterflow* plate-fin heat exchangers must use crossflow sections at both ends. Figure 5-22 shows some typical counterflow plate-fin flow configurations. These sketches represent the flows in the plane parallel to the plates of the device.

In order to reach the effectiveness value of 0.95, it is necessary that the flow distribution to the various stacked passages be very close to uniform. Any non-uniformity in flow distribution reduces the inherent ability of the heat-exchanger core to reach its intended performance. Note that the principal differences in the heat exchanger arrangements in Figure 5-22 relate to the relative flow paths of the fluids in the entry/exit sections. The terms Z-flow and U-flow are used to describe these options. Of these possible arrangements, the Z-flow configuration provides the most efficient use of surface area. This is explained as follows: in plate-fin heat exchangers there are, in actuality, no defined flow channels within a single flow passage. However, with regard to flow distribution across a passage, it is instructive to envision separate flow channels or flow filaments across the heat exchanger passage. In the Z-flow configuration, the various flow channels are all of the same length. In the U-flow configuration, the flow channels are of varying lengths. This variation in length inherently produces a non-uniform flow distribution in a U-flow heat exchanger. This non-uniformity will decrease the unit's effectiveness over a similar Z-flow unit. Thus, for the same effectiveness, a U-flow entry/exit design will result in a larger heat exchanger than a Z-flow design. The centerfed U-flow configuration represents a compromise between the Z-flow and U-flow designs. While somewhat less efficient in surface area utilization than the Z-flow unit, the centerfed U-flow unit is somewhat easier to design and manufacture.

The several fin-containing flow channels in the plate-fin recuperator are stacked one upon another, alternating between hot-stream and cold-stream passages (Figure 5-21). It is necessary to distribute flow to these various passages from the connecting piping. This is accomplished by manifolds (sometimes called headers) which are essentially conduits which run perpendicular to the planes of the flow passages, and which deliver fluid to, or withdraw fluid from the various passages. Based upon the relative flow configuration of the flows in the manifolds, Z-and U-flow configurations are possible. Figure 5-23 shows some manifold configurations. Comparison

PLAN VIEW OF COUNTERFLOW PLATE-PIN HEAT-EXCHANGER CONFIGURATIONS

Figure 5-22 PLATE-FIN HEAT EXCHANGER FLOW CONFIGURATIONS

of Figures 5-22 and 5-23 should make clear the differences in heat-exchanger passage configurations and manifold flow configurations.

In contrast to the entry/exit sections of the recuperator, U-flow is the preferred flow configuration within the manifolds. Z-flow manifolds (of the same areas as U-flow manifolds) always produce higher pressure losses within the system, and less uniform flow. In fact, from theoretical grounds (related to the change in pressure with decreasing flow along the length of the manifold), uniform flow can only be produced in Z-flow if the manifolds are either very large, or if the inlet manifold is tapered (as shown in Fig. 5-23). Tapered manifolds would be very difficult to produce with this type of heat exchanger, and the actual taper would have to be obtained experimentally.

5.5.3 DC Recuperator Design

Because of the volume constraints imposed by the power conversion vessel, the design of the DC recuperator is particularly challenging. For this reason, the design of the DC recuperator is treated in some detail in the following subsections. Afterwards, the design of the IDC recuperator is summarized in Section 5.5.4, with emphasis upon the differences.

5.5.3.1 Selection of Flow Paths

Because of the combined requirements in the DC of (1) very high effectiveness, (2) low pressure-drop allowance, and (3) very tight envelope, it was decided that the heat-exchanger flow regime with respect to the entry/exit sections would be Z-flow. Similarly, it was decided that the manifold configuration for the high-pressure cold flow would have to be U-flow.

For the low pressure side of the recuperator, the approach was somewhat different. The lower-pressure, hot turbine exit flow enters the annular recuperator envelope at the top of the array of heat exchangers. The hot flow is directed downward in a plenum which is formed partially by appropriate baffling between recuperators and the pressure vessel, and partially by the inlet faces of two paired recuperators. The hot flow enters the recuperator passages, transfers heat to the cold, higher pressure stream, and exits into the relatively large volume of the pressure vessel. (This is shown later in Fig. 5-25.)

The basic nature of this flow arrangement of hot helium - downward in a plenum, into the recuperators, and then out of the recuperators and downward to the precooler - resembles a Z-flow manifold system. In general, for high-effectiveness applications, Z-flow manifold arrangements should be avoided. Fortunately, the areas that can be used for the distribution and collection of the hot flow are large enough to allow the attainment of 0.95 effectiveness with this geometry.

5.5.3.2 Recuperator Envelope

The original annular recuperator envelope for the DC was as follows:

SIDE VIEW MANIFOLD CONFIGURATIONS

U-FLOW MANIFOLDS

**Z-FLOW MANIFOLDS
CONST. AREA INLET MANIFOLD**

**Z-FLOW MANIFOLDS
WITH TAPERED INLET MANIFOLD**

Figure 5-23 MANIFOLD CONFIGURATIONS

Outside diameter	6.10 m (20.0 ft)
Inside diameter	3.66 m (12.0 ft)
Length	4.57 m (15.0 ft)
Allowable Arc	310 deg

Working with this envelope, it became apparent that there was not sufficient room to house the several recuperator heat exchangers and still have sufficient flow area for hot-gas distribution. All types of known fin surfaces were tried in several different possible plate-fin layouts, with 4, 5, or 6 parallel units.

Coordination between Allied-Signal and General Atomics led to a redefinition of the envelope as follows:

Outside diameter	6.71 m (22.0 ft)
Inside diameter	3.66 m (12.0 ft)
Length	5.79 m (19.0 ft)
Allowable Arc	310 deg

The significant change in the envelope is the outer diameter increase to 22 ft.

5.5.3.3 Core Description

The initial DC recuperator design involved five heat exchangers in parallel. This design was fully checked with analytical tools at Allied-Signal, including heat-transfer in the core, cold-side manifold flow distribution, hot-side "manifold" flow distribution, and stress. It was considered that this design would meet all requirements of the problem statement.

However, with an odd number of parallel units, the sealing between recuperator cores, and between the cores and the pressure vessel would be cumbersome. These seals or baffles are required to direct turbine discharge flow into the recuperator inlets (as discussed above), and prevent it from mixing with the outlet flow. Therefore, a new design involving six parallel heat exchangers was produced. Figure 5-24 shows a planform of this refined design.

Each heat exchanger has a core width of approximately 914 mm (36 in). The counterflow section length, where essentially all of the heat exchange takes place, is 940 mm (37 in), with an overall length (including manifolds) of about 2260 mm (89 in). The stack height is about 4110 mm (162 in). The design of the entry sections has included considerations for packaging in the annular envelope, providing adequate room for the high-pressure manifolds and sufficient flow area for the hot-gas distribution into and out of the recuperator.

This recuperator design uses very compact offset plate-fin surfaces in the counterflow (center) section. Such interrupted heat-transfer surfaces are sometimes called "strip fins." They provide a very high level of heat-transfer enhancement with low pressure drop, along with high surface area per unit volume. The triangular entry and exit sections employ plain (non-offset) fins to distribute the flows to/from the fins in the counterflow section.

DIRECT-CYCLE RECUPERATOR PLANFORM

Figure 5-24 DC RECUPERATOR PLANFORM

The recuperator uses a construction concept employing tube plates which are contoured to integrally form the sealing barriers at the sides of the flow passages and around the manifold rings. This concept, using "formed tube sheets," has been employed for several years by Allied-Signal in a variety of applications.

The material of construction for all parts of the recuperator is 316L stainless steel. This low-carbon, austenitic stainless steel was chosen for its excellent brazability, and its adequate strength at the operating temperatures.

The various flow passages - formed by the contoured tube plates and the fins contained between them - would normally be stacked one upon another and brazed into a single unit. If produced in one integral unit, the height of the recuperator module (the "stack height") would be 4105 mm (161.6 in). In actuality, each recuperator module will be composed of a number of separately brazed units (of approximately 762 mm (30 in) height) which will be welded together with appropriate fittings. The number of individual units to be used in a single stack has not yet been determined. However, the final stack height of the recuperator will not be much larger than the 4105 mm (161.6 in) value, which, of course, is much less than the allowable 5790 mm (228 in).

The calculated weight of the bare core (thermally active portions of one heat exchanger) is about 14,500 kg (32,000 lb). With the addition of mounting provisions, mating piping conditions, etc., the weight of a single unit would be approximately 15,970 kg (35,200 lb). The weight of six units would be approximately 95,800 kg (211,200 lb).

The cold-side high-pressure manifolds are integrally constructed with the heat exchanger core. They have been sized to provide nearly perfect distribution of flow to the various flow passages along the stack height.

5.5.3.4 Pressure Drop Allocation

The overall pressure drop (sum of the two $\Delta P/P$'s) allocation for the DC recuperator is 0.0200. The core (no manifolds) was designed for a pressure drop of 0.0180, providing a 10% allowance for manifolds and connecting piping. This specification is given to the design program as a sum, rather than individual pressure drops on each side, since it is practically impossible to predefine pressure drops for both sides and meet them precisely in counterflow units. (With cross flow heat exchangers this is possible.)

After the design process for the manifolds was completed, the pressure drop of the recuperator unit increased to 0.0185. This leaves 7.5% of the overall pressure drop for connecting piping.

5.5.3.5 Layout in the Pressure Vessel

Figure 5-25 shows the layout of the six DC recuperator heat exchangers in the pressure vessel annular envelope. Also shown on this figure are the regions which will be sealed off down the length of the recuperator modules to produce the inlet plenum (manifold) for the hot, lower-pressure flow. The rest of the annular envelope can be considered as the outlet plenum

FIGURE 5-25 DC RECUPERATOR LAYOUT

for the hot side. The large differences in flow area between inlet and outlet plenums are intentional in order to promote uniform flow in this Z-flow manifold configuration.

Provisions will be made to provide nearly uniform flow distribution of the hot, turbine outflow to the three recuperator inlet plenums.

Cold, high pressure helium from the compressor stages is ducted to the bottom of the recuperators and distributed to the six individual inlet manifolds. Six heat exchangers are positioned in a back-to-back arrangement, so as to allow more efficient setup of the connecting piping.

5.5.3.6 Connecting Piping and Manifolds

The proper design of the inlet/outlet transitions is essential for achieving the design recuperator effectiveness of 95%. The design requirements associated with the inlet/outlet transitions are:

- Distribute helium mass flow uniformly across the frontal area of the hot side inlet.
- Position cold side inlet/outlet to accommodate U-flow.
- Minimize pressure losses in inlet/outlet transitions.

To fully use the available heat transfer surface area, the pressure distribution at the hot side inlet has to provide uniform inflow to the recuperator. Flow guide vanes at the turbine discharge are utilized to normalize the highly turbulent flow coming from the turbine outlet. Because the frontal area is significantly smaller at the hot gas inlet plenum than at the hot gas outlet plenum, the resulting back pressure assists in achieving uniform gas inflow to the unit. As seen in Figure 5-25, the hot gas inlet plenum is located inwards, away from the power conversion vessel wall, to avoid contact of the hot helium at 510°C (950°F) with the uninsulated vessel inside surface. Within the recuperator, the helium is cooled to 132°C (269°F) and contact with the vessel inside surface is permissible on the outlet side of the recuperator.

Because the chosen cold-side manifolding arrangement is the preferred U-shape, both the inlet and outlet of the cold side have to be located at the same end of the unit. Positioning the inlet/outlet piping at the bottom allows for easier maintenance and replacement access to the recuperator from the top, and also clears the top of the recuperator for unconstrained design of the hot side inlet area. The selected design solution incorporates two circular donut-shaped headers, which collect cold side inlet (from the high pressure compressor) and cold side outlet helium (from the recuperator outlet), respectively. Inlet and outlet pipes associated with each of the six recuperator modules are individually connected to those headers. Flanges are provided in the inlet/outlet pipes to accommodate removal of the recuperator module. From the cold side outlet header, the helium flows upwards into the outer portion of the annular cross-vessel. Hot side plena and cold side pipes are designed with smooth transitions and large cross-sectional areas to minimize pressure losses. The exception is the hot side inlet plenum, which has to provide a required minimum pressure loss for reasons discussed above.

5.5.4 IDC Recuperator Design

The IDC recuperator is also designed for an effectiveness of 0.95 and a combined relative pressure loss of 0.0200. However, the envelope for the IDC recuperator is not nearly as constrained as that of the DC. Because of this, a more conventional *centerfed U-flow* configuration was chosen. This design, although inherently somewhat larger (see Section 5.5.2), has certain advantages related to stress design, manufacture, and plant ducting over the Z-flow configuration used for the DC. The IDC design also employs somewhat looser and taller fins than the DC. This also has advantages in the manufacturing process, lowering the cost.

Figure 5-26 shows the resultant centerfed U-flow planform that was selected to provide the required heat transfer within the allowed pressure drop. Four parallel recuperator units are used in the IDC system. Each heat exchanger is approximately 1524 mm (60 in) wide, 3175 mm (125 in) long (including the integral manifolds), and 3023 mm (119 in) in height (stack height). Each core weighs about 25,400 kg (56,000 lb), or total of 101,600 kg (224,000 lb) for the four heat exchangers.

The four IDC recuperator modules are contained within two ASME Section VIII pressure vessels, each containing two cores. The cores are oriented so that the high pressure piping runs straight through the end plates. Each end plate penetration for the high pressure pipes (4 per vessel) is provided with an expansion joint.

The turbine exhaust helium enters the top of the vessel through two nozzles provided with thermal shields. Each thermal shield is welded to a shroud attached to each core which serves as a plenum to direct the gas to the lower pressure core inlet. The shroud will be welded to the core (or to bars built into the core to facilitate alignment and welding) to provide a seal between the low pressure inlet and outlet. No shroud is provided on the low pressure core outlet so that the shell of the vessel will be exposed to the lower temperature, lower pressure helium.

It is possible that the core itself could be qualified as an ASME VIII pressure boundary. This would greatly simplify the design by eliminating the large cylindrical shell and substituting individual plenums on the low pressure inlets and outlets of each core.

Consideration was also given to manifolding the connections from two or four cores inside a pressure vessel to reduce the number of connections to the vessel. It was decided that this would increase the volume of the vessel (to allow for the space required for the piping as well as access to make the piping welds) and increase the potential for thermal stress problems with few benefits in compensation. In fact, having multiple (smaller) piping connections is beneficial for the thermal stresses in the external piping runs.

5.5.5 Technical Issues/Limitations

The major technical issues in the design and development of the DC recuperator are summarized below:

- Flow distribution to the various passages of the six parallel heat exchangers on the hot, turbine discharge side

INDIRECT CYCLE RECUPERATOR

Figure 5-26 IDC RECUPERATOR

Manifold computations indicate that the vertical flow distribution will be very close to uniform. However, the geometry is very complex and further aerodynamic analysis is warranted. The turbine discharge flow must be uniformly directed to the three recuperator inlet areas at the top of the recuperator array.

- Mechanical design of the high-pressure inlet and outlet connections

The design of the connections between the inlet and outlet ducting and the recuperator cores is critical in the areas of stress, pressure drop, and thermal growth. This requires careful design coordination between General Atomics and Allied-Signal Aerospace.

- Mechanical design of the recuperator end sections and the integral manifolds

This area is critical due to the high pressure differential they will see.

- Heat leaks out of the turbomachinery-recuperator enclosure

Heat leaks out of the enclosure have not yet been addressed with respect to the overall efficiency of the recuperation process. If there is too much heat leakage from the high temperature gas, the effectiveness of the recuperator could be compromised.

For the IDC recuperator, the following issue has been identified:

- Heat leaks out of the recuperator enclosure

The surfaces of the enclosure provide a significant potential for heat losses, potentially compromising the effectiveness of the recuperator.

5.5.6 Incremental Development Requirements

There are several aspects of the recuperator design that need to be verified, given the novelty of using a plate-fin recuperator design in a nuclear high pressure helium environment. These key design concerns are:

Materials Compatibility. The initial material selection (316 stainless steel) is different from that in the brazed plate-fin recuperators used for industrial gas turbines (i.e. 409 stainless steel). There must be assurance that the selected material is compatible with the reactor environment for the required life of [40] years, and that the braze alloy is compatible with the helium circuit chemistry (e.g. effect of silver).

Module Integrity. The internal recuperator pressure differential for this application is around 4.8 MPa (700 psid), which is much higher than typical industrial applications [around 0.8 MPa (120 psid)]. Testing may be required to verify the integrity of the plate-fin type of construction under these more demanding conditions.

Helium Flow Distribution. Helium flow maldistribution has a major adverse effect on the performance of high effectiveness heat exchangers. Testing will probably be required to simulate the flow conditions at the recuperator hot and cold inlets. Simulated conditions would include also the impact of cold bypass flow (mixing effect on the hot side inlet flow).

Leakage Detection Within Recuperator. There are two types of potential leakages within the recuperator, 1) plate perforation within the matrix and 2) surface leaks at the headers and manifolds. Since the recuperator is a helium-to-helium heat exchanger, a leak will impact only the plant performance. If there is a loss of plant power output, it will be difficult to diagnose whether it is attributable to recuperator leakage, bypass flow, or seal leakage within the system. Accordingly some testing is necessary to characterize the recuperator "leakage signature". The most likely means of leak detection will be by the use of acoustic instrumentation. The tests should be aimed at addressing the following: 1) identification of which module is leaking, and 2) monitoring of leakage degradation.

5.6 INTERCOOLER AND PRECOOLER (DC)

The principal functions of the precooler and intercooler are to reduce the temperature of the primary helium at the inlets to the first stage and second stage compressors, respectively.

5.6.1 Key Design Requirements

In addition to the principal functions, stated above, the precooler and intercooler designs must satisfy the following key design requirements:

- The precooler and intercooler shall be designed to accommodate all plant design duty cycle events (yet to be determined) in accordance with ASME Code, Section III requirements.
- The heat exchanger (HX) designs shall be sufficiently compact to fit within the following allocated volumes in the power conversion vessel:

Precooler	- Inner Diameter:	3.20 m (10.5 ft)
	- Outer Diameter:	5.58 m (18.3 ft)
	- Overall Height:	3.23 m (10.6 ft)

Intercooler	- Inner Diameter:	2.93 m (9.6 ft)
	- Outer Diameter:	5.27 m (17.3 ft)
	- Overall Height:	2.53 m (8.3 ft)

- The precooler shall be designed to transfer 123.5 MWt at 100% of its rated load. The corresponding conditions are provided in Table 5-6.
- The intercooler shall be designed to transfer 98.7 MWt at 100% of its rated load. The corresponding conditions are provided in Table 5-6.
- The HXs shall be designed for a [40-year] plant life.
- The HXs shall be designed to accommodate in-service inspection, leak detection and isolation (plugging) of individual tube circuits.

5.6.2 Intercooler and Precooler Design

The selection of a reference heat exchanger design was accomplished through a comparison of three candidate configurations against the design requirements, listed above. The three candidates were:

- Modular radial flow heat exchangers with vertical tubes, as shown in Figure 5-28;
- Modular "rectangular" (vertical flow) heat exchangers with horizontal tubes, as shown in Figure 5-29; and,
- Helical coil (vertical flow) heat exchangers, as shown in Figure 5-30.

Table 5-6 PRECOOLER/INTERCOOLER DESIGN POINT CONDITIONS

<u>Primary Helium</u>	<u>Precooler</u>	<u>Intercooler</u>
Mass Flow, kg/s (10 ³ lbs/hr)	240.3 (1907)	240.3 (1907)
Inlet Temp., °C (°F)	132 (269)	112 (233)
Outlet Temp., °C (°F)	33 (91)	33 (91)
Inlet Press., MPa (psia)	2.61 (378)	4.33 (628)
Max. Pressure Drop, kPa (psi)	19 (2.8)	19 (2.8)
<u>Secondary Water</u>	<u>Precooler</u>	<u>Intercooler</u>
Mass Flow, kg/s (10 ³ lbs/hr)	*	*
Inlet Temp., °C (°F)	27.2 (81)	27.2 (81)
Outlet Temp., °C (°F)	*	*
Inlet Pressure, kPa (psia)	34 (50)	34 (50)

* Waterside conditions are based on a maximum water flow velocity through the tubes of 10 ft/s. Mass flows through the HXs and consequent outlet temperatures vary for each concept depending on the number and size of tubes selected. Resultant values for the candidate concepts are shown in Tables 5-7, 5-8 and 5-9.

Initial design emphasis was placed on a radial flow, finned-tube heat exchanger configuration. The use of finned tubes provides increased heat transfer area per unit of tube length on the primary helium (shell) side of the heat exchanger as compared to smooth tubes. Therefore, it was expected to be the most compact concept. Typical applications for this type of heat exchanger are for much lower energy transfer. Radial HXs typically contain large numbers of small diameter, finned tubes which are closely packed and run between common headers, where mixing can occur after each tube pass. However, for the GT-MHR application, individual tube circuits are required to accommodate in-service inspection, leak detection and tube plugging. In order to define individual tube circuits, 180° tube bends are required at the ends of each tube pass instead of common headers. Even using minimum radius tube bends still results in large spaces between tube passes that are ineffective for heat transfer but which increase the overall volume required for the heat exchanger. Also, the vertically oriented tubes in the radial HXs are not totally drainable, and the tight radius tube bends make in-service inspection difficult. However, the radial heat exchanger configuration, as defined in Table 5-7, very nearly fits within the allocated volume. With some minor adjustments, it can be designed to fit in the allotted space.

Figure 5-27 RADIAL PRECOOLER/INTERCOOLER

Figure 5-28 "RECTANGULAR" PRECOOLER/INTERCOOLER

Figure 5-29 HELICAL PRECOOLER/INTERCOOLER

Table 5-7 RADIAL PRECOOLER/INTERCOOLER SIZING DATA

Tube Bundle:	PRECOOLER	INTERCOOLER
HEAT DUTY (MWt)	123.6	98.6
Water Flow Rate, kg/s (lbs/hr)	1051 (8345X10 ³)	939 (7452X10 ³)
Water Outlet Temp., °C (°F)	55.6 (132)	52.2 (126)
LMTD, °C (°F)	26.9 (48.5)	22.7 (40.8)
No. of Modules	12	12
No. of Passes	8	8
No. of Rows/Pass	4	4
No. of Tubes	9792	9024
No. of Lead-ins/Lead-outs, (circuits)	1224	1128
Bundle Height, with bends, mm (in)	2375 (93.5)	2471 (97.3)
Heated Tube Length/Pass, mm (in)	1958 (77.1)	2057 (81.0)
Total Heat Transfer Surface Area, m² (ft²)	1.26X10 ⁴ (1.36X10 ³)	1.23X10 ⁴ (1.32X10 ³)
Bundle Width, mm (in) (Plate to Plate)	869 (34.2)	803 (31.6)
Bend Height, mm (in) (Inside Plate to Top of Bundle)	208 (8.17)	208 (8.17)
Tube Material	1/2Cr-1/2Mo	1/2Cr-1/2Mo
Tube OD/Thickness, mm (in)	22.2/1.65 (0.875/0.066)	22.2/1.65 (0.875/0.066)
Fjn Height/Thickness, mm (in)	3.1/0.3 (0.122/0.012)	3.1/0.3 (0.122/0.012)
Fins/mm (Fins/in)	1.2 (30)	1.2 (30)
Circumferential Pitch, mm (in)	33.4 (1.315)	33.4 (1.315)
Radial Pitch, mm (in)	28.9 (1.139)	28.9 (1.139)
Inside Bundle Radius to CL of Tube, mm (in)	1621 (63.8)	1524 (60.0)
Outside Bundle Radius to CL of Tube, mm (in)	2781 (109.5)	2685 (105.7)
Minimum Vessel Radius, mm (in)	3352 (132)	3352 (132)
Shellside Pressure Drop, kPa (psi) (Heated Region)	6.21 (0.90)	3.59 (0.52)
Tubeside Pressure Drop, kPa (psi) (Heated Region)	80.7 (11.7)	79.3 (11.5)

As an evolutionary step, in order to make the radial heat exchanger tubes totally drainable, a "rectangular" heat exchanger configuration, with horizontally oriented finned tubes, was considered. However, to fit within the allocated annulus, the "rectangular" heat exchanger required a larger number of shorter length tube passes than the radial configuration. And, because of the 180° tube bends, each tube pass requires additional ineffective volume. As a result, the "rectangular" heat exchanger, as defined in Table 5-8, requires much more vertical height than was allocated. Also, because vertical helium flow through the heat exchangers requires a 180° reversal of flow direction at either the inlet or the outlet of each heat exchanger, additional volume is required within the power conversion vessel to accomplish the flow reversal.

Finally, helical coil heat exchangers were sized for the precooler and intercooler applications. The results, tabulated in Table 5-9, show that the helical configuration makes more efficient use of the allocated heat exchanger volume. In fact, if finned tubes are used for the helical coil heat exchangers, the resulting volumes are much smaller than the allocations. However, the helical coil designs also require vertical helium flow with 180° reversal at either the inlet or the outlet. And, the tube lead-ins and lead-outs, shrouds and supports will require additional volume. Therefore, the total precooler/intercooler allocation may be required and should be maintained until more detailed design can be accomplished.

Based on the evaluations, the results of which are summarized above, the helical coil configurations are recommended for the precooler and intercooler applications for the 450 MWt DC GT-MHR plant.

The helical coil designs make maximum use of the allocated volumes, with fewer, although longer, finned tubes. The tube bundles are totally drainable and incorporate individual tube circuits from the inlet tubesheet to the outlet tubesheet. Therefore, each tube circuit is inspectable and plugable from outside of the power conversion vessel.

The water velocity in the tubes is approximately 10 ft/s. This reduces the potential erosion/corrosion phenomena for subcooled waterflow in pipes, and allows for less stringent water chemistry requirements. Simultaneously, the resulting water-side heat transfer coefficient remains high.

The helium flow, from the recuperator, is downward through the annulus between the intercooler and the vessel shell. It enters the top of the precooler (see Figure 5-30), flows downward through the precooler, then turns upward to the inlet of the first stage compressor. The helium leaving the first stage compressor is routed downwards to enter the top of the Intercooler where it flows downward through the heat exchanger, then turns upward to the inlet of the second stage compressor.

The water flow enters the bottom of the HXs through the lead-in tubes and flows helically upward, producing cross-flow/counter-flow in the HXs. It exits the HXs at the top via lead-out tubes. The water enters and exits the tubes through 4 inlet and 4 outlet tubesheets for each HX. The tubesheets are located at the inlet and outlet nozzles in the power conversion vessel.

Table 5-8 "RECTANGULAR" PRECOOLER/INTERCOOLER SIZING DATA

Tube Bundle:	PRECOOLER	INTERCOOLER
HEAT DUTY (MWt)	123.6	98.6
Water Flow Rate, kg/s (lbs/hr)	1881 (14930X10 ³)	1928 (15300X10 ³)
Water Outlet Temp., °C (°F)	42.8 (109)	39.4 (103)
LMTD, °C (°F)	30.0 (54.0)	25.9 (46.7)
No. of Modules	12	12
No. of Passes	16	16
No. of Rows/Pass	4	5
No. of Tubes	10368	11904
No. of Lead-ins/Lead-outs (circuits)	648	744
Bundle Height, Tube CL to Tube CL, mm, (in)	4458 (175.5)	5502 (216.6)
Heated Tube Length/Pass, mm (in)	1054 (41.5)	871 (34.3)
Total Heat Transfer Surface Area, m ² (ft ²)	1.24X10 ⁴ (1.33X10 ⁵)	1.17X10 ⁴ (1.26X10 ⁵)
Bundle Width, mm (in) (Plate to Plate)	800 (31.5)	770 (30.3)
Bend Height, mm (in) (Inside Plate to Top of Bundle)	324 (12.75)	389 (15.32)
Tube Material	1/2Cr-1/2Mo	1/2Cr-1/2Mo
Tube OD/Thickness, mm (in)	38.1/2.84 (1.50/0.112)	38.1/2.84 (1.50/0.112)
Fin Height/Thickness, mm (in)	3.28/0.3 (0.129/0.012)	3.28/0.3 (0.129/0.012)
Fins/mm (Fins/in)	1.2 (30)	1.2 (30)
Circumferential Pitch, mm (in)	57.15 (2.250)	59.18 (2.330)
Longitudinal Pitch, mm (in)	57.15 (2.250)	59.18 (2.330)
Inside Radius Edge of Tube Bend "Rmin", mm (in)	1493 (58.78)	1436 (56.54)
Outside Radius Edge of Tube Bend "Rmax" mm (in)	3195 (125.8)	3086 (121.5)
Minimum Vessel Radius, mm (in)	3251 (128)	3353 (132)
Shellside Pressure Drop, kPa (psi) (Heated Region)	14.5 (2.1)	15.2 (2.2)
Tubeside Pressure Drop, kPa (psi) (Heated Region)	66.2 (9.6)	43.4 (6.3)

Table 5-9 HELICAL PRECOOLER/INTERCOOLER SIZING DATA

Tube Bundle:	PRECOOLER	INTERCOOLER
HEAT DUTY (MWt)	123.6	98.6
LMTD, °C (°F)	20.2 (36.4)	17.6 (31.7)
Water Flow Rate, kg/s (lbs/hr)	543 (4312X10 ³)	543 (4312X10 ³)
Water Outlet Temp., °C (°F)	81.1 (178)	70.6 (159)
Total Heat Transfer Surface Area, m ² (ft ²)	1.76x10 ⁴ (1.89X10 ⁵)	1.59x10 ⁴ (1.71X10 ⁵)
Heated Tube Length/Circuit, m (ft)	53.7 (176.2)	48.46 (159.0)
Heated Bundle Height, mm (in)	1864 (73.4)	2113 (83.2)
Bundle Outer Diameter, mm (in) (Tube CL)	5751 (226.4)	4572 (180.0)
Bundle Inner Diameter, mm (in) (Tube CL)	3366 (132.5)	1869 (73.6)
Tube Material	1/2Cr-1/2Mo	1/2Cr-1/2Mo
Tube OD/Thickness, mm (in)	28.6/2.16 (1.125/0.085)	28.6/2.16 (1.125/0.085)
Fin Height/Thickness, mm (in)	3.18/0.3 (0.125/0.012)	3.18/0.3 (0.125/0.012)
Fins/mm (Fins/inch)	1.2 (30)	1.2 (30)
Helix Angle (°)	1.99	2.50
Number of Tubes	388	388
Number of Columns	31	35
Transverse Pitch, mm (in)	39.8 (1.565)	39.8 (1.565)
Longitudinal Pitch, mm (in)	39.8 (1.565)	39.8 (1.565)
Shellside Pressure Drop, kPa (psi) (Heated Region)	13.8 (2.0)	17.2 (2.5)
Tubeside Pressure Drop, kPa (psi) (Heated Region)	211 (30.6)	195 (28.3)

5.6.3 Technical Issues and Limitations

No specific technical issues have been identified, and no technical limitations have been approached in the recommended helical coil heat exchanger concepts. The thermal environments are benign and the tubes are sized to withstand the full shellside (external) pressure.

The selected tube material is ½ Cr-½ Mo Steel. After the plant design duty cycle transients have been determined, and the heat exchanger designs have been evaluated for the resulting

temperatures and pressures, the tube material selection will be revisited and either verified or changed.

5.6.4 Development Requirements

The effects of the high flow velocities on the primary helium side of the heat exchangers will be quantified and evaluated in a test program which is outlined below. Also, the design methods (thermal sizing and flow resistance) for finned-tube heat exchangers will be verified and/or refined using the test results.

The design data needs (DDNs) presently identified are:

- Inlet Flow Distribution Tests
- Acoustic Response Tests
- Shroud Seal Tests
- Flow Induced Vibration Tests
- Finned Tube Bundle Flow Resistance and Heat Transfer Tests
- Tube Inspection Methods Development.

Additional DDNs may be identified during the detailed design development.

5.7 INTERCOOLERS AND PRECOOLER (IDC)

5.7.1 Key Design Requirements

The indirect cycle (IDC) Precooler and Intercoolers reject waste heat from the secondary helium cycle to circulating water which is then cooled by an evaporative cooling tower. The helium flow rate and the helium inlet and outlet temperatures were established as described in Section 4.4 and are reproduced in Table 5-10.

Table 5-10 PRECOOLER AND INTERCOOLER OPERATING CONDITIONS

	Precooler	Intercooler 1	Intercooler 2
Flow, kg/hr (lb/hr)	899,900 (1,984,000)	899,900 (1,984,000)	899,900 (1,984,000)
Inlet Temperature, °C (°F)	109 (228)	72 (162)	91 (196)
Outlet Temperature, °F (°F)	30 (86)	30 (86)	30 (86)
Inlet Pressure, MPa (psia)	2.51 (364)	3.32 (481)	4.99 (723.5)
Maximum Helium Pressure Drop, kPa (psia)	19.3 (2.8)	14.5 (2.1)	17.2 (2.5)

The helium outlet temperature is to be achieved when the ambient air wet bulb temperature is 18°C (65°F). Both the precooler/intercoolers and the cooling tower contribute to the difference between the helium outlet temperature and the ambient wet bulb. The state of the art for mechanical draft cooling towers is an approach (outlet water temperature minus ambient air wet bulb temperature) of about 4.4°C (8°F). Therefore the precooler and intercoolers were designed for a cooling water inlet temperature of 23°C (73°F).

To utilize standard fill materials in the cooling tower, the maximum temperature of the water returning to the tower must be less than 54°C (130°F) at the maximum expected ambient air wet bulb temperature. To prevent formation of scale deposits with the quality of water normally maintained in evaporative cooling systems, the maximum temperature of the tube wall in contact with the water should be kept below 71°C (160°F) at the maximum expected ambient air wet bulb temperature.

The heat exchangers will be shell and tube designs using straight tubes with circulating water on the tube side to facilitate tube cleaning. (In the absence of the need for cleaning, it would be desirable to have the helium on the tube side because it has the higher design pressure.) Allowances for fouling will be based on the Recommended Good Practice section of the TEMA Standards. On the tube side, a value of 0.18 hr m²°C/J (0.001 hr ft²°F/Btu) for cooling tower water with treated makeup was used. The TEMA Standards do not give a

recommended value for helium. The helium is expected to be very clean so $0.09 \text{ hr m}^2\text{°C/J}$ ($0.0005 \text{ hr ft}^2\text{°F/Btu}$), the lowest value shown for any fluid, was used on the shell side.

The maximum tube velocity will be based on the Heat Exchange Institute Standards for Power Plant Heat Exchangers. No specific velocity limits were imposed on the shell side, but the specified maximum pressure drop and tube vibration constraints have the effect of limiting shell side velocities.

The maximum tubesheet diameter considered was 3048 mm (120 in) based on the capabilities of major heat exchanger manufacturers.

5.7.2 Intercooler and Precooler Design

The design selected for the intercoolers and precooler for the IDC are depicted in Figures 5-31 and 5-32, respectively. The bases for the design selections are summarized in the following.

5.7.2.1 Tube Selection

A great variety of tube sizes, materials and configurations can be used in shell and tube heat exchangers. For this application, plain (unfinned) tubes of type 304L stainless steel with an outside diameter of 19.05 mm (0.75 in) and a nominal wall thickness of 1.65 mm (0.065 in) were selected.

Finned tubes were considered because the helium resistance is the largest contributor to the overall resistance for plain tubes. However, it was determined that the tube side fouling resistance limited the reduction in the size of the heat exchangers that could be achieved with finned tubes and, as a result, the cost difference between finned tubes and plain tubes would not be justified.

The tube material, type 304L stainless steel, was selected because it is commonly used in power plant condensers and is suitable for the water quality found at most sites with fresh water makeup.

The tube outside diameter selected was 19.05 mm (0.75 in). This is the smallest size generally used in steam condensers which are the closest analog in steam power plants to the precooler and intercoolers. Other power plant heat exchangers cooled by raw water generally use either 15.88 mm (0.625 in) or 19.05 mm (0.75 in) tubing. Using 19.05 mm (0.75 in) rather than 15.88 mm (0.625 in) tubes facilitates mechanical cleaning by reducing the number of tubes. Decreasing the tube diameter would have the effect of increasing the shell side pressure drop (assuming the total flow, tube velocity and tube pitch ratio are the same in both cases). Decreasing the tube diameter would also increase the cost of the heat exchanger because the number of tubes to be installed would increase. (It is assumed that the tube to tubesheet joints would be welded and expanded to minimize helium leakage.)

The tube wall thickness selected was 1.65 mm (0.065 in) in order to withstand the external helium pressure which is much higher than the water pressure. The shell side design pressure

Figure 5-30 IDC INTERCOOLERS 1 AND 2 DESIGN

Figure 5-31 IDC PRECOOLER DESIGN

has not been established, but may be based on a worst case transient such as loss of cooling water flow or generator trip without reactor trip and will probably be in the range of 4.8 MPa to 6.9 MPa (700 to 1000 psig).

5.7.2.2 Shell Configuration

Several shell configurations were examined, including counter flow, cross flow and split flow.

Counter flow arrangements with segmental type baffles had excessive pressure drops.

The split flow shell arrangement (with half the helium flow in counter flow and half the flow in cocurrent flow) had acceptable pressure drops, but the mean temperature difference correction factor ("F" factor) was very low. Designs with a low "F" factor are undesirable because a small deviation from the design operating conditions can result in a large decrease in the performance of the heat exchanger.

The cross flow arrangement had the lowest pressure drop of the configurations considered with an acceptable "F" factor. Therefore this configuration was selected.

The effects of non uniform distribution of the helium flow through the tube bundle were investigated. Experimental results published by the Heat Transfer Research Institute showed that an air cooled condenser with a non uniform air flow distribution caused by an upstream forced draft fan had performance within a few percent of that predicted assuming a uniform air flow distribution. A simplified spreadsheet model using the temperature conditions of the precooler indicated that a longitudinal flow rate varying linearly from 20% above average at the nozzles to 20% below average at the ends and midway between the nozzles reduced the "F" factor by approximately 5%. In the proposed design, the following features will minimize the non-uniformity of flow through the tube bundle. Each of the shells has two inlet and two outlet nozzles. Open segmental areas, approximately equal to the nozzle flow areas, are provided above and below the tube bundle for longitudinal flow. This creates a "U" flow path similar to that used in the recuperators. In addition, the precooler consists of two shells in parallel.

The water outlet temperature distribution was also evaluated. The tube row on the helium inlet side of the bundle is exposed to a uniformly hot gas while the tube row on the helium outlet side of the bundle is exposed to much colder gas. As a result, the outlet water temperature of a tube is a function of its location in the bundle, and the tubes most prone to scaling will be in the row on the helium inlet side of the bundle. Using the simplified spreadsheet model mentioned in the preceding paragraph, it was determined that a single tube pass design requires a higher water flow rate than a two tube pass design to achieve the same margin with respect to scaling in the hottest tubes. A two tube pass design gives a better "F" factor than a one tube pass design but also increases the shell diameter and therefore the cost of the coolers. However the cost of other cooling water components such as the cooling tower, pumps and piping will be lower for the two tube pass design. This will more than offset the increased cost of the coolers. Therefore, a two tube pass design was selected.

A similar effect could be achieved by using two cross passes on the shell side, but this arrangement would have a higher helium pressure drop and, in addition, the shell would have a very high temperature gradient in the vicinity of the baffle separating the inlet and outlet chambers.

The specific performance parameters selected for the precooler and intercoolers are summarized in Table 5-11.

A calculation was prepared to evaluate the effects of thermal expansion for the two tube pass, one shell cross pass design for the precooler. With this arrangement, the inlet side of the shell is exposed to 109°C (228°F) helium for its full length while the outlet side of the shell is exposed to 30°C (86°F) helium for its full length. This temperature difference will cause the shell to bend. If a shell expansion joint is not provided, the bending will be restrained somewhat by the tube bundle and tube sheets acting as a composite beam. The analysis indicated that the proposed design would probably not require a shell expansion joint. It also showed that increasing the segmental areas to improve flow distribution will also increase the thermal stress levels. The analysis did not address the effects of the nozzle openings which are approximately half the shell diameter. The thermal loads generated in the shell would create asymmetric loads on the nozzles. Further analysis would be required to develop a detailed design which would accommodate all these thermal effects.

5.7.3 Technical Issues/Limitations

Further work is needed to confirm that all plant transients have been evaluated to establish the required design temperatures and pressures.

5.7.4 Development Requirements

While a finite element analysis may be necessary to design the precooler and intercoolers for the Indirect Cycle, the design methods required are state of the art and no other equipment development would be required.

Table 5-11
PRECOOLER AND INTERCOOLER PERFORMANCE PARAMETERS

	Precooler	Intercooler 1	Intercooler 2
Duty, kJ/hr (Btu/hr)	368 x 10 ⁶ (349 x 10 ⁶)	197 x 10 ⁶ (187 x 10 ⁶)	288 x 10 ⁶ (273 x 10 ⁶)
LMTD, C (°F)	27.5 (49.5)	16.9 (30.4)	22.7 (40.9)
Effective Mean Temperature Difference, °C (°F)	25.6 (46.0)	15.3 (27.6)	21 (37.8)
Number of Shells	2	1	1
Heat Transfer Surface per Shell, m ² (ft ²)	2867 (30865)	4923 (52995)	4921 (52970)
Shell ID, mm (in)	2083 (82)	2184 (86)	2184 (86)
Segmental Height (under nozzles), mm (in)	508 (20)	508 (20)	508 (20)
Tube Type	Plain	Plain	Plain
Tube Length, m (ft)	12.2 (40)	12.2 (60)	12.2 (60)
Number of Tube Passes	2	2	2
Tube OD, mm (in)	19.05 (0.75)	19.05 (0.75)	19.05 (0.75)
Tube Pitch, mm (in)	23.81 (0.9375)	23.81 (0.9375)	23.81 (0.9375)
Number of Tubes per Shell	4052	4618	4618
Helium Pressure Drop (including inlet & outlet nozzles), kPa (psi)	3.93 (0.57)	8.20 (1.19)	5.58 (0.81)

5.8 TURBOMACHINERY AND GENERATOR (DC)

5.8.1 Key Design Requirements

- A major design goal is to provide a high aerodynamic performance - mechanically simple machine with reliable steady state and transient operating characteristics.
- Simple startup, shutdown and afterheat removal operation capabilities are highly desirable.
- For reasons of plant availability, low maintenance, and investment protection, the elimination of lubricant and/or sealant fluids (in contact with working helium) is considered to be very important.
- A high degree of overspeed protection is required.
 - Maximum of 10% overspeed is to be achieved by using active bypass flow control valves.
 - Active valves are to be backed up by passive pressure relief valves, limiting the maximum overspeed to 20%.
- Design of the turbomachine must provide for remote disassembly and maintenance.
- The goal for polytropic efficiencies is 90% for the compressors and 92% for the turbine.

Other thermodynamic parameters are as follows:

• Working fluid	Helium
• Compressor overall pressure ratio	2.82
• Turbine overall pressure ratio	2.61
• Helium mass flow rate, kg/s (1000 lb/h)	239 (1894)
• Turbine inlet temperature, °C (°F)	850 (1562)
• Compressor inlet temperature, °C (°F)	33 (91)
• Compressor inlet pressure, MPa (psia)	2.59 (375)
• Turbine inlet pressure, MPa (psia)	7.07 (1025)
• Turbine bypass flow fraction, %	0.50

5.8.2 Mechanical Design

Turbomachine

The turbomachinery arrangement is shown in Figure 5-33. The turbomachine design includes the following features:

- A single turbomachine rotor, solidly coupled to the synchronous type high speed electric generator.

Figure 5-32 Turbomachinery Arrangement

- Center high pressure type vertical rotor design with thrust loads balanced to minimize thrust bearings and shaft seals requirements.
- The entire rotor, including the generator, is supported by active magnetic bearings; two radial bearings and one double acting thrust bearing.
- Catcher bearings are of the ball bearing type with a minimum life of [20] drops.
- A lightweight and rigid type of rotor construction was chosen for reasons of higher critical speeds, lower required bearing stiffness and remote disassembly.
- Optimization of the compressor and turbine aerodynamic designs leads to a preference for operating speeds higher than 60 Hz.
- A single intercooler was selected over the possibility of two intercoolers. The additional inlet and outlet losses with two intercoolers would substantially offset the gains from lowering the overall compressor adiabatic head. In addition, packaging of the Power Conversion System within a single vessel would be further complicated.
- Following are the main turbomachine parameters:

- Gross power output, MWe	222
- RPM	5165
- Number of compressor intercoolers	1
- Number of LP compressor stages	12
- Number of HP compressor stages	16
- Number of turbine stages	9

Generator

- The generator is of the solid rotor, synchronous type. The compressor and turbine aerodynamic design dictates that the operating speed be higher than 60 Hz.
- The entire electric generator is submerged in a nonradioactive helium environment at about turbine discharge pressure level and is cooled via a dedicated helium to water heat exchanger. The submerged design allows for elimination of high pressure shaft seals from helium to ambient.
- Windage losses in the generator rotor cavity have been calculated on the basis of known test data, based on the rotor Reynolds number and the radial gap between rotor and stator. Depending on the generator detail design and gap selection, the calculated windage losses are in the range of 200 to 400 kW.
- Following are the main generator parameters:

- Generator power output (.9 power factor), MVA	~ 247
- Generator cavity pressure, MPa (psia)	2.65 (384)

- Rotor tip speed, m/s (ft/s)	213 (700)
- *Active rotor volume power density, kW/l	100
- *Active rotor diameter, mm (in.)	789 (31.06)
- *Active rotor length, mm (in.)	~ 4582 (180.4)
- *First "free-free" mode critical speed, rpm	~ 10,036

Frequency Convertor

- Because of nonsynchronous speed, a frequency convertor to 60 or 50 Hz is required.
- Technology for this convertor is based on high voltage direct current (HVDC) convertor technology. This type of convertor is currently being planned for a similar application at a 150 MW power level. Technology for the HVDC convertors is well established.
- The convertor efficiency in this size is being estimated to be no less than 98.5%. This translates into a loss of about 0.75% in net plant efficiency.
- The same convertor and turbomachinery configuration can be utilized for both the 60 and 50 Hz markets.
- The same convertor is utilized to motor the generator for plant start-up and for afterheat removal operations.

Magnetic Bearings Versus Oil Bearings

The premium placed on high reliability and reduced maintenance demands has resulted in the selection of magnetic bearings for the initial DC turbomachinery design. A key advantage of magnetic bearings over oil bearings is elimination of the potential for oil ingress into the working fluid. This consideration previously led to the selection of magnetic bearings for the SC main and shutdown cooling circulators. In addition to reducing the probability of forced outages, elimination of the oil from the primary system has other corollary advantages:

- Elimination of complicated shaft seals and expensive helium/oil separation systems and reduced demands on the helium purification system
- Elimination of environmentally negative effects and costs associated with disposing of contaminated oil, filters, etc.

Additional advantages of magnetic bearings that are of significance to the DC design include the following:

- Magnetic bearings provide additional capability for controlling the dynamic behavior of rotating machinery. By appropriate external control inputs, the characteristics of the

*Applies to the power generating part of the rotor only.

bearings may be modified as a function of rpm, shaft dynamic response, etc. Examples of how this capability has been applied in other applications are:

- Programming the stiffness and damping of the bearings, as a function of rpm to avoid passing through critical speeds (i.e., the critical speed is "moved" to a different rpm as the shaft rotational speed approaches it).
 - Reducing the transmission of vibration into the turbomachine support structure. A factor of 20 to 30 reduction, compared with oil bearings, has been recently demonstrated in a large fan application.
 - Eliminating or reducing shaft vibration by allowing the shaft to rotate about its center of inertia, rather than its geometric center.
- Magnetic bearings provide built in diagnostic capability to detect the effects of mechanical or aerodynamic instabilities that potentially could occur as a result of blade imbalance, circuit flow blockage or incipient mechanical failures.
 - Since there is no physical contact, there is no wear in magnetic bearings. This means that turbomachinery maintenance intervals should not be dictated by the bearings.

Active magnetic bearing technology has established an excellent track record in the gas pipeline and petrochemical industries, contributing greatly to reducing unscheduled shutdowns, process flow contamination, and maintenance costs. Programs are currently in place to develop magnetic bearings for gas turbine engines for use in military aviation and naval applications. For the DC GT-MHR, additional development will be required to substantiate the design of a large, high speed vertically oriented machine.

There are some potential disadvantages and additional development risks associated with the selection of magnetic bearings. The key disadvantage is that magnetic bearings require some type of "catcher bearings" to support the rotor in case of a complete malfunction of the magnetic bearings. These catcher bearings are now envisioned as antifriction bearings, utilizing dry lubricants sealed within the bearings. Other types, such as sleeve bearings, are also being considered. Many existing applications utilize an uninterruptible power supply (UPS), in conjunction with the active magnetic bearings, to bring the equipment down safely in case of a power failure.

A significant development effort will be required to qualify the magnetic bearings and associated catcher bearings for use in the DC turbomachinery. Key issues to be addressed relate to the size and vertical orientation of the DC turbomachine, and the catcher bearings, as noted above. For additional information on the development requirements associated with magnetic bearings for the DC GT-MHR, see Section 14.1.2.

5.8.3 Control System Design

The basic design goal of the DC control system is to take advantage of the inherent simplicity of the direct cycle design and use the simplest control strategy possible to satisfy load following requirements, meet plant thermal requirements, obtain the highest possible efficiency for part load operation and use inherently stable control features. The plant is required to be

capable of base load operation and, in addition to follow load at $\pm 5\%/min$, $\pm 0.5\%/min$, and to accommodate $\pm 10\%$ steps within the load range of 25% to 100% of plant rated electrical output.

A basic control concept to meet these requirements is depicted in Figure 5-34 and described as follows:

1. The generator is coupled to the electric grid through a solid-state variable frequency converter which converts the nonsynchronous generator output to 60 Hz. The variable frequency converter circuits are controlled so as to transfer all available generator output to the grid at a constant frequency.
2. The electric load demand signal controls the turbine control bypass valve position, which bypasses part of the compressor outlet flow into the turbine discharge. Varying the control bypass valve position quickly changes the flow through the reactor core and the turbine, thereby rapidly changing the turbine power and the generator electric power output. The flow through the control bypass valve varies from approximately 0.5% of total design helium flow at 100% load to approximately 15.0% of total design helium flow at 25% load. Turbine and compressor performance is strongly affected by control bypass flow; therefore, a 15% change in bypass flow can cause a 75% change in power output. If electrical load is lost, the control bypass valve and protection bypass valves are opened to control turbine speed and keep it below 115% of rated speed. These valves are sized such that either valve will prevent turbomachine or generator damage due to overspeed.
3. The electric load demand signal is also used as an index to set the turbine inlet temperature setpoint. The reactor negative temperature coefficient and the reactor control rods are used to control turbine inlet temperature to this setpoint. The range of turbine inlet temperature setpoints is from 850°C (1562°F) at 100% load to 794°C (1462°F) at 25% load. Small changes in turbine inlet temperature also affect turbine power output. Due to the large heat capacity of the massive graphite core, the turbine inlet (core outlet) temperature is varied slowly, at less than 2.8°C (5°F)/min.
4. The ultimate heat sink (cooling tower) water flow/pump speed is controlled to maintain the high pressure compressor inlet helium temperature at 33°C (91°F).

Startup, shutdown, and load following plant control and operation sequences and the results of a computer based simulation of plant response are described in Section 8.

The control approach, described above, was developed as a first step in designing a control scheme for the DC plant configuration. For this first step, turbine speed control and helium inventory control were not considered. Since the turbine/generator speed was kept constant throughout the normal load range, the basic control scheme, and the results shown in Section 8, envelope control schemes for both high speed, non-synchronous, and 3600 rpm, synchronous, turbomachines. Helium inventory control was deferred for consideration until after evaluation of the capabilities of the existing helium supply and purification equipment. As expected, and as indicated in the evaluations described in Section 8, this basic control scheme meets the control requirements, but has several disadvantages. In particular, at part load, a significant fraction

Figure 5-33 DC CONTROL SYSTEM

of the helium flow is bypassed from the high pressure side of the cycle at the compressor outlet to the low pressure side of the cycle at the recuperator inlet. Under these conditions, the precooler and intercooler see an increased steady state heat load, and plant net efficiency is significantly decreased.

Follow-on work is now underway to evaluate turbine speed control, in conjunction with the basic bypass control scheme, described above. Reducing turbine speed correspondingly reduces mass flow and, hence, power output. While an efficiency penalty results from non-optimum operation, that penalty would be small compared to that associated with bypass control. The HVDC variable frequency convertor has the capability of controlling turbine/generator speed throughout the load range in conjunction with helium bypass control. Initial evaluations of this improved control scheme show that adjustment of turbine/generator speed as a function of load may allow full closure of the helium bypass valve at part load, once steady state operation is reached at the lower power level. This results in a significant improvement in plant net efficiency and a significant reduction in precooler and intercooler steady state heat loads at part load conditions, while meeting all plant control requirements. After completion of the second phase of the controls development, the capabilities of the helium supply and purification system will be evaluated as a potential approach to inventory control for assisting in controlling longer term, steady state, part load operation. Core outlet temperature control will also be evaluated in conjunction with the above.

In summary, the final design of the DC plant control system will include appropriate elements of helium bypass control, core outlet temperature control, turbine/generator speed control and helium inventory control to achieve optimum plant steady state and transient operation throughout the load range and for startup and shutdown.

5.8.4 Safety and Investment Protection Design

Overspeed Protection/Mitigation

The turbomachine/generator is protected against overspeed by a set of active, fast acting bypass valves and, as a back-up, by a set of passive pressure relief valves.

Both the active and passive valves bypass the cold helium flow from the compressor discharge to the turbine exhaust. They are sized for the maximum helium flow required to prevent turbomachine overspeed for the case of instant load shedding to zero load with the electric generator and the reactor initially at full power.

The active valves will be designed to limit the full load shedding overspeed to less than approximately 115% speed. These are fast acting helium valves required to operate in the 250° to 150°C (300°F) range. Similar commercial valves are available to meet these requirements and no feasibility problems are foreseen for these valves.

The passive pressure relief valves are self-actuated by increased pressure differential between the compressor discharge and the turbine exhaust. Because of the constant helium inventory in the loop, the increase in the turbomachine speed causes a fairly fast drop in the turbine exhaust pressure and a fast rise in the compressor discharge pressure. The relief valves are being

evaluated as a potential passive means of limiting overspeed to less than approximately 120% speed. The turbomachine is designed to take a short time overspeed to 135% of the full power speed.

The passive valves are designed as pressure relief valves with a provision to have the capability to be externally actuated, i.e., exercised to open and closed positions, to allow for surveillance tests to demonstrate the valves' operational readiness. The active and passive valves will be redundant, each with full flow capability.

Missile Containment

The compressor and turbine housings are shielded by continuous missile shields positioned at the outside diameter of the housings containing the rotors. The missile shields are designed to contain fragments of the rotors, assuming failure at about 140% speed. Figure 5-33 depicts the missile shields as solid cylinders.

The missile shields are actually designed as continuously wrapped metal sheet, requiring missiles to penetrate each individual layer separately. By comparison, a solid ring forging could be the subject of large crack propagation type failure.

This type of multilayer containment ring was extensively tested as part of the Fort St. Vrain helium circulator program. It was proven by explosive tests to be able to contain the compressor wheel at runaway speed.

5.8.5 Technical Issues/Limitations

Based on evaluations to date, there are no known feasibility issues that would prevent the successful development of the DC concept. However, there are some items that will require more technical effort and engineering development than others.

Turbomachinery and Generator

Optimization of the aerodynamic design, for the size and cycle conditions of the point design, leads to a preference for operating speeds higher than 3600 rpm. The point design incorporates a high speed (5165 rpm), 247 MVA generator, that would require development. Based on already existing high speed, synchronous type multimegawatt motor applications, and discussions with large generator manufacturers, the indications are that such a machine can be successfully developed.

Electric generator structural and rotor dynamic (vibration) parameters tend to limit the generator rotor maximum rotational speed. At the same time, the compressor aerodynamic design dictates the number of compressor stages for a given rpm. Because of the low helium volumetric flow and generator rpm limitations, a larger number of stages is required for this turbomachine than would be required for a machine with higher speed and with the same cycle conditions. An acceptable compromise between the generator design (structural considerations and rotor dynamics) and turbomachinery design (aerodynamics) appears to be possible for the single rotor design. Further optimization of the rotational speed will be required. It is noted

that a larger than 450 MWt unit design would be easier, since the rotational speed for optimum aerodynamic design would be lower.

Magnetic Bearings

When operating at full speed, the weight of the vertical rotor is offset by the net of a thrust imbalance between the compressor thrust and the turbine thrust. However, at start-up, or when operating depressurized, when there is no aerodynamic thrust, the entire rotor weight is being supported by the upper magnetic thrust bearing. In case of malfunction of the magnetic bearings, the rotor relies on the catcher bearings for safe run-down. Sizing of the catcher bearing is not an issue, but cooling of the catcher bearing during acceleration and run-down may be. A dedicated water cooled heat exchanger, in direct contact with the bearing housing, as shown in Figure 5-33, has been incorporated into the design. There is a potential to use a backup magnetic bearing powered by an uninterruptible electric power supply. This would give further margin for the current [20]-drop life requirement for the antifriction bearing. There are also other possibilities that will need future investigation.

On a plus side to the above problem, the maximum axial load under normal operating conditions occurs only in the lower rpm range, which also generates less frictional heat.

Lateral supports for the magnetic bearings will have to maintain concentricity over the life of the plant. An engineering solution to this problem must be designed into the power conversion vessel.

Peripheral Seals

Peripheral seals are provided at the interfaces between the turbomachinery housings and the stationary parts within the power conversion vessel. Leak tightness of these seals is crucial to high plant efficiency. In addition, the seal design must provide for removal of the turbomachine, when required for maintenance. An effort has already been undertaken to design seals that will comply closely with potentially noncircular and nonconcentric mating surfaces. The conceptual design of a multistacked segmented seal, actuated by helium pressure differential, has already been developed. More work is needed in this area.

On the plus side to the above problem, the center high pressure type turbomachine design minimizes the number of high differential pressure seals. Most of the peripheral seals are at lower differential pressures.

5.8.6 Development Requirements

Aerodynamic Testing

No new aerodynamic subcomponent testing will be needed to confirm the compressor and turbine basic performance. Reynolds numbers are high and Mach numbers are low, so that existing cascade data and computational methods can produce a very predictable aerodynamic design.

It is planned that first aerodynamic testing will be accomplished during the integrated power conversion system test (see below).

Bearings/Rotor Dynamic Tests

Subcomponent testing of individual magnetic bearings will be required to verify the bearing's performance with the bearing size and bearing loads fully simulated.

Subcomponent testing of catcher bearings can be incorporated with the magnetic bearing tests, thus simulating the catcher bearings' performance under the condition of failed magnetic bearings.

Turbomachine and Generator Structural Tests

Burst tests of the highest stressed wheels should be performed at temperature in order to verify the overspeed margins.

All turbine and compressor wheels should be spun for the first unit and later on a production basis, to [140%] of nominal operating speed.

Fully assembled turbomachine rotors, both for the first unit and production units, will be full speed balanced in a vacuum. Generator rotors will be similarly spin balanced.

Structural members of the turbomachinery casings that contain instrumentation and magnetic bearing cables should be tested at full temperature to verify the adequacy of cooling means and thermal and electrical insulation.

Peripheral Seals Testing

Subcomponent testing of the various peripheral seals will be initiated early within the development effort. Key features to be verified include acceptable leakage rates during operation, tolerance to distortion and misalignment, ease of insertion and removal and the ability to renew the seals, as required during maintenance activities.

Integrated Power Conversion System (PCS) Test

A non-nuclear integrated Power Conversion System (PCS) test has been recommended by the GT-MHR Evaluation Team as a prerequisite to installation in the prototype GT-MHR. As an initial reference, the following basis has been suggested:

- Full scale.
- Design temperatures same as for the GT-MHR.
- 20% of rated power (approximately 90 MWt).
- 20% of nominal operating pressure to match volumetric flows.

The test article would consist of the PCS pressure vessel and all components contained therein. The externally mounted bypass control valves and their actuators are also included as

part of the test article. The components comprising the test article may subsequently be installed in the prototype GT-MHR plant, and their cost allocated to that plant.

The proposed test facility includes the systems, structures, and components necessary to:

- Provide heated helium at the required temperature, pressure, and flow conditions to the test article.
- Supply cooling water to the precooler and intercooler at required temperatures and flows.
- Provide adequate support for the PCS vessel and its internal components.
- Obtain data from PCS tests.
- Support the needs of the testing staff.

This test is envisioned as a full temperature, full speed test of all the components inside the PCS. The test would fulfill the following requirements:

- Verification of aerodynamic and thermal performance of all the components within the PCS. Some Reynolds number effects will have to be calculated.
- Verification of turbomachinery blades vibrational response. The magnitude of the blades' response can in this case be closely scaled up with helium density.
- Noise level generated within the PCS vessel can be similarly scaled up with density. The noise frequencies will remain the same.
- Centrifugal stresses will be duplicated and thermal stresses will be closely duplicated.
- Rotor dynamics/bearings operation will be verified for both normal operations and selected upset and failure modes (e.g., loss of load response, magnetic bearing failures).
- Operation of active and passive flow control valves will be evaluated under normal and abnormal operating conditions.
- Frequency convertor and electrical controls in the 30 MW range will be tested.
- Thermal distortions within the PCS will be monitored.
- The capability to remotely assemble and disassemble the PCS following operation at temperature will be verified.
- Operation of seals at component interfaces will be verified following temperature cycling and disassembly/reassembly.

5.9 TURBOMACHINERY AND GENERATOR (IDC)

5.9.1 Key Design Requirements

- A major design goal is to provide a high aerodynamic performance machine with reliable steady state and transient operating characteristics.
- A high degree of overspeed protection is required.
 - A maximum of 10% overspeed is to be achieved by using active bypass flow control valves.
 - Active valves are to be backed up by passive pressure relief valves, limiting the maximum overspeed to 20%.
- The goal for polytropic efficiencies is 90% for the compressors and 92% for the turbines.

Other thermodynamic parameters are as follows:

● Working fluid	Helium
● Compressor overall pressure ratio	3.08
● Turbine overall pressure ratio	2.82
● Helium mass flow rate, kg/s (1000 lb/h)	249 (1980)
● Turbine inlet temperature, °C (°F)	775 (1427)
● Compressor inlet temperature, °C (°F)	30 (86)
● Compressor inlet pressure, MPa (psia)	2.48 (360)
● Turbine inlet pressure, MPa (psia)	7.24 (1050)

5.9.2 Mechanical Design

Turbomachine

The turbomachine employs a two-shaft design: a low speed shaft and a high speed shaft. This is done to achieve maximum aerodynamic efficiency with a minimum number of total stages on the two shafts and to permit the use of a direct-drive 60 Hz generator. Employing two shafts also reduces the length of each shaft to about half of a single-shaft machine, thus reducing the bearing loads and the size of the bearings.

The low speed shaft contains the low pressure compressor and power turbine and drives the generator, all of which operate at 3600 RPM. By placing the low pressure compressor on this shaft, overspeed control is improved.

The high speed shaft is designed to run at about 10,000 RPM and contains the high pressure turbine and the medium pressure and high pressure compressors. After startup, this assembly seeks its own breakeven speed, depending on the turbine inlet temperature, since the power from the high pressure turbine is used only to drive the medium and high pressure compressors. By designing this rotor assembly for high speed, the total number of compressor stages required is reduced. A cross-section drawing of the turbomachine is shown in Figure 5-35.

Figure 5-34 TWIN-SHAFT TURBOMACHINE

The turbomachine and generator are oriented horizontally in a conventional manner. Standard oil-lubricated bearings are employed with two radial and one thrust bearing used on each rotor assembly and two radial bearings on the generator. A shaft seal is employed on the power turbine shaft and it is connected through a coupling to the generator shaft.

Two intercoolers are employed in the compression process to reduce the compressor power requirement, one after the low pressure compressor and one after the medium pressure compressor. The intercoolers contribute to increased cycle efficiency in conjunction with the recuperator, which is designed for 95% effectiveness.

The turbomachine main parameters are as follows:

- Turbine shaft power, MWe 218
- Number of compressor intercoolers 2
- Low-speed rotor
 - RPM 3600
 - Number of LP compressor stages 5
 - Number of LP turbine stages 9
- High-speed rotor
 - RPM 10,000
 - Number of MP compressor stages 6
 - Number of HP compressor stages 7
 - Number of HP turbine stages 3

Generator

The generator is a conventional design unit, similar to those used in combustion gas turbine or steam turbine driven power plants. It is of the solid rotor synchronous type. It is cooled by a closed pressurized hydrogen system which transfers the heat through hydrogen-to-water coolers. The generator employs a brushless exciter system which makes use of a set of rotating diodes on a shaft driven directly by the generator shaft. The exciter derives all of its power from the main generator shaft which makes it a very reliable system. The generator employs oil-lubricated bearings and hydrogen seals.

Following are the generator design parameters:

- KVA rating 253,000
- KW rating 215,000
- Power factor 0.85
- Voltage 13,800
- Efficiency, % 98.5

Magnetic Bearings as an Alternative

While oil-lubricated bearings appear to be quite acceptable for the indirect cycle design, magnetic bearings are a potentially attractive alternative. The advantages of magnetic bearings would be similar to those described for the direct cycle system. Among these would be elimination of the high pressure oil circulation and cooling system for the bearings, improved control of shaft dynamics and built in diagnostic monitoring. For a hydrogen cooled generator, sealing of the generator shaft would still be required. This could be accomplished either through conventional oil seals or dry gas seals. The latter have recently been employed in a large hydrogen supply compressor for a refinery cracking tower with good results.

5.9.3 Control System Design

As discussed in Section 2.2.2, the GT-MHR designs evaluated in this report are primarily configured for base load operation, and utilize a bypass control scheme for power maneuvering. While fast and effective, bypass control reduces the thermal efficiency of the power conversion cycle at reduced loads. If it is necessary to operate the plant for an extended period of time at reduced power, the secondary loop helium purification system could be utilized to reduce the helium inventory, thus allowing efficient operation of the plant at the lower power level. An alternative approach would be to shut down one or more modules of the plant and operate the remaining modules at full power. If the plant is to be operated routinely in a load following mode, or on a daily or weekly load cycle, additional equipment would be required to add and remove helium from the cycle as the power demand varies.

In the context of the above, power maneuvering requirements for the GT-MHR include the ability to respond to changes in load demand at $\pm 5\%/min$ and $\pm 10\%$ steps within the load range of 25% to 100% of plant rated electrical output. An approach to meeting these requirements is depicted in Figure 5-36.

During normal power operation with the generator synchronized to the utility grid, power conversion cycle response to load demand changes is accomplished by opening or closing flow bypass control valve V_1 , which bypasses flow from the high pressure duct between the recuperator outlet and the intermediate heat exchanger (IHX) inlet to the low pressure duct leading from the low-pressure turbine exhaust to the recuperator inlet. Adjusting valve V_1 effectively regulates the secondary loop helium flow through the IHX and the high-pressure and low-pressure turbines and, thus, power output. At full power, control valve V_1 would be closed, thereby avoiding efficiency losses. Thermal power from the reactor is coordinated with the varying load demand. This is accomplished by varying the speed of the main circulator to increase or decrease the primary helium flow through the reactor. The control rod positions are adjusted accordingly. Cooling water flow rates to the precooler and intercoolers are adjusted to maintain the helium temperatures at the compressor inlets at a constant 30°C.

Startup is initiated by operating the generator as a variable speed motor through a frequency converter to drive the low speed compressor. Valve V_2 is initially opened to bypass the low speed turbine. The helium flow established by the low pressure compressor induces rotation of the high speed shaft and establishes the pressure differentials required for startup. Heat input from the primary system is coordinated with the secondary loop startup, using the control rods

Figure 5-35 IDC CONTROL SYSTEM

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CONTROL SYSTEM FOR THE INDIRECT CYCLE MGR-GTI TWIN-SHAFT HELIUM TURBOMACHINE

and primary helium circulator. When sufficient power is being delivered from the primary system to sustain synchronous speed at zero load, valve V_2 is closed. The generator is then synchronized to the grid across the frequency convertor, the main breaker is closed and the frequency convertor shut down. Additional detail regarding startup is provided in Section 8.1.2.

5.9.4 Safety and Investment Protection Design

The turbine-generator is protected against overspeed damage in the event of a sudden full loss of electric load or generator shaft failure. Protection is provided by a set of large, redundant, active, fast-opening bypass valves (represented by V_2 and V_4), working in conjunction with the normal control bypass valve (V_1). These valves bypass helium flow around the high pressure and low pressure turbines, respectively (Figure 5-36). These protective bypass valves are sized to limit turbine overspeed to 110% of design speed.

In the event that the protective bypass valves are actuated, one result would be a significant increase in the helium temperature at the inlet to the low pressure side of the recuperator. An attemperation valve (V_3) is provided to moderate the helium temperatures under such conditions by bypassing cooler helium from the high pressure side of the cycle at the high pressure compressor outlet to the low pressure side of the cycle before the recuperator inlet.

5.9.5 Technical Issues/Limitations

The primary technical issue is the lack of experience with the design, manufacturing, and operation of large, closed cycle helium turbomachinery. Several smaller closed cycle turbines, with air as the working fluid, have been built and operated in Europe. Two medium size turbines, using helium as the working fluid, were also designed and operated there. (See Section 14.2 and Table 14-2 for additional information on prior closed cycle gas turbine experience.)

Existing computational methods and test data provide an adequate basis for the aerodynamic design of the compressor and turbine stages. The turbine inlet temperature of 775°C is lower than the operating metal temperatures of present-day combustion turbine blades, so that the strength of existing blade materials will be more than sufficient without the requirement for any nozzle or blade cooling.

The principal difference in the mechanical design of closed cycle turbines from open cycle machines is the higher gas pressure that must be contained. This, of course, requires heavier turbine casings and design of the power shaft seal for a larger pressure difference. Also, helium is more difficult to seal at high pressure than air.

5.9.6 Development Requirements

No need for aerodynamic subcomponent testing is anticipated. Confirmatory aerodynamic data would be obtained from the first turbomachine installed in a demonstration nuclear power system. Bearings and oil seals would be designed for higher pressure differences and confirmatory rig testing of prototype models would be required. As with production units, full speed balancing and overspeed tests would be accomplished in a vacuum vessel on the two turbine-compressor rotor assemblies prior to assembly within their respective casings.

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5.10 DIFFERENTIAL REQUIREMENTS/IMPACTS ON OTHER MAJOR SYSTEMS AND COMPONENTS RELATIVE TO THE GT-MHR (DC AND IDC)

In addition to the major systems and components specifically addressed in Sections 5.1 through 5.9, other direct cycle (DC) and indirect cycle (IDC) systems and components would exhibit varying degrees of differences from their SC-MHR (SC) counterparts. Some of these differences are summarized in the following.

5.10.1 Secondary Vessels and Piping (IDC)

The secondary vessels and piping found in the IDC power conversion system are analogous to vessels and piping found in the steam power conversion and feedwater supply systems of the SC. Both are located in the secondary power conversion loop and, as such, may be designed to conventional codes, such as the ASME Boiler and Pressure Vessel Code, Section VIII for unfired vessels.

In general, however, the IDC vessel and piping components are substantially larger than their steam cycle counterparts and, in the case of the high temperature sections of the power conversion cycle, see more demanding performance requirements. The higher temperature requirements for the IDC components reflect the higher maximum working temperatures of the gas turbine cycle, 775°C (1427°F), compared with the steam cycle [540°C (1005°F)]. The significantly larger sizes of the IDC components are due to the higher temperatures and lower pressures in the high temperature portion of the cycle, and the fact that the IDC operates completely in the gaseous phase, while the feedwater train of the SC operates in the liquid phase. The gas-turbine cycle also exhibits greater sensitivity to pressure losses, thus leading to larger piping components. Tables 5-12 and 5-13 identify the major vessels and piping components found in the secondary IDC loop. Additional details regarding the design approach and arrangement of these components may be found in Sections 4.1 and 6.2.3. Of the vessels,

Table 5-12 IDC SECONDARY VESSELS

VESSEL	NO.	TEMP. °C (°F)	PRESSURE MPa (psia)
Turbomachine Casing	1	775 (1427)	7.24 (1050)
Recuperator	2	442 (828)	2.55 (370)
Precooler	2	109 (228)	2.51 (364)
Intercooler 1	1	72 (162)	3.32 (481)
Intercooler 2	1	91 (196)	4.99 (725)

Table 5-13
IDC PIPING SECONDARY LOOP PIPING COMPONENTS

FROM	TO	NO.	SIZE mm (in)	TEMP °C (°F)	PRESSURE MPa (psia)
IHX Secondary Outlet	HP Turbine Inlet	1	1524 (60)	775 (1427)	7.25 (1052)
HP Turbine Outlet (Manifold + Ducts)	LP Recuperator Inlet	1 4	2134 (84) 1067 (42)	442 (828)	2.56 (372)
LP Recuperator Outlet	Precooler Inlet	4	762 (30)	109 (228)	2.51 (364)
Precooler Outlet (Ducts + Manifold)	LP Compressor Inlet	4 1	1067 (42) 1524 (60)	30 (86)	2.49 (361)
LP Compressor Outlet (Manifold + Ducts)	Intercooler 1	1 2	1524 (60) 1067 (42)	72 (162)	3.32 (481)
Intercooler 1 (Ducts + Manifold)	IP Compressor Inlet	2 1	1067 (42) 1524 (60)	30 (86)	3.30 (479)
IP Compressor Outlet (Manifold + Ducts)	Intercooler 2	1 2	1524 (60) 1067 (42)	91 (196)	4.99 (724)
Intercooler 2 (Ducts + Manifold)	HP Compressor Inlet	2 1	1067 (42) 1524 (60)	30 (86)	4.97 (721)
HP Compressor Outlet (Manifold + Ducts)	HP Recuperator Inlet	1 4	1016 (40) 406 (16)	91 (196)	7.53 (1092)
HP Recuperator Outlet (Ducts + Manifold)	IHX Inlet	4 1	406 (16) 1524 (60)	425 (797)	7.49 (1086)

the turbomachine casing and recuperator vessels are potentially exposed to high temperatures. The turbomachine casing will require cooling, while it is expected that the recuperator vessel will be internally insulated. None of the IDC vessels are considered beyond the state of the art.

The largest major piping component in the SC is the main steam line, at 356 mm (14 in). The main steam line operates at 17.3 MPa (2515 psia) and 540°C (1005°F). By contrast, the analogous high pressure helium supply duct from the IHX to the HP turbine inlet is 1524 mm (60 in) diameter and contains helium at 7.25 MPa (1052 psia) and 775°C (1427°F). For practical materials, this helium hot duct will need to be internally insulated. The lines from the LP turbine outlet to the LP recuperator inlet and from the HP recuperator outlet to the IHX inlet must also operate at high temperatures. Depending upon the selection of materials, internal insulation may or may not be required. Other IDC ducting components are large, but operate

at relatively mild temperatures. Again, none of these components are expected to be feasibility issues.

5.10.2 Shutdown Cooling System

The required capacity of the Shutdown Cooling System (SCS) is expected to be higher for the DC than for either the SC or IDC. The SC and IDC are expected to have similar SCS capacity needs. The reason for the higher SCS capacity requirement in the DC is that the turbomachinery, when shutdown, introduces an open primary coolant flow path in parallel with the core, effectively taking SCS flow away from the core. In both the SC and IDC, a shutoff valve is provided in conjunction with the circulator to limit core bypass flow.

Bypass flow is a particularly significant concern during simultaneous refueling and turbomachinery maintenance operations that require withdrawal of the turbomachine from the power conversion vessel. Under these conditions, the hot duct and the cold helium annular space in the cross vessel are completely open to each other at the power conversion system end. This results in a requirement to plug the hot duct under these conditions. Please refer to Section 11.2.5.4 for further discussion.

5.10.3 Refueling System

There are no major differential requirements for the DC or IDC Refueling System, relative to the SC. Since the nuclear reactor configuration, the fuel cycle, and the plant arrangement are essentially the same for the gas turbine and for the steam cycle plants, no significant differentiating features are anticipated in the Refueling System itself. However, the desire to simultaneously refuel and conduct turbine maintenance operations does lead to special shutdown cooling system and service equipment needs in the DC. These are discussed in Sections 5.10.2 and 11.2.5.4, respectively.

5.10.4 Helium Purification System

Relative to the Steam Cycle, both gas turbine concepts operate at approximately the same peak temperatures. However, the average fuel temperatures are higher in both gas turbine concepts than in the Steam Cycle: 820°C versus 630°C. Higher operating temperatures tend to increase circulating radioactivity in the primary coolant and the radionuclide load on the helium purification system (HPS). The average chemical contamination of the primary coolant is expected to be similar between the gas turbine and the steam cycle concepts. Since the size of the HPS is controlled by chemical cleanup requirements, no major variation in the design of this system is expected, unless there is a quite large increase in circulating radioactivity.

A feature unique to the IDC is a secondary helium coolant loop. The helium in this loop will require chemical purification capacity of its own, which would best be provided by a separate purification system in order to ensure separation between primary and secondary coolant. However, the load on this second purification system is anticipated to be lighter. Therefore, the system will probably be smaller than the primary coolant purification system.

A potential option for both the DC and IDC cycles is the use of the HPS to vary the pressure of the power conversion loop as a means of maintaining high cycle efficiency when operating at reduced power for extended periods. Sections 2.2.2 and 5.8.3 provide some additional information on this potential approach. If the HPS is to be used for this purpose, additional capacity would be required in the DC HPS or IDC secondary loop HPS, respectively.

5.10.5 Reactor Cavity Cooling System

The Reactor Cavity Cooling System (RCCS) is the third, and only safety-related system for removal of residual decay heat from the MHR core. In the steam cycle design, it consists of a system of ducts and panels which are installed in the reactor cavity, surrounding the reactor vessel. When the normal, forced circulation cooling systems are unavailable, heat from the fuel is transferred by conduction and radiation to the graphite core materials, then to the metallic core internals (principally the core barrel), and finally to the reactor pressure vessel. If helium pressure is maintained, convection will play a significant role in transferring heat from the fuel to the vessel. This type of transient is described as a pressurized conduction cooldown (PCC). If the helium pressure is lost, the contribution made by natural convection of the helium coolant will be much less. This type of event is referred to as a depressurized conduction cooldown (DCC).

Heat is transferred from the surface of the reactor vessel to the RCCS panels by surface-to-surface radiation and free convection of the air in the space between the vessel and the panels. Radiation tends to be the dominant heat transfer mechanism between the vessel and the panels. The panels and ducts are configured so that ambient air can enter the system and flow to the bottom of the reactor cavity, cooling the concrete structure as it does so. When air enters the riser section of the panels, it is heated and becomes less dense, inducing natural circulation. To improve buoyancy, the heated air is allowed to escape from the duct system at a high elevation. The system has no moving parts, requires no motive power, and cannot be affected by operator action. Numerous cross connections and common plena provide for redundant inlet and outlet flow paths for the individual tubes which make up the riser panels.

Calculations for the steam cycle design indicate that the system performs as desired. During normal operations, parasitic heat losses are on the order of 400 kWt. During accident conditions, when the vessel temperature is higher, the maximum heat transferred is on the order of 2 MWt. The transient performance of the MHR is such that decay heat energy is initially stored in the moderator, which rises in temperature. As the reactor materials heat up, more heat is transferred to the vessel and to the RCCS; meanwhile, the rate at which decay heat is generated decreases. The peak temperatures for the fuel, internals, and reactor vessel typically occur between 3 and 4 days after the start of the event, and after that peak, the rate at which heat is removed by the RCCS is greater than the rate at which heat is generated. During PCC events, the vessel and metallic internals are close (within 60°C) to their design temperature limits, while the fuel has considerable margin (on the order of 200°C). During DCC events, higher temperatures are allowed because hoop stresses are not present in the vessel. Less heat is transferred within the vessel by convection during this event, and while the peak vessel and internals temperatures are higher, the margin to the higher allowable vessel temperature is greater than for the PCC events. The DCC fuel temperatures are higher, and margins reduced, but still adequate.

A key parameter affecting RCCS performance is the reactor vessel emissivity. In the steam cycle design, these parameters are fixed by the choice of SA 533 Grade B carbon steel for the vessel material. This carbon steel is widely used in the light water reactor industry and has been fully characterized and codified. In either the indirect or direct Brayton cycle gas turbine concepts, the temperature of the gas returning to the core during normal operation is too high for this material. Therefore, a material with higher tensile strength at elevated temperatures must be used, and a carbon steel alloy with 9% chromium and 1% molybdenum has been selected. Section 5.4 describes these requirements and the selected vessel material. Little is known about the long term surface conditions for vessels made of SA 533 Gr B, and even less information is available for the 9Cr-1Mo-V material. The surface condition inside the vessel is as important as the condition outside. It is generally expected that the emissivity of the exterior surface, which is hot and exposed to air, will improve with age, gradually approaching an emissivity of 0.9, or higher. The 9Cr-1Mo-V material may improve more slowly than the ordinary carbon steel. This aspect could bear on the need to enhance the surface characteristics of the inside of the vessel, but has no effect on the design or performance of the RCCS.

The hottest point of the 9Cr-1Mo-V vessel will normally see temperatures near that of the core inlet (490°C). The parasitic heat loss to the RCCS will increase because of the higher vessel temperature. The temperatures during PCC and DCC events for the gas turbine concepts using 9Cr-1Mo-V vessels will also be higher than for the steam cycle. The rate at which heat is deposited in the core internals and the vessel will be the same as with the steam cycle; however, the sensible heat already present in the core internals and vessel will be higher, because of their higher operating temperatures. The peak temperatures reached by the fuel, vessel internals, and vessel will depend on the starting temperatures, rate of decay heat generation, and rate of heat rejection to the RCCS. Detailed calculations of these effects have not been made; however, it is expected that the margins (allowable temperature minus peak temperature) will increase for the vessel for both the IDC and DC designs.

The physical configuration of the RCCS system is limited primarily by interferences which exist with the reactor vessel support system. It is not expected that the core or vessel weight, in either of the gas turbine concepts, will be sufficiently different to affect the physical layout, relative to the steam cycle version. Therefore, the support scheme and the RCCS physical dimensions are expected to be unchanged.

5.10.6 Heat Rejection Train (Circulating Water System, etc.)

In the steam cycle design, essentially 100% of the core power is transferred to the main steam system. Approximately 42% of the energy transported by the main steam system is converted by the turbine-generator into electrical energy. The remainder, about 260 MWt, is rejected to the circulating water system and transferred to the atmosphere in wet, forced draft cooling towers. To get the maximum thermodynamic efficiency from the turbine cycle, the condenser and feedwater heating train are designed to extract this energy at the lowest possible temperature (around 43°C) and, as a result, the condenser and heat exchangers in the system are based on small temperature gradients across heat exchanger surfaces, and generally have large areas. The size of the cooling towers is based on the energy to be transferred to the atmosphere, the hot temperature of the circulating water, and the wet-bulb temperature of the atmosphere.

In the indirect gas turbine design, the core power is essentially all transferred to the secondary helium system. The intermediate heat exchanger must use a small temperature gradient to achieve this transfer, so the hot and cold points of the secondary helium system are necessarily a little bit cooler than the core outlet and core inlet temperatures. In the DC, the reactor coolant is used as the working fluid, so the hot and cold temperatures for the DC are the same as the core outlet and inlet temperatures. By employing highly effective recuperators, both cycles are able to achieve much higher efficiency than the steam cycle: about 49% for the DC and 47% for the IDC. This means that the total heat rejected by the cooling towers is about 228 MWt for the DC and about 239 MWt for the IDC. The circulating water system is used in both GT cycles to transfer energy from the precooler and intercooler to the cooling tower. In the case of the DC, it has been assumed that an isolation heat exchanger will be required to maintain tight control of water chemistry and to reduce the possibility of helium leaking directly to the atmosphere via the circulating water system. This strategy also reduces the possibility for introduction of foreign objects and materials into the precooler and intercooler. The aerodynamic and thermodynamic considerations for design of the precooler and intercooler result in rejection temperatures which are on the order of 125°C. Both the lower total heat rejection requirement and the higher rejection temperature act to reduce the required size of the cooling tower.

The circulating water requirements are also different from the steam cycle, with lower mass flow rates and higher dynamic head requirements. Because the IDC has slightly lower efficiency, but the DC requires an intermediate cooling water loop, the cooling towers for the two designs are expected to be about the same. Both will be reduced to about 70% of the size required for the steam cycle. The current DC and IDC designs use rectangular, wooden, mechanical draft cooling towers. The cooling tower approach temperature is 4.5°C (8°F) at 18°C (65°F) wet bulb temperature. The cooling tower range is 17°C (30°F).

The circulating water systems should be similarly reduced in overall size; however, the intermediate cooling loop for the DC will result in the need for additional water pumps and one or more large heat exchangers.

The other auxiliary cooling systems are affected by the requirements of the primary cycle design. In the steam cycle, the nuclear island cooling water system and the energy conversion area cooling water system are sized to serve auxiliary loads such as HVAC, circulator motor cooling, helium purification system operation, turbine lube system cooling, and generator cooling. For the IDC, many of the same loads are present. Both NI and ECA systems will continue to be required to serve HVAC and purification systems. The twin shaft IDC turbine-generator and auxiliaries will impose somewhat higher loads on the ECA cooling system. The IDC main circulator will be the largest load for the NI cooling water system, significantly larger than in the steam cycle. In the DC design, the ECA cooling water system will not be required, although a dedicated HVAC system will be needed for the operations center. The NI cooling water system will be larger because the circulator cooling load will be replaced with a generator space cooling load. Further design work will be needed in order to add precision to the needs for auxiliary cooling and support systems for the selected cycle design.

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SECTION 6

PLANT LAYOUT AND STRUCTURAL DESIGN

6.1 PLANT LAYOUT AND STRUCTURAL DESIGN - DIRECT CYCLE

6.1.1 Key Design Requirements

The layout and physical dimensions of the reactor system affect the diameter and depth of the reactor building. In evaluating the changes which may be required to accommodate the GT-MHR Direct Cycle, the following requirements are observed:

- Full embedment of the reactor building is preserved, to provide access to the top of the reactor for refueling, maintain an adequate RCCS air column length, and to enhance seismic performance.
- The silo portion of the building is kept in a cylindrical shape, to provide adequate resistance to soil and groundwater pressure without introducing excessive wall thickness. The 1-diameter spacing from silo to silo is preserved to assure excavation stability during construction.
- The design strategy to maximize modular, off-site fabrication, by arranging auxiliary and support systems in dedicated process units for each reactor and configuring them for over-the-road transportation is continued.
- The design features necessary to provide for reactor fuel handling and storage are continued from the steam cycle design.

6.1.2 Nuclear Island Layout

The layout of the nuclear island is dominated by the reactor building. For the direct cycle GT design, the length and diameter of the power conversion vessel control the dimensions of the silo portion of the building. Based on the 7.39 m (24 ft 3 in) outside diameter of the power conversion vessel flange, the silo diameter (measured at the center of the walls) is 24.94 m (81 ft 9.75 in) (see Figure 6-1). The cross vessel length and wall thickness between the reactor cavity and the power conversion vessel cavity remain unchanged from the steam cycle design. The position and thickness of the turbomachinery cavity walls are changed to accommodate the larger vessel diameter and to allow for the embedment of heavy support beams. It is expected that the greater weight of the power conversion vessel will require four support points instead of the two used for the steam generator.

The silo (Figure 6-2) must accommodate the length of the power conversion vessel below the cross vessel centerline, and must also accommodate the machinery used to service the shutdown cooling circulator and heat exchanger. The silo depth required to create space below the shutdown cooling system is greater than that required for the power conversion vessel, and

Figure 6-1 REACTOR BUILDING CROSS SECTION - DC

Figure 6-2 REACTOR BUILDING ELEVATION - DC

the silo depth required is 39.17 m (128 ft 6 in). Any significant increase in the length of the power conversion vessel will result in an increase in the silo depth.

The rectangular portions of the reactor building, which provide access to the silo and house reactor support systems, instruments and controls, retain the requirements and functions which existed for the steam cycle design. Therefore, as the silo diameter increases, the length and width of the rectangular portions of the building must also increase. In addition, the criteria to maintain a 1-diameter spacing between silos will result in the silo center to center distance increasing with the silo diameter. One effect of this will be to increase the span of the maintenance enclosure bridge crane. Some of the increases in building volume and cost which result from these changes can probably be reversed when there is an opportunity to optimize the layout further.

6.1.3 Overall Plant Layout

In the direct cycle design, the turbine building is eliminated and its functions performed by the reactor building. Most of the functions performed by other ECA structures persist, and are served in the following ways (see Figure 6-3):

- The operations center is retained and relocated to a position adjacent to the NI personnel services building. Future layout developments will be needed to create appropriate indoor connections between the two buildings.
- The double plant security boundary, which surrounds only the NI in the steam cycle design, is revised to continue to use part of the operations center as part of the security boundary. All persons entering the plant must pass through the single outer security boundary, and will be able to have access to parking, the switchyard, the cooling towers, and parts of the operations center. Pedestrians (plant personnel and visitors) needing to enter the high security zone will do so through the security control facilities in the operations center. Vehicular traffic needing to enter the security area will be inspected before passing through a double gate in the boundary.
- The standby power supply facility, fire protection facility, and auxiliary boiler/water treatment facility are retained and relocated near the reactor service building.
- The ECA warehouse and TG maintenance facility are combined with the NI warehouse facility and relocated to the northeast part of the site. On-site maintenance of the direct cycle turbomachinery equipment will be performed in the combined facility. To create access for transportation of the turbine-compressor unit in a shielded cask, the helium storage structure has been relocated. It is expected that a vehicle similar to the carrier for on-site spent fuel storage casks will be used to move the turbine-compressor units.

The remainder of features from the steam cycle nuclear island are unchanged from the steam cycle design. Figure 6-3 shows the on-site spent fuel storage area and other modular auxiliary support systems. The circulator power conditioning module has been replaced with a larger generator output power conditioning module (frequency convertor) that will convert the

STRUCTURES

- HS HELIUM STORAGE STRUCT.
- NW WAREHOUSE
- PS PERSONNEL SERVICES BLDG
- R1 REACTOR BLDG 1
- R2 REACTOR BLDG 2
- R3 REACTOR BLDG 3
- R4 REACTOR BLDG 4
- RS REACTOR SERVICE BLDG
- RW RADIOACTIVE WASTE MANAGEMENT BLDG
- SD REMOTE SHUTDOWN BLDG
- S INTERM. SPENT FUEL STORAGE LOCATION
- OC OPERATION CENTER
- AB AUXILIARY BOILER AREA/ WATER TREATMENT
- SPS STANDBY POWER SYSTEM
- T/M TURBO MACHINERY MAINTENANCE FACILITY
- FP FIRE PROTECTION

PREFABRICATED MODULES

- 1. H-COOLING WATER SYSTEM PUMP MOD.
- 2. SPENT FUEL COOLING SYSTEM
- 3. MINVAC SYSTEM
- 4. SC GRC. POWER CONDITIONING
- 5. GENERATOR POWER CONDITIONING
- 6. SCW SYSTEM
- 7. AIR-COOLED CHILLER
- 8. LIQUID NITROGEN RECONDENSER

OTHER

- 9. CASK WASHDOWN BAY
- 10. UNIT TRANSFORMER
- 11. VLP CONTAINMENT FILTERS
- 12. FIRE WATER STORAGE
- 13. DEMINERALIZED WATER STORAGE
- 14. SPS FUEL STORAGE

FIGURE 6-3 PLOT PLAN - DC

generator output to 50 or 60 Hz. Individual step-up transformers will be located near these modules and will be connected to the plant switchyard. Circulating water access to each reactor building will be by underground piping. The pumps and heat exchanger for the intermediate cooling loop will be located near the cooling towers.

6.1.4 Technical Issues/Limitations

At the present state of conceptual design development, there are no technical issues or design limitations related to the direct cycle plant layout or structural design which have been identified.

6.1.5 Development Requirements

At the present state of conceptual design development, there are no design data needs or other development requirements related to the direct cycle plant layout or structural design.

6.2 PLANT LAYOUT AND STRUCTURAL DESIGN - INDIRECT CYCLE

6.2.1 Key Design Requirements

The key design requirements for the indirect cycle arrangement are, in general, the same as for the direct cycle. With the indirect cycle gas turbine, as with the steam cycle, the opportunity exists to utilize the strategy of separated construction. This strategy has been used for the indirect cycle, but was modified to minimize the distance between the nuclear island and the energy conversion area. Based only on preliminary judgement, it has been assumed that the large diameter [1524 mm (60 in)] and, in the high temperature regions, expensive material used in the secondary helium lines will make it cost effective to keep their length as short as possible. With that objective in mind, the layout strategy for the indirect cycle has been modified. The other criteria which strongly affect the reactor building are as follows:

- Full embedment of the reactor building
- Cylindrical silo shape
- 1-diameter spacing from silo to silo
- Maximum use of modular, off-site fabrication
- Fuel handling and storage features from the steam cycle design.

6.2.2 Nuclear Island Layout

The layout of the nuclear island is dominated by the reactor building. For the indirect cycle GT design, the length and diameter of the intermediate heat exchanger vessel control the dimensions of the silo portion of the building. Based on the 7.32 m (24 ft) outside diameter of the IHX vessel flange, the silo diameter (measured at the center of the walls) is 24.5 m (80 ft 4 in) (see Figure 6-4). The cross vessel length and wall thickness between the reactor cavity and the IHX vessel cavity remain unchanged from the steam cycle design. The position and thickness of the IHX cavity walls is changed to accommodate the larger vessel diameter and to allow for the embedment of heavy support beams. It is expected that the greater weight of the IHX will require four support points instead of the two used for the steam generator.

LEVEL B4 PLAN AT EL.(-)178'-0"

Figure 6-4 REACTOR BUILDING CROSS-SECTION - IDC

The silo must accommodate the length of the IHX vessel below the cross vessel centerline, which is greater than the length of the steam generator. It must also accommodate access space for a machine to be used to enter the center manifold of the IHX for tube plugging (estimated length required is 7.62 m (25 ft) below IHX bottom flange). The silo depth required is 53.64 m (176 ft) (see Figure 6-5). A decision to keep the main circulator at the top of the IHX to facilitate circulator maintenance results in the secondary helium piping connecting to the IHX at the bottom of the silo. The long secondary helium lines which result may actually be superior to shorter lines which would be used if the circulator were at the bottom of the IHX, because of the need to accommodate stress in the pipes.

The rectangular portions of the reactor building, which provide access to the silo and house reactor support systems, instruments and controls, retain the requirements and functions which existed for the steam cycle design; therefore, as the silo diameter increases, the length and width of the rectangular portions of the building must also increase. In addition, the criteria to maintain 1-diameter spacing between silos will result in the silo center to center distance increasing with the silo diameter. As with the direct cycle, this will increase the span of the maintenance enclosure bridge crane. In addition, the secondary helium system includes isolation valves. The requirements for these valves have not been established, but they are conceptually similar to valves which were designed for high temperature process heat applications. The valves included in the layout (see Figure 6-6) are graphite lined and water cooled. In order to accommodate them, it has been necessary to eliminate one stairwell and to extend the rectangular portion of the building. Some of the increases in building volume and cost which result from these changes can probably be reversed when there is an opportunity to optimize the layout further.

6.2.3 Power Conversion Area Layout

The physical arrangement of major equipment and piping in the Power Conversion Area is shown in Figures 6-7 through 6-10 for a single unit. The Power Conversion Area is arranged so that one crane can be used for more than one unit. The plan area used for the Power Conversion Area for the Indirect Cycle is comparable to that required for the Steam Cycle. However, unlike the Steam Cycle, most of the equipment is located at grade level.

The turbine-compressor-generator set has been located on an elevated pedestal to allow piping and electrical connections to be made on the bottom of the equipment as well as the sides and top. The location and arrangement of these connections are not based on a specific turbomachinery design and may require some revisions when a detailed turbomachinery design is developed. Space has also been allocated at grade for supporting equipment for oil lubricated bearings and generator hydrogen cooling. Space has not been allocated for lay down of turbine-compressor-generator components during major maintenance. When a detailed turbomachinery design is developed, the space requirements for lay down should be evaluated.

The layout and arrangement of the Power Conversion Area helium piping is also shown on Figures 6-7 through 6-10. The pipe diameters are based on a helium velocity of approximately 30 m/s (100 ft/s). This velocity produces pressure drops which are below those assumed in the cycle point design described in Section 4. Optimization of piping cost versus the performance

6-9

KEY PLAN

Figure 6-8 GROUND FLOOR PLAN - IDC

3-3
(EW-203A & B)

4-4
(EW-203A & B)

Figure 6-9 ELEVATION, END VIEW - IDC

penalties associated with helium pressure drops would probably result in smaller piping diameters.

High pressure, high temperature helium is carried from the outlet of the intermediate heat exchanger to the gas turbine inlet in a 1524 mm (60 in) diameter pipe. It is assumed that this piping will be insulated on the inside to reduce the normal operating temperature of the pipe wall and therefore the rate of creep. To accommodate thermal expansion, a vertical loop was added to the layout in the area adjacent to the turbine building - reactor building wall. Consideration was given to a coaxial piping arrangement similar to one used on an earlier closed cycle helium plant. In that arrangement, the high pressure helium was carried from the intermediate heat exchanger in an insulated pipe which was located coaxially inside the pipe which carried the high pressure helium to the intermediate heat exchanger. However, for the IDC, this arrangement would have required an outer pipe about 2.13 m (7 ft) in diameter with a wall thickness of about 107 mm (6 in). Such a pipe would be extremely rigid. It was decided that there were no significant advantages to the coaxial arrangement to offset the problems anticipated in achieving a design to accommodate thermal expansion. Therefore the coaxial arrangement was not used.

The Gas Turbine exhausts into an 2134 mm (84 in) diameter pipe which divides into four parallel 1067 mm (42 in) pipes connecting to the low pressure inlet of each recuperator module. Parallel piping runs were used for the piping connecting to the low pressure inlets of the recuperator modules because the smaller diameter individual lines are more flexible than the single, 2134 mm (84 in) pipe. For similar reasons, the helium is carried from the low pressure outlets of the recuperator modules to the precooler inlet nozzles in four parallel 762 mm (30 in) pipes. Piping from the four 1067 mm (42 in) precooler outlet nozzles combine into a single 1524 mm (60 in) pipe which connects to the low pressure compressor inlet. The low pressure compressor discharges into a 1524 mm (60 in) pipe which splits into two 1067 mm (42 in) branches connecting to the first intercooler. Two 1067 mm (42 in) pipes from the first intercooler outlet nozzles combine into a common 1524 mm (60 in) pipe which returns to the intermediate pressure compressor inlet. The intermediate pressure compressor discharges into a 1524 mm (60 in) pipe which splits into two 1067 mm (42 in) branches connecting to the second intercooler. Two 1067 mm (42 in) pipes from the second intercooler outlet nozzles combine into a common 1524 mm (60 in) pipe which returns to the high pressure compressor inlet. The high pressure compressor discharges into a 1016 mm (40 in) diameter pipe which divides into four parallel 406 mm (16 in) pipes connecting to the high pressure inlet of each recuperator module. From the high pressure outlet of each recuperator module, 406 mm (16 in) pipes combine into a common 1524 mm (60 in) pipe to return the helium to the intermediate heat exchanger.

Space has been allocated for tube removal and replacement for the precoolers and intercoolers. Space has also been allocated for core removal and replacement for the recuperators.

The startup frequency converter is shown below the generator. The actual space required for this component has not been determined.

6.2.4 Overall Plant Layout

The overall plant layout is shown in Figure 6-11, and was based on the Steam Cycle overall plant layout. It differs in the following respects. The turbine building has been located adjacent to the reactor building to minimize the length of helium piping. Nuclear Island equipment located between the turbine building and reactor building in the Steam Cycle has been relocated to the opposite side of the reactor building for the Indirect Cycle. The boundaries of the security area were adjusted accordingly. Specifically, the double plant security boundary, which surrounds only the NI in the steam cycle design, has been relocated to use part of the reactor and turbine buildings as part of the security boundary. The water treatment and storage areas have been replaced by a secondary helium storage area. The circular cooling towers were changed to rectangular mechanical draft cooling towers.

6.2.5 Technical Issues/Limitations

At the present state of conceptual design development, there are no technical issues or design limitations related to the indirect cycle plant layout or structural design which have been identified. The increased depth of the silo increases the challenge in its construction, and the size and material of the intermediate helium piping may require elaborate support systems to accommodate stresses, but these issues are resolvable with current design and construction capabilities.

6.2.6 Development Requirements

At the present state of conceptual design development, there are no design data needs or other development requirements related to the indirect cycle plant layout or structural design.

Figure 6-11 IDC PLOT PLAN

SECTION 7

ENHANCED STEAM CYCLE OPTION

In order to assure a "level playing field" for comparisons with other plant options, an enhanced steam cycle (SC-MHR) design was evaluated and compared with the reference 450 MWt SC-MHR plant, as well as the two gas turbine (GT-MHR) designs. The power level for the enhanced SC was selected on the basis of providing margins for core related accidents that are equivalent to those provided in the GT-MHRs.

7.1 TECHNICAL DESCRIPTION OF CHANGES

The enhanced steam cycle design generates 475 MW thermal power. This increase in power from the current 450 MWt steam cycle design is accomplished by increasing the power density by 5.6%. No new materials are required; the existing 84 column core and reactor vessel configurations are utilized without change; and the nuclear island and energy conversion area arrangements are not expected to have major changes. While a point design remains to be developed, the core operating pressure and core inlet/outlet temperatures are also expected to remain unchanged. The size of the circulator, steam generator, and possibly the shutdown cooling system components will increase to reflect the increased power and/or its effects. The larger steam generator may also result in a small increase in the silo diameter.

7.2 TECHNICAL EVALUATION

The power level of 475 MWt was chosen so that limiting temperatures would be at or within allowable values without deviating from the existing 84 column core configuration and the current reactor vessel diameter. These temperatures and their corresponding limits are summarized in Table 7-1.

The limit for the reactor vessel has been conservatively assumed to be the value at which vessel stresses would exceed allowable values prescribed by the ASME code. The limit for the fuel is widely acknowledged to be a conservative value at which failure of the silicon carbide coating is not expected. The limit for the control rods has been conservatively set to assure that melting of the Alloy 800H rod structure will not occur.

The remainder of the components are well within their allowable limits for all design conditions.

As in the GT design, the fuel cycle availability factor would decrease as the core power is increased. Increasing the core power level will require more fuel, which will reduce the worth of the control rods and, therefore, decrease the shutdown margin. An increase in fuel also makes the temperature reactivity coefficient less negative, which reduces the change in reactivity from operating to shutdown conditions and, therefore, partially offsets the reduction in shutdown margin. Again, as in the GT designs, a trade study would be required to determine the optimum amount of fuel to maximize the product of power level and availability, while

TABLE 7-1 475 MWt LIMITING TEMPERATURES

<u>Event</u>	<u>Limiting Component</u>	<u>Calculated Temperature</u>	<u>Temperature Limit</u>
Pressurized conduction cooldown	Reactor vessel (midwall)	406°C (762°F)	427°C (800°F)
Depressurized conduction cooldown	Fuel	1586°C	1600°C
Depressurized conduction cooldown	Control rods (in outer reflector)	1177°C (2150°F)	1177°C (2150°F)

maintaining a negative temperature coefficient under all conditions. The potential inclusion of erbium or rhodium should be addressed in the trade study.

The increased power density will cause both peak and average fuel temperatures to increase by approximately 15°C. This is not expected to significantly affect fission product release.

The increased core power will result in an increase in core pressure drop and the required circulator power. Because the power density is increased by 5.6%, the flow through the core will also increase by 5.6%, the core pressure drop will increase by 11% [or 3.5 kPa (0.5 psi)], and the circulator power will increase by 18% (or approximately 1 MW). The effects on core dynamic behavior (e.g., resistance to fluctuations and vibrations) is expected to be small. The effect on the circulator is also manageable, since the increased power requirement remains well within circulator power limits. It is assumed that the steam generator will be resized for the increased power and flow, which will result in an increase to the steam generator vessel diameter and the silo diameter. The extent of the increase is estimated to be approximately 152 mm (6 in).

The neutron fluence to the reactor vessel will also increase in proportion to the increased power level, or 5.6%. This is not expected to affect the vessel availability or related plant startup time to any significant extent.

There are no incremental development requirements over the 450 MWt steam cycle design unless it is determined that erbium or rhodium is needed to maximize core availability. If that determination is made, the stability of the selected material and the process used to apply any necessary coating must be confirmed.

7.3 OVERALL ASSESSMENT

The 475 MWt is feasible, and maintains design margins comparable to the 450 MWt GT designs. Overall, while the core physics problems are similar, the lower temperatures of the

steam cycle relative to the GT, and the experience gained over the past years in steam cycle design, lead to the expectation that there would be fewer technical problems with the initial development of a steam cycle design than for a GT design. However, because of its greater efficiency, the operating cost improvement of a GT design would be greater than that of the enhanced steam cycle design.

SECTION 8

PLANT OPERATIONS ASSESSMENTS

The operations and control characteristics of the direct cycle (DC) and indirect cycle (IDC) gas turbine concepts were assessed and compared with the reference steam cycle (SC) concept. As expected, there are significant differences in the operation and control of these three concepts. However, all were found to be acceptable.

In most respects, the operational characteristics of the IDC reactor and primary coolant loop are similar to those of the SC. In both cases, the heat produced in the core is transferred, via a heat exchanger, to a secondary loop containing the power conversion cycle. During normal operations, helium flow through the core is provided by the main circulator, which may be controlled in a manner that is independent of, but coordinated with, the secondary cycle conditions. This capability provides a great deal of flexibility in operations, and core power output is normally varied simply by increasing or decreasing the helium flow through the core.

The principal differences between the IDC and SC are found in the secondary loop. In the case of the SC, the control strategy centers on coordination of the feedwater supply and the steam turbine admission valves. By contrast, the initial IDC control scheme uses turbine bypass valves to regulate the power output of the turbomachinery. In this respect, the control approach for the IDC secondary loop is similar to that of the DC, below. However, differences arise from their respective turbomachine designs. The IDC employs a twin shaft turbomachine with a synchronous generator, while the DC employs a single shaft turbomachine design and frequency conversion between the generator and grid.

The operational characteristics of the DC differ from both the SC and IDC, in that flow through the core cannot be controlled independently of flow through the turbomachine. Because of this significant difference, the principal emphasis of the plant operations assessment that follows has been on the DC. To facilitate this evaluation, General Atomics has developed a real time plant level simulation of the DC and its control characteristics, and has modeled the response of the plant to both normal power demand variations and the more demanding plant operational challenge of a loss of grid electrical load event.

The key operational issues associated with the IDC relate to plant startup, given the twin shaft turbomachinery configuration, and the response of the turbomachinery to the loss of grid electrical load event. These have been addressed by MIT and the results are also presented.

8.1 STARTUP AND RISE TO POWER

The requirement for startup and rise to power is the ability to transition from depressurized cold shutdown to a condition of readiness to supply power to the grid within a total time of 24 hr. The startup sequences for the DC and IDC in meeting this requirement are outlined below.

8.1.1 Direct Cycle

The direct cycle startup sequence differs significantly from the SC startup sequence, due to the location of the power conversion loop components in the primary system. The proposed sequence is as follows:

1. If vessel nil ductility transition (NDT) requirements allow, confirm full helium inventory. Otherwise, pressurize to maximum allowable pressure.
2. Confirm that control bypass is closed.
3. Power the electric generator with the variable frequency power supply and motor it to circulate helium.
4. Using heat input in the form of compressor work, warm the precooler and intercooler tube bundles and stabilize helium and water temperatures and flows.
5. Use the heat of compression to heat the vessel and pressurize to full inventory, while satisfying NDT requirements (i.e., temperature and pressure curves).
6. Bring the reactor critical.
7. Increase the reactor power and turbine inlet temperature, while opening the bypass valve until the turbomachine is at full speed with minimum power input from the variable frequency converter, and synchronize with the grid.
8. Adjust the reactor power (turbine inlet temperature) and control bypass position to produce electric power and reach the design point at 25% electric load.
9. Ramp to 100% electric load, as required by the dispatcher.

8.1.2 Indirect Cycle

Startup of the IDC reactor and primary loop heat transport system are similar to the SC case. Principal differences are found in the secondary loop with the startup of the twin shaft turbomachine. In starting up the secondary loop, the generator, in conjunction with a frequency convertor, is used as a variable speed motor to drive the low speed shaft. The bypass valve across the low speed turbine (this valve is identified as "V₂" in Figure 5-36) is opened. The resulting pressure difference serves to drive the turbine on the high speed shaft. Meanwhile heat is added through the IHX and, at some point, rotation of the high speed shaft becomes self sustaining. At that point, the bypass valve across the low speed turbine is closed and the generator is synchronized with the electrical grid across the frequency convertor. The main breakers are then closed and the frequency convertor is shut down. The rise to power is then continued.

MIT has estimated that, at full inventory, approximately 10-12 MW (say 15 MW with margin) would be required to achieve startup. The minimum value of 10 MW, above, is approximately twice the theoretical value of approximately 5 MW, if a starter motor were to be provided on the high speed shaft. The difference accounts for flow mismatches. Note that providing a starter motor on the high speed shaft was evaluated as a difficult problem, due to the power requirements, combined with the high speed of rotation. Either a new high speed motor design or a clutching arrangement would be required, as well as an additional high pressure shaft seal. It was also noted that a reduced power input could be used if the secondary loop helium inventory were reduced during startup. However, the capability for achieving plant restart without the need for reducing the secondary loop helium inventory is an offsetting factor that must be considered.

While the startup time was not estimated for the IDC, it is expected to be less than the steam cycle, for reasons similar to those cited for the DC. There is added complexity, relative to the DC, given the additional systems involved.

8.2 NORMAL POWER MANEUVERING

Normal power maneuvering was evaluated for the initial DC design, as a means of better understanding the more closely coupled interactions of the core and turbine flows. The evaluated design is described in Section 5.8.3, and is based on constant turbomachinery speed and the use of bypass control to regulate power output. The more advanced control system designs referred to in Section 5.8.3 (e.g., turbomachine variable speed control, inventory control) were not considered in this initial evaluation. The normal power maneuvers evaluated for the DC have already been assessed for the SC, and found to be acceptable. The IDC has not been specifically evaluated for these maneuvers; however, the primary system response of the IDC is expected to resemble that of the SC. The secondary response of the IDC would be similar to the DC results, below.

8.2.1 Power Range Operations and Control Response

Normal power maneuvering requirements are as follows:

1. Rapid load ramps of $\pm 5\%/min$ within the 25% to 100% electric load range.
2. Normal load ramps of $\pm 0.5\%/min$ within the 25% to 100% electric load range.
3. Step load changes of $\pm 10\%$ of electric load.

The initial operations assessments for the DC were limited to the bounding normal power range maneuvering requirements of $\pm 5\%/min$ and $\pm 10\%$ load steps.

The control design for normal power range maneuvering is described in Section 5.8.3 and the four major control loops are shown in Figure 5-34. Basically, load range control is effected by adjusting one or more control bypass valves that divert flow from the high pressure side of the cycle at the compressor outlet to the low pressure side of the cycle at the turbine outlet. This has the effect of reducing helium flow through the reactor and turbine. As the helium flow through the reactor is reduced, the naturally occurring feedback from the negative temperature coefficient of reactivity provides a corresponding reduction in core power output. Control rod

adjustments are coordinated with the negative temperature coefficient to maintain the desired turbine inlet (reactor outlet) helium temperature. The turbine is maintained at constant speed by the variable frequency converter, which controls the generator output so as to maintain a constant frequency conversion ratio between the generator and electrical grid. Helium inventory is also kept constant.

The availability of commercial valves to perform the control and protection bypass function was investigated. One vendor was contacted (Masoneilan) and they had valves capable of this function for helium service. Valves could be controlled with fast acting pneumatic or electrohydraulic actuators. The valves had up to a 1600°F capability. Masoneilan also could supply an "N" stamped valve if required.

The first case considered is a load ramp from 100% down to 25% electrical load at the required maximum rate of 5%/min. In this case, the dispatcher commands a load decrease and the control bypass is opened in a controlled fashion to bypass the compressor outlet flow into the turbine discharge. This unloads the turbine by decreasing flow through the core and turbine while relatively increasing the load on the compressor, thus decreasing the net power supplied to the generator. The variable frequency converter matches the electrical power passed to the grid by the decreasing generator output at constant speed. The command to lower load also causes the control system to initiate a feedforward control signal to drive control rods in, so as to adjust to the new reactor power level and to reduce turbine inlet (core outlet) helium temperature from 1562°F at 100% power to 1462°F at 25% electrical power. The core stored heat plays a significant role in the dynamics of the control response, and turbine inlet (core outlet) temperature varies only very slowly and reaches 1462°F at about 20 min. This is an important feature of the direct cycle, which inhibits rapid thermal transients and prevents thermal cycling of components.

The 100% to 25% load following transient at -5%/min was modeled on the Control Development System (CDS) simulator and results are shown in Figures 8-1 and 8-2. The requirements for the -5%/min load ramp are shown on Figure 8-1 as the generator power setpoint and the DC control response, generator power, matches nearly exactly. Figure 8-2 shows the turbine inlet temperature, as it is ramped to 1462°F, and the slight increase in core inlet temperature.

The second case considered is a load ramp from 25% up to 100% at a +5%/min ramp rate. In this case, the load dispatcher commands a load increase and the bypass valve is closed in a controlled fashion to pass more helium through the reactor core and the turbine, thus increasing the net power supplied to the generator. The variable frequency converter matches the increased generator power output to the grid to maintain constant generator speed. In this load ramp, the command to raise load causes the control system to initiate a feedforward control signal to drive control rods out to adjust to a new reactor power level and increases the turbine inlet (core outlet) helium temperature back to 1562°F.

The 25% to 100% load following transient at +5%/min was modeled on the CDS simulator and results are shown in Figures 8-3 and 8-4. The requirements for the +5%/min load ramp are shown on Figure 8-3 as the generator power setpoint and the GT-MHR control response, generator power, matches nearly exactly. Figure 8-4 shows the turbine inlet

Figure 8-1 100% TO 25%, -5%/MIN LOAD RAMP, POWER RESPONSE

Figure 8-2 100% TO 25%, -5%/MIN LOAD RAMP, TEMPERATURE RESPONSE

Figure 8-3 25% TO 100%, +5%/MIN LOAD RAMP, POWER RESPONSE

Figure 8-4 25% TO 100%, +5%/MIN LOAD RAMP, TEMPERATURE RESPONSE

temperature as it is ramped to 1562°F and the slight decrease in core inlet temperature. As was the case in the SC design, to maintain the +5%/min load ramp, the reactor power must be driven slightly over 100%. This is because of the slow temperature response of the core. Further studies will investigate the effect of limiting the reactor power to 102% by ramping up more quickly at first and then slowing as the reactor reaches full power, but still averaging 5%/min over the ramp.

The third and fourth cases considered are the -10% load step and +10% load step. In both of these cases, the control bypass valve is controlled to meet the step electric load demand. Reactor power would be adjusted to meet the new power level and to control turbine inlet (core outlet) temperature comparable to the $\pm 5\%/min$ load ramps. These transients were modeled on the CDS simulator and the results are shown in Figures 8-5 and 8-6. Figure 8-5 shows the -10% step load decrease and Figure 8-6 shows the +10% step load increase following the step change in generator power demand.

8.2.2 Accommodation of Duty Cycles

Duty cycle events specified within the normal power maneuvering envelope include the following:

1. Electric load rejection to house electric load.
2. Turbine trip with no reactor trip.

The first of these, electric load rejection to house electric load, was evaluated for both the DC and IDC. The initiating event is a loss of grid electrical load.

8.2.2.1. Direct Cycle

Loss of grid electrical load causes rapid turbine acceleration. On loss of electrical load in the DC, the control and protection bypass valves open and keep the turbomachine speed below 120% of design value. The reactor and turbine are kept on line and the turbine speed is stabilized at 100% speed. Assuming the generator and variable frequency converter are capable of supplying house electrical load only, the direct cycle appears to be capable of powering house load after loss of electrical grid load. This event was modeled with the CDS simulator and the results are shown in Figures 8-7 and 8-8. These results show that turbine overspeed is controlled and the speed is eventually stabilized at 100% design value.

In summary, the results of initial evaluations indicate that the DC is capable of meeting all requirements pertaining to power range load following, as well as the loss of electric grid load duty cycle transient.

8.2.2.2 Indirect Cycle

MITT evaluated the control response of the IDC for a similar loss of grid electric load transient. The analysis was conducted using the GTSim model, described in Reference 8-1. The results are depicted in Figure 8-9, and show that low speed shaft overspeed is limited to slightly less than 10%. The high speed shaft overspeed is considerably less, on the order of 4%.

Figure 8-5 -10% LOAD STEP, POWER RESPONSE

Figure 8-6 +10% LOAD STEP, POWER RESPONSE

Figure 8-7 LOSS OF ELECTRICAL GRID LOAD, POWER RESPONSE

Figure 8-8 LOSS OF ELECTRICAL GRID LOAD, TEMPERATURE RESPONSE

SIMULATION CASE: LOAD REJECTION WITH NORMAL PROTECTIVE CONTROL**450 MWth MHTGR TWIN-SHAFT HELIUM TURBOMACHINE** **Figure 8-9 IDC LOAD REJECTION WITH NORMAL PROTECTIVE CONTROL**

MIT also evaluated the limiting overspeed, which would result if both the control and protective bypass valves failed to open. The results are shown in Figure 8-10. It can be seen that the high speed shaft is self limited to an overspeed on the order of 15 %. No damage would be expected at this speed. In theory, the low speed shaft RPM would reach a maximum of 160% at about four seconds. Thereafter, its rotational speed would decline to a steady state value of approximately 55%. In this range of rotational speed, failure of the generator would be likely.

8.3 SHUTDOWN

The requirement for shutdown is the ability to shutdown from power conditions to cold depressurized conditions for refueling in 24 hr.

8.3.1 Direct Cycle

The direct cycle shutdown sequence is as follows:

1. Ramp load down to 25% electric load.
2. Reduce reactor power and turbine inlet temperature and begin closing the bypass valve to reach the minimum self-turbining condition.
3. Trip the reactor and let the turbine continue to self-turbine with stored core heat, gradually closing the bypass valve until it is fully closed.
4. When stored core heat can no longer power the turbine, the variable frequency converter will automatically begin powering the generator to motor the turbine and compressor to remove decay heat.
5. Decrease helium inventory for refueling.
6. Continue powering generator to provide shutdown cooling or switch to the shutdown cooling system.

The shutdown time is estimated to be less than 24 hr.

8.3.2 Indirect Cycle

In the case of the IDC, the power conversion loop could be used to remove heat in a manner similar to the DC. For this means of heat transport to be used, the main circulator in the primary system must also be operating. When the temperature differences across the turbines are insufficient to sustain self-turbining, the low speed shaft turbine bypass valve would be opened and the generator used as a motor to continue circulation of the helium. Alternatively, the shutdown cooling system could be employed at that time. Throughout the shutdown period, the primary loop operations would be similar to those used for the steam cycle.

SIMULATION CASE: LOAD REJECTION WITHOUT PROTECTIVE CONTROL
450 MWth MHTGR TWIN-SHAFT HELIUM TURBOMACHINE

Figure 8-10 IDC LOAD REJECTION WITHOUT PROTECTIVE CONTROL

8.4 REFUELING

The design of the fuel handling system is the same for either the direct cycle, indirect cycle or the steam cycle configuration. Refueling is accomplished with the reactor module shut down and depressurized to slightly subatmospheric conditions. Cooling is provided by either the shutdown cooling system or appropriate elements of the heat transport and/or power conversion systems, if available. Preparations for refueling include removing selected neutron control assemblies and mounting fuel handling equipment over the reactor vessel. Fuel blocks are removed one core sector at a time and transferred to local refueling storage facilities (LRSF) where new fuel is pre-stored. The spent fuel is left at the LRSFs. The new fuel and the irradiated fuel to be returned to the reactor are picked-up and moved to the reactor module. These operations are repeated for the remaining one-sixth core sectors.

The estimated refueling outage time, which includes start-up and shutdown operations, for either of the power conversion modes is approximately 15 days. This time includes a 72% efficiency factor to allow for unexpected minor equipment maintenance and work schedule problems.

The fuel handling system is designed to allow for maintenance operations throughout the plant with minimum interference. The system has its own dedicated equipment including its lifting and positioning cranes. The plant general maintenance crane is not used for any refueling operations and is available for other maintenance activities during refueling. When fuel handling equipment is moved over the plant operating floor, it passes beyond the reactor vessel side of the plant. The equipment does not move over power conversion components of any reactor module. Maximum maintenance activities are possible at the power conversion vessels during refueling, including access with the plant maintenance crane.

REFERENCES

- 8-1 Yan, X.L. and Lidsky, L.M., "Dynamic Analysis and Control System Design for an Advanced Nuclear Gas Turbine Power Plant", MITNPI-TR-035, MIT, Cambridge, MA, 1990

SECTION 9

PLANT SAFETY AND INVESTMENT RISK ASSESSMENTS

An assessment was made of the relative safety and investment risk characteristics of the steam cycle (SC) and gas turbine (GT) Modular Helium Reactor (MHR) designs. Key transients and failure modes were examined and, of these, three categories were judged to be of particular importance. First are events that require residual heat removal. These include pressurized and depressurized conduction cooldowns, which have been a major focus of prior SC safety assessments. A second category relates to turbomachinery failures, which are a safety issue in the case of the direct cycle (DC) but largely an investment risk issue in the SC and indirect cycle (IDC) designs. The third category involves heat exchanger failures, which have been particularly important in SC safety assessments.

The following subsections assess the relative safety and investment risk characteristics of the gas turbine and SC designs for the three event categories listed above. A summary qualitative assessment is provided in the final subsection.

9.1 CONDUCTION COOLDOWN

In the MHR, pressurized and depressurized conduction cooldowns (PCC and DCC) are important types of accidents. PCC and DCC events occur when the Heat Transport System (HTS) or Power Conversion System (PCS) and the Shutdown Cooling System (SCS) have failed to perform their respective functions of providing forced cooling of the core after reactor shutdown. Decay heat is then removed by the Reactor Cavity Cooling System (RCCS). The thermal transient behaviors of these two types of conduction cooldown accident conditions were analyzed for SC, IDC gas turbine, and DC gas turbine operating conditions. The analyses are based on nominal material properties and best estimate decay heat and power distributions. Decay heat and power distribution are based on SC uranium loadings. Both the decay heat and its distribution have been averaged over the second half of the fuel cycle to form a nominal estimate of the long term conditions in the fuel cycle. To assure a conservative estimate of component temperatures, this nominal decay heat has been increased by 12% to account for uncertainties in core power, decay heat, power distribution and material properties.

9.1.1 Options for Residual Heat Removal

Residual heat in the core, following a reactor shutdown, includes sensible heat stored in the graphite and heat continuing to be generated by radionuclide decay. The heat has to be removed in order to prevent damage to the fuel, the components inside the reactor vessel and the vessel itself.

The MHR has three independent options for removing the residual heat with the reactor module either pressurized or depressurized:

- Primary Cooling System
- Shutdown Cooling System
- Reactor Cavity Cooling System

The "primary cooling system" is the system that removes the heat from the core during normal power operation. In conjunction with related energy conversion and heat rejection systems, the primary cooling system provides the first option for removing residual heat when the plant is shut down. The design of the primary cooling system is quite different in the SC, DC and IDC concepts, respectively. The other two systems, the SCS and RCCS, employ similar designs in all of the three reactor module types.

The primary cooling system in the steam-cycle MHR is called the Heat Transport System (HTS). The HTS, which includes the main helium circulator and steam generator, is used to remove residual heat by circulating helium through the core. The heat is rejected from the primary system through the steam generator. The feedwater and main steam systems transport the heat from the steam generator and reject it to the atmosphere, via the main condenser, circulating water system and cooling tower.

The primary cooling system in the IDC is similar to that of the SC, and is also called the HTS. The main difference between these two HTSs is that the steam generator is replaced by the intermediate heat exchanger (IHX). The heat transferred within the IHX is rejected to the atmosphere via the secondary loop power conversion system, circulating water system and cooling tower. The power conversion system includes the turbomachinery, recuperator, precooler and intercoolers. During the early stages of residual heat removal, circulation in the secondary helium loop may be maintained without the need for external power, provided that the primary loop main circulator and circulating water system pumps remain operable. Secondary helium heated in the IHX expands through the turbines, thus driving the compressors and maintaining circulation. This mode of heat removal continues until the turbine inlet temperature drops below a limiting temperature that can sustain turbine operation. After that, the generator may be powered, acting as a motor to drive the turbomachinery for circulating the secondary coolant.

The DC gas-turbine reactor modules use a quite different primary cooling system. This system is called the power conversion system (PCS), and includes the turbine/compressor and generator unit, the recuperator, the precooler and the intercooler. Their functions for heat removal are similar to the IDC secondary loop. During the early stages of residual heat removal, hot helium from the core expands through the turbine, turning the compressor and, thus, maintaining the circulation of helium. Through this means, heat is taken from the core and rejected through the precooler and intercooler to an intermediate water cooling loop. From there it is rejected to the circulating water system and cooling tower. This mode of core cooling continues until the core outlet temperature drops below the limiting temperature that can sustain turbine operation. After that, the generator has to be powered as a motor to drive the turbomachinery for circulating the primary coolant and transporting the residual heat from the core to the precooler and intercooler.

In all of the designs, the SCS is used when the primary cooling system is not available. The residual heat is removed from the core by circulating helium with the shutdown cooling

circulator. This heat is transported to the shutdown cooling heat exchanger, where it is rejected to the atmosphere by the shutdown cooling water system. The RCCS, which is based on natural circulation and is always operating, is the third system capable of removing decay heat. No forced circulation of helium is considered in this case, and the residual heat is rejected to the atmosphere solely by passive means.

9.1.2 Pressurized Conduction Cooldown

In postulated events in which both the primary heat sink [PCS (DC) or HTS (IDC)] and the SCS are not available to provide forced cooling of the reactor, decay heat is then removed by thermal radiation and natural convection from the vessel to the RCCS. The results of PCC analyses for all four concepts are shown in Table 9-1.

The analysis of the PCC conditions shows that each concept has different component temperatures. There are two important reasons for this. The first reason, which applies to the two SC cases, is the 25 MW difference in core thermal power. The second reason, which applies to the gas turbine cases, is the higher normal operating temperatures. These higher temperatures translate into more stored heat in the core at the start of an accident. There is also an important differentiation between the two gas turbine cases: equilibrium pressure. The higher equilibrium pressure of the IDC will result in more primary coolant natural circulation flow out of the core. This higher flow will lead to lower fuel temperatures, but it will also increase the metallic component temperatures.

Table 9-1 shows that for PCC conditions, the peak fuel temperature of the gas turbine is higher than that of the SC. Also, because of the higher equilibrium pressure of the IDC, natural circulation through the core is greater and the peak fuel temperature is lower than in the case of the DC gas turbine. However, fuel temperatures remain below the 1600°C fuel design limit for the steam cycle and both gas turbine cycles. Figure 9-1 shows the peak fuel temperature as a function of time. The peak control rod temperature follows the same behavior as the fuel for the same reasons as the fuel. The control rod temperatures meet the limit for all of the concepts. The control rod PCC temperature limit has been established such that inspection or replacement would not be required prior to restart.

In the case of the core metallic components (excluding the vessel), the IDC shows the higher values. This primarily results from the relatively higher levels of natural circulation (in turn due to the higher pressure), which transports the heat out of the core and to these components. The peak core barrel temperatures occur at the top of the core barrel, near the vessel flange, where the upper plenum shroud rests on the core barrel. This part of the barrel is exposed to the upper plenum where hot natural circulation flow is coming up from the core. The current code case limit of 1400°F for the core barrel is exceeded for the IDC case. However, it is estimated by GA that a new code case at 1650°F is possible. By extending the code case to the higher limit, it will allow margin for future growth in core power and design.

On first glance, the peak vessel temperatures for the gas-turbine designs do not seem to be consistent with respect to the core barrel temperatures. However, the peak vessel temperature is not at the same elevation as the peak core barrel temperature. It occurs below the flange in the vessel sidewall. The thermal behavior of this area of the vessel is more

Table 9-1
COMPARISON OF PCC PEAK COMPONENT TEMPERATURES

	SC	SC	IDC-GT	DC-GT	Limits
ANALYSIS INITIAL CONDITIONS					
Equilibrium Pressure (psia)	1025	1025	1022	730	
Core Inlet Temperature (°F)	550	550	915	915	
Core Outlet Temperature (°F)	1300	1300	1562	1562	
Core Power (MW)	450	475	450	450	
MAXIMUM COMPONENT TEMPERATURES (conservative decay heat)					
Fuel (°C)	1088	1134	1278	1391	1600
Operating Control Rods (°F)	1451	1502	1651	1757	1875 ^(a)
Core Barrel (°F)	1307	1339	1453	1300	1400 ^(a)
Vessel Midwall (°F)	745	762	824	849	800/1000 ^(b)
Plenum Element (°F)	1559	1597	1740	1594	1875 ^(a)
Upper Plenum Shroud (°F)	1342	1373	1489	1336	1650 ^(a)

^(a)Values are investment risk limits.

^(b)The two limits are for SA533 and 9Cr-1Mo-V, respectively.

dependent on the peak graphite reflector temperatures than on direct heating resulting from the natural circulation flow going to the upper plenum. Because pressure and, thus, natural circulation are lower in the DC, the fuel and reflector temperatures are higher. Thus, the DC vessel temperature ends up being higher than that of the IDC vessel. All of the cases meet the vessel temperature requirements. The SC vessel limit is the code case allowable for SA533 under pressurized conditions. The gas turbine limit will require a code case. However, the 9Cr-1Mo-V creep failure time at 1000°F is on the order of 1×10^5 h. (See Section 5.4.3 for a more detailed discussion of 9Cr-1Mo-V.) Figure 9-2 shows the peak vessel temperature as a function of time.

9.1.3 Depressurized Conduction Cooldown

A DCC event is assumed to be initiated by a small primary coolant leak. The reactor trips automatically on low reactor coolant pressure and the operating control rods are inserted. The primary heat sink is assumed to fail immediately after the initiating event and the SCS fails to start on demand. The RCCS then removes decay heat from the vessel by natural convection and thermal radiation. The results of the depressurized condition analyses are shown in Table 9-2.

In contrast to the PCC results, the equilibrium pressure is the same in the DCC case for all designs (i.e., atmospheric pressure). Therefore, Table 9-2 shows the results for the two gas turbine designs as being the same. The table indicates that for DCC conditions, the peak fuel temperature remains below the 1600°C limit for both steam and gas turbine cycles. Figure 9-3

Figure 9-1 COMPARISON OF PCC PEAK FUEL TEMPERATURES

Figure 9-2 COMPARISON OF PCC PEAK VESSEL TEMPERATURES

Table 9-2
COMPARISON OF DCC PEAK COMPONENT TEMPERATURES

	SC	SC	IDC-GT	DC-GT	Limits
ANALYSIS INITIAL CONDITIONS					
Equilibrium Pressure (psia)	14.7	14.7	14.7	14.7	
Core Inlet Temperature (°F)	550	550	915	915	
Core Outlet Temperature (°F)	1300	1300	1562	1562	
Core Power (MW)	450	475	450	450	
MAXIMUM COMPONENT TEMPERATURES (conservative decay heat)					
Operating Fuel (°C)	1540	1586	1584	1584	1600
Control Rod (°F)	2125	2150	2150	2150	2150 ^(a)
Core Barrel (°F)	1274	1307	1322	1322	1400 ^(a)
Vessel Midwall (°F)	746	769	784	784	900/1000 ^(b)
Plenum Element (°F)	1385	1430	1515	1515	2150 ^(a)

^(a)Values are investment risk limits. Values for control rods and plenum elements represent points where components would need to be replaced before resuming operation.

^(b)The two limits are for SA533 and 9Cr-1Mo-V, respectively.

shows the DCC peak fuel temperatures as a function of time. Even though the peak fuel temperature rises faster for the gas turbine cases, it also falls off faster than the SC. The results of the four different cases indicate that, with respect to the base case SC, an increase of 25 MW in power is roughly comparable to the increased normal operation temperatures of the gas turbine. For the DCC analysis, the difference in component temperatures is due entirely to the higher power in the case of the 475 MW SC, and due to the higher normal operating temperatures for the gas turbine cases.

The operating control rods are located in the first row of outer, replaceable reflectors. However, because the outer ring of fuel does not have fuel in the corner positions of the hexagonal ring, control rods were placed there. Therefore, in a two-dimensional model (axial and radial) the azimuthal character of the control rod layout is lost. The peak temperatures of the control rods are rough estimates because of their locations. Table 9-2 indicates that these estimated control rod peak temperatures are right at the limit of acceptable temperatures.

For the SC, the PCC and DCC vessel temperatures are similar because the vessel must increase in temperature to radiate enough power to the RCCS to allow the core to cool down. However, for the gas turbine, the PCC vessel temperature is much higher than the DCC. The reason for this is that the normal operating inlet temperature of the gas turbine is high, with respect to the required RCCS power removal capability. Upon loss of forced flow, the vessel rapidly cools off in the DCC case (Figure 9-4), but the natural circulation flow that is present in the PCC case helps to prevent rapid cooling of the vessel. Therefore, because the vessel does

Figure 9-3 COMPARISON OF DCC PEAK FUEL TEMPERATURES

Figure 9-4 COMPARISON OF DCC PEAK VESSEL TEMPERATURES

not cool down as much during the PCC as during the DCC, the peak vessel midwall is higher for the PCC gas turbine case. Nonetheless, both the DCC and PCC cases result in peak midwall vessel temperatures less than the normal operating vessel temperature of 860°F.

9.1.4 GA/ORNL Benchmark Calculation

In conjunction with the conduction cooldown analyses discussed above, GA and ORNL have initiated methods comparisons to assure valid results. Benchmark calculations for the PCC and DCC accident conditions are being developed. Work is still in progress on the PCC benchmark, hence, only the DCC case will be discussed here. This comparison involves the TAC2D code, used by GA, and MORECA, a three-dimensional, MHR simulation developed by ORNL.

The DCC benchmark calculation is based on steady-state and accident conditions for the 450 MWt gas turbine design. GA has supplied material properties, nuclear power distributions, and physical modeling data relevant to the 450 MWt gas turbine to ORNL for use in MORECA. Because MORECA is a three-dimensional code, for the purposes of the benchmark calculation, the azimuthal power factor has been set to 1.0 to minimize the effects of three-dimensional versus two-dimensional calculations. The computer code POKE was used by GA to calculate the initial conditions of the active core for input to TAC2D. The steady state calculations show excellent agreement between POKE and MORECA on fuel temperatures and helium flow splits in the core. Hence, the differences in initial core and vessel temperatures between TAC2D and MORECA are insignificant.

Once agreement in the initial temperatures was found, a benchmark analysis of DCC conditions could be made. The results of DCC calculations for TAC2D and MORECA are shown in Table 9-3.

Table 9-3
TAC2D/MORECA DCC TEMPERATURE COMPARISON

	GA/TAC2D	ORNL/MORECA
Peak Fuel, °C	1584	1576
Core Barrel, °F	1322	1231
Vessel Midwall, °F	784	889

The peak fuel temperature calculated by TAC2D is slightly higher than the MORECA value. The peak fuel temperature calculations are in good agreement. For the case of the core barrel and vessel temperatures, there are some important modeling differences that are not easily

changed in both computer models. These differences primarily involve the RCCS. MORECA incorporates an RCCS model that is based on the original 350 MWt SC design. The RCCS has gone through significant changes since then and needs to be upgraded in the MORECA model before a consistent comparison of metallic component temperatures can be made. The incompatible RCCS models do not affect the comparison of fuel temperatures. The reason for this is that the fuel temperature is governed largely by the decay heat levels and the core heat capacity. Differences in the RCCS model will make only a few degrees difference in the peak fuel temperature.

9.2 MODES, PROBABILITY AND POTENTIAL CONSEQUENCES OF TURBOMACHINERY MECHANICAL FAILURES

A differentiating characteristic of the DC gas turbine design is the location of the turbomachinery in the primary system. While this design selection results in maximum plant efficiency and a significant reduction in overall plant complexity, the safety implications of the turbomachinery location must be addressed. For the IDC gas turbine and SC designs, failure of the turbomachinery primarily represents an investment risk issue.

As a means of obtaining additional insight, a review was made of the few known examples of major gas turbine failures and the probability of such failures was estimated using existing databases. Finally, the consequences of turbomachinery failures were addressed in some detail for the DC case.

9.2.1 Modes and Potential Consequences of Turbomachinery Mechanical Failures

Major failures in gas turbomachinery are relatively rare and there is limited documentation providing descriptions of such events (Reference 9-1). The following examples illustrate major failures that have been experienced in conventional gas turbine equipment and provide some insight into the design implications for the GT-MHR.

- A. One (1) unit experienced failure of the inlet guide vanes, which broke off and caused damage to the compressor and turbine sections. This outage required 10,288.2 downtime hours to repair the gas turbine.
 - The turbine and compressor damage occurred in less than one (1) minute, while the unit required approximately twenty (20) minutes to coast down to a full stop.
 - The incident occurred at 28,092 cumulative fired hours and 247 cumulative fired starts.
 - There were no missiles or projectiles that escaped through the casing. All the broken blades and vanes remained in the compressor, turbine, or exhaust casings. Small pieces were also found in the exhaust stack.
- B. One (1) aero-derivative unit experienced a failure of the tie bolt on the free turbine, which released two (2) free turbine disks from the unit. The disks then broke through the casing, causing additional damage within the plant enclosures.

- The failure of the tie bolt occurred at or near full speed (3600 rpm). The entire event, including the time for the disks to come to rest, lasted less than one (1) minute.
 - At the time of the failure, the gas turbine and generator began coasting down to a full stop (duration unknown).
 - The unit was retired after this failure; however, an outage requiring 1-2 weeks to remove and replace the engine and power turbine, and perform all other repairs was estimated by company personnel.
- C. One (1) unit experienced a turbine blade failure (broke at root), which caused major damage to the compressor, turbine, and exhaust components. This outage required 2381 hours of total downtime to repair the gas turbine.
- The incident occurred at 1562 cumulative fired hours and 215 cumulative fired starts.
 - There were no projectiles or missiles that exited the casings.
 - The failure was instantaneous, and the damage occurred in less than one (1) minute, while the unit required approximately thirty (30) minutes to coastdown to a complete stop.
- D. A failure of the stub shaft between the compressor and turbine sections on one (1) unit occurred, which resulted in the following damage:
- The compressor rotor dropped down, and began rubbing the blades on the casing. This caused the compressor to come to a grinding halt.
 - The turbine experienced an overspeed condition, and the turbine blades began to rub against the casing. The turbine blades began to break off and the casing fractured. This allowed fragments of both the blades and casing to come out of the turbine casing.
- E. One (1) unit experienced a failure due to a ruptured compressor wheel. The wheel ruptured into three (3) separate pieces within the gas turbine, while the unit was operating at full speed (3600 rpm). This rupture was caused by metallurgical defects within the compressor wheel. Two (2) pieces of the wheel exited the casing, and traveled one-quarter ($\frac{1}{4}$) of a mile from the unit. The other piece was contained within the compressor casing.
- F. One (1) gas turbine design experienced cracks of the turbine blades due to material problems on at least two (2) units. This caused extensive damage to the turbine, and exhaust sections, after the blades broke off from the rotor.

- G. One (1) gas turbine design had problems with the compressor blading, which would rub against the casing before shearing off from the rotor. These blades would then "corncob" the remaining compressor blades. This occurred on at least one (1) unit.

This historical investigation of the major failure events that have occurred on gas turbines shows that there have been very few major failures that have generated missiles or projectiles which have exited the turbine casings. Most comments received from either the gas turbine manufacturers or the gas turbine operators concluded that these incidents are rare occurrences. The gas turbine design philosophy has been to ensure that the turbine casings can contain all missiles or projectiles.

9.2.2 Probability of Turbomachinery Mechanical Failures

The probability of turbomachinery failures was estimated, using data applicable to ground power stations. The basis for categorizing failures, factors influencing the likelihood of failures, and the results of the current probabilistic evaluation are summarized in the following subsections.

9.2.2.1 Basis for Classifying Turbomachinery Failures

In accordance with various international standards (e.g., DIN, MIL-STD, ANSI/IEEE, ISO, API-STD, NERC), turbomachinery damage is classified into Malfunction Classes according to its extent:

- Major failure (MC1)
- Failure (MC2)
- Major malfunction (MC3)
- Outage, minor malfunction (MC4).

In the examples below, the types of failures usually associated with Classes MC1 and MC2 are illustrated. It should be noted that many of these are not applicable to the GT-MHR turbomachinery. Within Malfunction Class 1 (Major Failure) five critical malfunctions are:

- Overspeed (over 110%, due to failure of the protection system)
- Spontaneous rupture of the rotor
- Build-up of an explosive gas mixture in the machine before start-up (failure of the protection system against explosion)
- Flame extinguished (failure of the protection system, detection by means of a flame monitor)
- Leakage from fuel control and shut-off valves (failure of the protection system, automatic purging).

Within Malfunction Class 2, typical failures include:

- Journal or thrust bearing temperature too high (non-detected)
- Insufficient lube oil supply (non-detected)
- Bearing failure

- Blade failure
- Turbine entry temperature too high (non-detected)

9.2.2.2 Factors Influencing Turbomachine Failure Probability

Depending on the Malfunction Class, passive precautions and protective actions are implemented to keep failure occurrences as low as possible. Typical active protection means are:

- Alarms
- Protective load-shedding with controlled shutdown
- Protective load-shedding with trip at no-load condition
- Opening of the generator breaker
- Turbine trip.

Testing is important to safety. Reliability of the active protection actions must be addressed since it affects failure occurrences. Testing plays an important role to increase the functional readiness of protection equipment and thus reduce failure occurrences. Several types of testing are current practice:

- Testing during overhauls and repairs
- Testing before start-up and shutdown
- Continuous monitoring by comparison of parallel protection devices
- Automatic testing during operation
- Manual testing during operation.

Almost all current customer specifications call for a 2-out-of-3 redundancy in the turbomachinery protection system. This requirement states that "no single failure of a sensor or logic circuit component shall cause a spurious turbine trip or inhibit a genuine trip from occurring". This strives for higher availability of the unit. The 2-out-of-3 protection logic has been implemented for detection of:

- Overspeed
- Flame-out
- Lube oil pressure at the bearings
- Bearing temperature
- etc.

9.2.2.3 Turbomachinery Failure Probability

In order to determine representative failure probabilities for land based, power generation turbomachinery units, several data sets were reviewed.

One significant "damage" database of 316 units was reviewed on the basis that all important damage events were systematically reported. In this data set, the number of cumulative hours for each unit was significant enough to be interpreted on a statistical viewpoint. The total number of operating hours amounts to 11,200,000. The average number of operating hours per engine and per year is roughly 4500, which is typical for baseload units. The average

number of cumulative operating hours is 35,400 (average unit age: 7.9 years). The number of unit-years amounts to roughly 2500. The period of operation ranges from 1978 to 1991.

This database did not experience any rotor disintegrations, i.e.:

Rotor disintegration frequency = 0 Events/2500 unit-years

There is experience with "rapid deblading". For cases involving damage exceeding 3 vane or rotor rows, there are 16 events reported, i.e.:

Important turbine deblading frequency = 16 Events/2500 unit-years
i.e., 6×10^{-3} /unit-year

This figure should be treated with extreme caution. Indeed, in most of these events, the damage occurred due to foreign objects (external or from the combustion chamber). Furthermore, fuel-oil fired units (over 50% of the set) are more prone to damage than gas-fired units.

In the hot path section, low cycle fatigue (LCF) is one of the most difficult degradation phenomena to get under control. Uncertainties arising from aero-thermal boundary conditions, material properties, etc., lead to very high failure frequencies of parts situated in the hot gas path section. The total number of normal start-up and quick loading procedures affects drastically the failure occurrences of components (due to LCF). Most of the failures are reported within the first 24,000 operating hours for a gas-fired unit (or 16,000 hours for a fuel-oil fired unit). For purpose of illustration, 90% of all important failures in the combustion chamber occur within the first 24,000 operating hours for gas-fired units. If a part breaks down, it is therefore very likely that the turbine blading will be damaged.

This database did not experience any gas turbine overspeed failures. The main reason is that the occurrence of the following event: "nominal speed reaches 110% of the nominal value" depends on the succession of several undesirable events.

Thus: Turbine overspeed failures = 0 Events/2500 Unit-years

On behalf of ABB (Reference 9-2), a German insurance company and a consulting group independently carried out an analysis to determine failure occurrences of gas turbines. The following failure occurrences per engine and per unit-years were reported:

MC1	4.8×10^{-5} - 4.3×10^{-4}
MC2	3.4×10^{-3} - 3.1×10^{-2}

Most of the German utilities report failure data to a central administration (VGB). They reported the following figures (failure occurrence per engine per unit-years):

For GTs only:	MC1	2.25×10^{-4}
	MC2	1.61×10^{-2}

For Combined Cycles:	MC1	1.13×10^{-4}
	MC2	8.10×10^{-3}

Two other insurance companies reported similar figures for MC1 and MC2 failure occurrences.

As a comparison to the gas turbine failure probabilities, a comparable steam turbine data set was also reviewed. The steam turbine performance period was between 1982 and 1991, and included 500 steam turbines selected for their statistical data. The total number of unit-years amounts to 3100 and the average number of operating hours on a yearly basis is about 6500.

This database did not report any steam turbine (ST) rotor explosions (even though a crack was detected in 9 cases). Thus:

ST-Rotor disintegration frequency = 0 Events/3100 Unit-years

One important damage incident was reported due to overspeed (reason: the customer disconnected the overspeed alarms and sensors from the protection system!). The overspeed protection system failed once and led to a trip ("fail-safe"). It was also reported once that the overspeed protection system failed without causing a trip ("fail-dangerous"). Thus:

ST-Overspeed failure = 1 Event/3100 Unit-years

i.e., 3.2×10^{-4} /unit-year

Fifteen (15) cases of deblading were reported. Thus:

ST-deblading frequency = 15 Events/3100 unit-years

i.e., 4.8×10^{-3} /unit-year

The probability of major failures can also be inferred from the eight (8) relevant examples cited in Section 9.2.1. For this purpose, the estimate is based on the following total period hours for gas turbines:

Gas Turbines(ORAP):	5000 unit-years = 4.38×10^7 hours
Gas Turbines(NERC):	
1986 - 1990	4535 unit-years = 3.97×10^7 hours
1982 - 1985	<u>3309 unit-years = 2.90×10^7 hours</u>
Totals:	12844 unit-years = 11.25×10^7 hours

Based on the eight (8) failures that have been identified in Section 9.2.1 (not including combustion and fuel system failures), the Major Failure Probability can be estimated as:

Major Failure Probability	= $8/11.25 \times 10^7$	= 7.1×10^{-8} per hour
	= $8/12844$	= 6.2×10^{-4} per unit-year

From the available outage duration information for the major failures identified, the outage durations range from 2381 downtime hours to at least 10,288 downtime hours. The actual outage duration will vary based on the extent of the damage, and the availability of gas turbine parts.

Note that using the sum of the ORAP and NERC unit-years data results in a conservative estimate, since those databases reflect only a portion of the total period hours for gas turbines worldwide. However, such an estimate provides at least some basis for understanding the probability of a major gas turbine failure. Recognize that the assessment provides estimates based only on available data, and should be utilized accordingly (Reference 9-3).

9.2.3 DC Turbine Transient Studies

The data that has been reviewed for combustion turbines in the above subsections indicates that the probability of deblading failures is in the range of 10^{-3} per year. The likelihood of turbine deblading events in the DC gas turbine is expected to be much lower in frequency, however, due to the elimination of combustion system related failures and the plants relatively benign turbine operating environment (i.e., temperatures, chemistry, transients). However, turbine debladings in the DC, as compared to the IDC, are expected to be more challenging to the safety functions required to protect the fuel coatings. In particular, debladings in the primary coolant flow path may lead to large, rapid pressure changes in the system. Such a transient may potentially impact the safety functions of removing core heat and controlling core heat generation that are required to protect the fuel.

A computer model was developed to assess the rapid gas dynamics of the DC during accident conditions. A brief description of this model, steady state predictions of the thermal fluid parameters, comparison with point design calculations and preliminary results of two unprotected transients have been completed.

A schematic of the DC model is shown in Figure 9-5. A six-stage turbine component, two-stage compressor component, and generator are coupled to a single shaft. During steady-state operation, the generator is connected to the external grid and can be decoupled from the grid during accident conditions. The recuperator is modeled as a gas-to-gas counter flow heat exchanger with 30 node physical resolution to obtain 95% effectiveness during 100% power operating conditions. The precooler and intercooler are modeled as constant temperature heat sinks. A five-node reactor core is simulated using a center-coolant hole model, a typical heat removal system model that has been extensively used in SC reactor core simulations. A bypass valve activated by operational control systems during anticipated operational transients, and a protection bypass valve activated by plant protection systems, are also included in the model. In addition, a number of hydrodynamic volumes modeling flow paths such as core barrel channels, core upper and lower plenums, inlet and outlet volutes of the turbomachinery components, hot and cold ducts, and free volumes surrounding the reactor core are simulated.

This model has been used to obtain a baseline steady-state solution at 100% power operating conditions. Comparison of the steady-state predictions with the point design calculations indicate good agreement. Comparisons show that key thermal parameters such as the reactor outlet temperature, maximum pressure in the closed-loop system, pressure ratios

Figure 9-5 DIRECT CYCLE RELAP5/MOD3 MODEL

across turbomachinery components and most importantly, the overall plant thermal efficiency of 50% were predicted to be in agreement with point design calculations within 1%.

Two unprotected transient events involving turbomachinery failures have been simulated using the RELAP5 model. The events chosen are based on prior GT-HTGR experience which identified: (1) spurious external load trip, and (2) turbine deblading as being of primary importance. These transients are simulated without plant protection system actions in order to understand characteristics of various components and thereby determine control and protection needs during such events.

The load trip transient is initiated from 100% power operating conditions. At the transient initiation time, the generator is instantaneously disconnected from the external grid with no control or protective system actions. Major concerns during this transient are (1) rapid overcooling of reactor core, (2) high inertial loads due to sudden acceleration of the fluid, and (3) high pressure loads across various components.

At the initiation of the transient, a total of 420 MWt is being absorbed by the turbine element, which is utilized to drive the two compressor stages and generator. Removal of the generator load causes a sudden acceleration of the shaft. Figure 9-6 shows the time variation of the shaft speed as the shaft accelerates from the initial speed of 5165 rpm to a maximum value of 7700 rpm at 35 s from the initiation of the transient. Figure 9-6 illustrates the existence of three distinct time regimes: (1) initial overspeed dictated by the polar moment of inertia of the turbomachinery components (1 to 10 s), (2) intermediate turn-over time period during which off-normal turbomachinery performance plays a major role (10 to 30 s), and (3) equilibration regime during which thermal-fluid dynamic processes equilibrate to a quasi-steady state condition. Transient predictions indicate sufficient time is available for control/protective action, such as flow bypass, in order to limit the shaft overspeed. It is noted that turbomachinery component off-design performance characteristics exert a large influence on the prediction of the transient results.

The turbine deblading transient is initiated from 100% power operating conditions. Key safety concerns during this transient are: (1) core support integrity and (2) core flow reversals. Failure of the core support could challenge the safety functions of heat removal and control of core heat generation. Significant flow reversals could eject control rods and challenge control of heat generation. Loss of either of these functions could result in increased fuel temperatures, thus challenging the retention of fission products within the coated fuel particle.

During this transient, turbine deblading is assumed to result in loss of all blades except for the first stage stator guide vanes. As a result, the turbine acts as a flow restrictor with a minimum flow area of about $.6 \text{ m}^2$ (6 ft^2) that exists in the stator guide vanes. It is further assumed that the loss of all blades occurs in a time period of 0.05 s. The present turbine deblading model assumptions are deduced from prior GT-HTGR turbine deblading transient analysis experience (Ref. 9-4) and further investigation of the model assumptions needs to be performed.

Figure 9-7 shows the pressure response in the upper and lower core plenums during the turbine deblading transient. Model predictions show a rapid collapse of the pressure differential

Figure 9-6 SHAFT SPEED DURING DC LOAD TRIP TRANSIENT

Figure 9-7
PRESSURE-TIME VARIATION DURING DC TURBINE DEBLADING TRANSIENT

across high pressure and low pressure regions of the closed-loop system, and the pressure in the closed-loop system equilibrates in a time period of 2 s from the initiation of the transient. Model predictions indicate an initial core flow increase; however, flow reversal in the core does not occur.

Pressure differential across the core is a measure of the pressure forces acting on the core and the core supports. It also indicates any flow reversals that can occur during the transient. The initial pressure differential across the core during steady state operating conditions is 51 kPa (0.5 atm) and the pressure differential across the core increases to a maximum value of 304 kPa (3 atm) at 2 s from the initiation of the transient. Structural analysis of the reactor core supports from previous studies indicate that core support integrity is maintained up to a core pressure differential of 1 MPa (10 atm).

9.3 CONSEQUENCES OF HEAT EXCHANGER FAILURES

The consequences of heat exchanger failures were qualitatively evaluated for the gas turbine and SC designs based on the sources of water ingress available, the frequency of moisture ingress and the equivalent forced outage hours (EFOH). The major safety concerns regarding moisture ingress at full power are release of fission products from a small fraction of initially-failed fuel particles due to hydrolysis of the fuel and release of small amounts of fission products trapped in the core graphite due to oxidation of the graphite. Graphite oxidation and fuel hydrolysis are less important when the fuel temperatures are low during refueling. At shutdown conditions, the primary concern is an increase in core reactivity from enhanced neutron moderation due to the presence of moisture. Reactivity is not a concern during hot operating conditions, since adequate shutdown margin exists to control the smaller amount of water ingress corresponding to the reduced density of water at high temperature.

The sources of moisture ingress available in each design for shutdown conditions are given below. In the SC, potential sources of water include the steam generator (SG), shutdown cooling system heat exchanger (SCHE), main circulator motor cooling heat exchanger (MCMC) and the shutdown circulator motor cooling heat exchanger (SCMC). During power operation, only the steam generator is a potential water ingress source. In all the other sources, the water pressure is less than the primary coolant pressure.

The moisture ingress sources in the DC are the precooler, intercooler, SCHE, SCMC, and the generator cooling heat exchanger (GC). In the IDC, the water sources are the SCHE, MCMC, and SCMC. During full power operation, none of these are significant sources because the primary coolant pressure is higher than that of the water systems.

The amount of water available in the precooler and intercooler of the DC, given the current design incorporating an isolation heat exchanger, is comparable to the water available in the SG. If the precooler heat sink in the DC were not isolated, the water available would be approximately a factor of two times the SG amount. In either case, isolation of the precooler and intercooler would be relatively easy, since the cooling water source is outside of the primary vessel. However, the amount of water is less important than the mass transport rate of water to the core. For the DC and SC designs, at shutdown conditions, the water transport rates are judged to be similar since temperatures are also similar.

Table 9-4
FREQUENCY OF MOISTURE INGRESS AT SHUTDOWN CONDITIONS

450 and 475 SC		450 DC		450 IDC	
Component	Leak Frequency (/Plant Year)	Component	Leak Frequency (/Plant Year)	Component	Leak Frequency (/Plant Year)
SG	4.0E-1	Intercooler and precooler	1.4E-1		
SCHE	7.0E-2	SCHE	7.0E-2	SCHE	7.0E-2
MCMC	5.4E-3	GC	5.4E-3	MCMC	5.4E-3
SCMC	7.1E-5	SCMC	7.1E-5	SCMC	7.1E-5
Total	4.8E-1	Total	2.2E-1	Total	7.5E-2

For the SC and gas turbine designs, the frequencies of moisture ingress at shutdown conditions are provided in Table 9-4. The SG, intercooler and precooler leakage rates are dominant. The SG tube leak rate, 0.4 per plant year, is assumed the same as in the 350 MWt SC-MHR probabilistic risk assessment (PRA) and is due to bimetallic weld failure, similar weld failure, erosion, corrosion, fretting and mechanical wear. The intercooler and precooler tube leak frequency, 0.14 per plant year, is based on a correlation that predicts the frequency of weld failures. The generator cooler heat exchanger failure frequency is assumed the same as for the main circulator motor cooler heat exchanger. The total moisture ingress frequencies at shutdown conditions of the three concepts, SC, DC, and IDC are 0.48, 0.22, and 0.075, respectively.

A comparison of the EFOH for each heat exchanger is given in Table 9-5, which is a subset of the more detailed plant forced outage assessment presented in Section 10.2.2. The dominant heat exchanger forced outage contributors are the SG, intercooler and precooler, and the IHX. The IHX leak rate due to similar weld failure, mechanical damage, fretting, and wear failures is 0.2 per plant year. This is a factor of two smaller than the steam generator tube leak rate. However, the IHX has twice as many tubes as the SG and so the mean time to repair of the IHX tube is assumed twice as long as the SG tube. The resultant IHX EFOH is the same as that of the SG, 57 h per year. An assumption inherent in this analysis is that IHX leak contamination of the secondary side components has a negligible effect on future scheduled or unscheduled outages.

For the DC, the precooler and intercooler leakage frequency is a factor of 2.8 less than for the SG. The outage time is dominated by the time it takes to detect and plug a leaking precooler or intercooler tube, which, because it is a leak into the secondary side and because of the larger number of tubes, is assumed to be 1.5 times that of a SG tube detection and plugging.

Table 9-5
HEAT EXCHANGER EQUIVALENT FORCED OUTAGE HOURS (EFOH)

450 and 475 SC		450 DC		450 IDC	
Component	EFOH	Component	EFOH	Component	EFOH
SG	57.0	Intercooler and precooler	29.9	IHX	57.0
SCHE and SCMC	6.8	SCHE and SCMC	6.8	SCHE and SCMC	6.8
MCMC	3.4	GC	6.2	MCMC	3.4
		Isolation HX	2.2		
Total	67.2	Total	45.1	Total	67.2

This results in an EFOH of 29.9 h per year for the DC precooler and intercooler leaks. The total EFOH due to the heat exchangers for the SC, DC, and IDC concepts are 67, 45, and 67.

Overall, in terms of moisture ingress consequences, the IDC is the best since it has the lowest moisture ingress frequency and the least amount of water available for ingress. The DC also has a lower moisture ingress frequency than the SC. The plant down time as the result of heat exchanger failure is the least for the DC due to the low failure frequencies of the precooler and intercooler. The SC and IDC cycles have equivalent heat exchanger EFOH. The contribution of the heat exchanger to the total forced outage rate is only 12% or less; therefore, the IDC is judged best in overall terms.

9.4 OVERALL EVALUATION OF SAFETY AND LICENSABILITY

An overall assessment of the relative safety and licensability of the gas turbine and SC design was made. These are summarized below.

9.4.1 Overall Safety Evaluation

The qualitative safety evaluation of the gas turbine and SC designs was based on examining the important accident families which have been identified in previous SC safety assessments. Based on this evaluation a comparison of the plants was made.

The 350 MWt SC-MHR probabilistic risk assessment (PRA) identified four accident categories:

1. Primary coolant leaks with forced cooling (DFC).
2. Water ingress into the vessel with forced cooling (WFC).
3. Primary coolant leaks with loss of all forced cooling (DCC).
4. Water ingress with loss of all forced cooling (WCC).

Based on this PRA, the accident families can be briefly characterized. The DFCs are most frequent but with the smallest dose. The WFCs are less frequent but with a larger dose. The DCCs are the risk dominant accident family. The WCCs are least likely but have the largest consequence.

Differences pertaining to the DC and IDC plant designs for each of these accident families were considered. In addition, new accident categories specific to the gas-turbine designs were also considered.

Water ingress events pose a challenge to the safety functions of the core graphite and fuel. Moisture can oxidize core graphite reducing its strength, and it can react with defective fuel causing changes in the kernel density resulting in increased fission product release. While moisture ingress events have the potential for relatively large fission product release, events that result in offsite doses are expected to be of low frequency in the SC design. The 450 MWt SC plant design also includes a Vented Low Pressure Containment (VLPC) which has a filter on the vessel pressure relief trains. This filter significantly reduces the offsite dose consequences of those water ingress events that result in relief valve opening by removing condensible fission products.

Qualitatively, the risk of water ingress into the primary coolant during power operation is much less for the gas turbine designs, which do not have a steam generator. However, the risk of water ingress during cold shutdowns, which relates to reactivity addition rather than fuel hydrolysis or graphite oxidation, is approximately the same for all concepts.

The accident families which result in loss of primary coolant are designated the DFCs and DCCs as previously discussed. The dominant accidents that contribute to risk are in the DCC family - primary coolant leaks followed by loss of all forced cooling. The likelihood of this accident family depends on the frequency of primary coolant leaks and the probability that forced cooling is lost following the leak. The risk from this event is a function of the likelihood, along with the accident consequences, or offsite dose.

The DCC accident family is initiated by primary coolant leaks in the vessel system. The initiating event frequency increases with the number of vessel system penetrations. Based on early information, it appears that the DC power conversion vessel design will result in an increase in the number of penetrations, as compared to the SC. It is judged that the IDC will have about the same number of penetrations as the SC. In summary, compared to the reference 450 MWt SC, it is judged that the initiating event frequency of primary coolant leak is about the same in the 475 MWt SC and IDC gas turbine plant designs. The DC gas turbine is judged slightly worse.

The probability of successful main loop cooling following a leak is assumed better in the gas turbine designs. Failure of the main cooling loop is dominated by loss of power to the main

loop and mechanical failures which lead to loss of the heat sink. Loss of power is judged to affect all concepts equally. However, mechanical failures are expected to be more likely in the SC because the BOP has many more valves, pumps, reheaters, etc. The DC does not have a main circulator as compared to the IDC and is therefore simpler. In summary, compared to the reference 450 MWt SC it is judged that the probability of main loop cooling is about the same in the 475 MWt SC. The DC and IDC gas turbine plants are judged to be better.

Radionuclide releases associated with DCC events are influenced by the initial operating conditions within the core and the mechanisms for transporting radionuclides from the vessel to the vented low pressure containment (VLPC). The higher initial core operating temperatures in the GT-MHR designs lead to radionuclide releases from the core early in the accident that are higher than the SC. When considering the mechanisms that control the rate of release from the vessel to the VLPC, the equilibrium pressure within the vessel following loss of the turbine or heat transport system is particularly important. Because the equilibrium pressure in the DC vessel is lower, analyses indicate that the SC and IDC have longer and higher initial depressurization rates to the VLPC than the DC. This results in higher releases. Since the temperature effects dominate, both GT-MHR plants will have releases higher than the SC plants. However, the lower equilibrium pressure in the DC will result in lower releases than projected for the IDC.

Overall, the elimination of the water ingress event during power production represents a major safety advantage for the GT-MHR concepts. It eliminates the highest consequence event associated with the SC concept. In addition, the lower equilibrium pressure in the DC following loss of helium circulation provides a safety advantage relative to the IDC in terms of radiological releases associated with DCC events. This advantage must be weighed against the uncertainties associated with turbomachinery failure events.

9.4.2 Overall Licensability of GT-MHR Concepts

The licensing challenges of GT-MHR concepts can be evaluated by comparison with a comparably sized SC-MHR plant. Gas-cooled SC plants are licensable, as evidenced by Peach Bottom and Fort St. Vrain, licensed in the United States, and many other gas-cooled SC plants licensed in Europe and the United Kingdom. A uniqueness of the MHR, whether it be a SC or gas turbine plant is the passive decay heat removal capability of the reactor cavity cooling system (RCCS). This passive heat removal capability is made possible by design selections which include the power density and annular core configuration, high heat capacity of the graphite core, and an uninsulated reactor vessel for transmission of heat to the RCCS.

A number of capabilities are fundamental to the safety philosophy of the MHR. These capabilities are discussed, below, along with a judgement of the relative ability of the gas turbine concepts to achieve the capability, in comparison with the reference SC concept.

1. Fission Products Retained in Fuel Particle

The gas turbine concepts have higher core outlet temperatures with higher helium flow than a comparable SC concept during normal operation. During conduction cooldown

events, the gas turbine concepts have slightly higher fuel temperatures than the SC concept. Both however remain within the acceptable limit.

Assessment: The higher temperature during normal operation for the gas turbine will create a greater challenge.

2. Primary Coolant Not Required to Cool the Core

The capability to cool the reactor core with or without the primary coolant inventory remains essentially unchanged in the gas turbine versus SC concepts. This is true since the core decay heat can be removed by forced cooling or by passive heat removal by the RCCS.

Assessment: The concepts are essentially equivalent for decay heat removal regarding the retention or nonretention of the primary coolant.

3. Large Graphite Heat Capacity Results in Slow Predictable Thermal Transients

By design, the relatively large heat capacity of the graphite in the core essentially remains the same, regardless of the concept selected.

Assessment: The concepts are essentially equivalent regarding the capability to moderate thermal transients.

4. Passive Heat Removal

Passive heat removal, with or without the RCCS, is a function of core configuration, decay heat level, which is a function of prior operating history, and the quantity of core graphite to moderate the thermal transient. The translation of these functions to the capability to transmit core heat to the reactor vessel wall is calculable using existing methods. Of significance regarding this assessment, is the observation that the ability to dissipate the decay heat is independent of the cycle used for the conversion of thermal energy into electrical energy.

Assessment: Passive heat removal capability, given proper design selections, is independent of the means selected for conversion of reactor thermal energy into electrical energy.

5. Accidents Bounded by Safety Analysis

Extensive safety analyses have been performed for the SC, including accidents proposed by others. In all cases, credible accident scenarios were determined to satisfy the acceptance criteria. In question is whether there are unique characteristics of the GT-MHR which would pose a more difficult challenge, or conversely, a significant advantage.

Gas turbine deblading is one such issue. Such an event could result from a sudden loss of grid electrical load, combined with a failure of both the control and protective systems to prevent overspeed of the turbomachinery. The consequences of turbomachinery mechanical failures are perceived to be more difficult to bound by analysis than comparable SC events. The IDC would be comparable to the SC, since the turbomachinery is located in the secondary coolant loop in both cases. However there is more experience with the use of steam for energy conversion, as compared with closed gas turbine cycles.

Steam generator tube leaks in the SC concept result in water ingress into the primary coolant system, potentially leading to fuel damage due to hydrolysis of the fuel. While evaluated to be acceptable, this is the dominant consequence event for the SC. In both the DC and IDC concepts, there are water coolers in contact with primary coolant. However, the pressure differentials are such that, during operation, the primary coolant would leak into the water heat exchangers, as opposed to water into the primary system. The likelihood of leaks from the primary system is greater in the case of the DC, which includes the precooler and intercooler in the primary loop. The consequences of contaminated helium leakage into the water coolant will require evaluation.

Assessment: Turbine overspeed and turbine deblading are challenges for the DC gas turbine and will have to be accommodated in the safety analysis. Relative to the gas turbine concepts, the SC has a disadvantage in water ingress via the steam generator. These events have been and remain licensable.

REFERENCES

- 9-1 Letter, Strategic Power Systems to ABB/CENP, "...Major Failure Events Experienced by Turbomachinery," June 4, 1993.
- 9-2 Letter, ABB Power Generation, Ltd. to ABB/CENP, "...Analysis of (Gas Turbine) Field Data," March 26, 1993.
- 9-3 "MHTGR-GT Availability and Reliability Assessment: Existing Gas Turbine Technology," Report 2609301, Strategic Power Systems, March 26, 1993.
- 9-4 "RELAP5/MOD3 Code Manual," Vols 1-4, NUREG/CR-5535, EGG-2596, June 1990.

SECTION 10

AVAILABILITY, RELIABILITY AND CAPACITY FACTOR

The availability, reliability and capacity factors of the direct cycle (DC) and indirect cycle (IDC) gas turbine concepts were assessed as part of the current evaluation, using the existing steam cycle (SC) database as a starting point. In support of this effort, data related to the operation of conventional gas turbine electric generation and cogeneration power plants were obtained and analyzed. The results of the availability, reliability and capacity factor assessment are reported in this section in two major parts. First, the results of the turbomachinery database review is presented. Second, these results are utilized as input in deriving scoping estimates of the scheduled and forced outage characteristics and overall capacity factors for the various MHR design options.

10.1 TURBOMACHINERY RELIABILITY ASSESSMENT

An Availability and Reliability Assessment was performed by Strategic Power Systems, Inc.[®] (SPS) using the Operational Reliability Analysis Program (ORAP) database of industrial gas turbines. A statistical evaluation of this database provided availability, reliability, and service and capacity factors, as well as outage factors, failure rates, and mean-time-between failures and mean time to repair, all in accordance with IEEE standards. Definitions of these terms are summarized in Table 10-1. The results provided in Reference 10-1 show the historical trends in these parameters and the integrated values for the most recent 5-year period.

ORAP was established in the mid-1970s to provide feedback to turbomachinery designers and utility users in order to validate design changes for reliability improvement. As the program developed, the database was extended to cover a wider range of simple-cycle and combined-cycle plants and to provide reliability information on a system and component basis. The ORAP database is currently maintained and administered by SPS, which receives data from utility and cogenerator participants, reviews the data for accuracy, verifies the data with the reporting organization, and then enters it into the database. The ORAP information system is then converted into Reliability, Availability, and Maintenance (RAM) statistics.

The ORAP database comprises information on 373 domestic U.S. units and 163 international heavy-duty units. Most of these units are direct drive at 3000/3600 rpm with output power in the range 20-150 MWe. The database includes several aerodynamically coupled (twin shaft) units. The database also includes 84 domestic U.S. and 18 international aero-derivative units, most of which are aerodynamically coupled with high speed shafts up to 11,000 rpm. The power range for these units is 5-40 MWe. These are all open-cycle units with a collective time in operation approaching 10,000 unit-years.

The simple-cycle plant reliability of all heavy-duty units (> 50 MWe) showed a continual increase from 92.5% (in 1977) to 99.1% (in 1992) with values between 98.0% to 99.2% in the last 5-year period (1988-1992). The simple-cycle plant availability also demonstrated a continual increase from 79.9% to 94.0% over the same historical period, 1977 to 1992, with the latest 5-

Table 10-1 DEFINITION OF TERMS

Availability (AVAIL)	The percentage of time a unit is functional and able to serve load (service and ready standby hours) at any capacity.
Unavailability (UNAV)	The percentage of time that a unit is functionally unable to serve load either due to forced or scheduled outage.
Reliability (RELIAB)	The percentage of time that a unit is either functional and able to serve load at any capacity or is experiencing scheduled outage.
Forced Outage Factor (FOF)	The percentage of time that a unit is unable to serve load due to outage time associated with a failure to start, an automatic or manual trip from a state of operation, or a forced outage initiated from a state of reserve.
Scheduled Outage Factor (SOF)	The percentage of time that a unit is unable to serve load due to outage time associated with scheduled or unscheduled maintenance.
Starting Reliability (SR)	The probability that a unit which is classified as available and in ready reserve is able to be started and brought into synchronization within a specified period of time.
Starting Unreliability (SUR)	The probability that a unit which is classified as available and in ready reserve is unable to be started and brought to synchronization within a specified period of time.
Failure Rate per Million Fired Hours (FR/MFH)	The average number of forced outages (automatic or manual trips from a state of operation) that will be experienced in one million operating hours (fired/service hours).
Failure Rate per Year (FR/YR)	The average number of forced outages (automatic or manual trips from a state of operation) that will be experienced in a calendar year.

Table 10-1 (continued)

Mean Time To Repair (MTTR)	The average amount of time required to perform corrective actions to bring the unit out of a forced outage.
Mean Time Between Failures (MTBF)	The average amount of operating time (fired/service hours) that a unit will run before experiencing a forced outage (automatic or manual trip).
Service Factor (SF)	The percentage of time that a unit is operating (fired/service hours).
Capacity Factor (CF)	The percentage of maximum generation a unit generates for a given period of time.
Output Factor (OF)	The percentage of maximum generation a unit generates for the period of time that the unit is serving load (operational).

year period being 91.1% to 94.1%. The service factor also showed a continual increase from 17.7% to 30% in the total historical period, reflecting more baseloaded hours in the database. This is also reflected in the increasing capacity factor (5% to 27%) and output factor (28.6% to 92.8%). The North American Electric Reliability Council (NERC) Generating Availability Data System (GADS) was examined for the 1977 to 1990 period. Plant Reliability and Availability showed a very similar continual increasing trend. However, the NERC database showed very low service factor and capacity and operating factors, reflecting the largely peaking duty operation of these units.

The assessment examined a reduced set of data involving heavy-duty units, with power ratings greater than 50 MWe (70-100 MWe) and with service factors greater than 50%, reflecting base load duty. This data subset covers 34 domestic U.S. units of relatively recent design, 3 of which are in utility service and the remainder in cogeneration service. Prior to 1984, there were only 3-5 of these units in service; subsequently the number of units in the database rose rapidly and steadily to 34.

For this focused database, the simple-cycle plant reliability has remained approximately 99% for both utility and cogeneration units for the most recent 5-year period (1988-1992). The associated simple-cycle plant availability is 94% for the utility units and 95.8% for the cogeneration units. The service factor for the cogeneration units is much higher than the utility units (92.6% vs 65.6%). Similar relationships exist for the capacity factor (82.9% vs 53.5%) and the output factor (89.8% vs 80.5%).

Contributors to system and component RAM statistics were also examined. The major systems are: gas turbine, flange-to-flange (turbine, combustion, compressor, and bearings systems); gas turbine, controls and accessories; generator including control/protection and auxiliaries; and balance-of-plant (power distribution and station/plant equipment). The RAM statistics for the reduced database over the 1988-1992 period for major systems is summarized in Table 10-2 which is abstracted from Reference 10-1. The data were examined for the utility units and cogeneration units separately and combined. The gas turbine controls and accessories system is the single major contributor to the overall failure rate (FR), both on an annual basis and on a firing-hours basis. The details of this major system for cogeneration and utility applications are provided in Table 10-3, reproduced from Reference 10-1. The failure rate includes significant contributions from systems related to the combustor which is not relevant to the MHR. Table 10-2 also indicates that the turbomachinery itself is the smallest contributor to the failure rate.

The gas turbine (flange-to-flange) is the major contributor to the unavailability. The RAM statistics for this major system are summarized in Table 10-4 which is abstracted from Reference 10-1. The unavailability is the sum of the forced and scheduled outage factors, and, as is readily apparent, it is dominated by the scheduled outages. The turbine system is the largest contributor to the unavailability and to the individual forced and scheduled outage factors of the flange-to-flange gas turbine systems category. Returning to Table 10-2 for a comparison among major systems categories, it is noted that the scheduled outage factor is dominated by the gas turbine flange-to-flange category which, in turn, is dominated by the turbine system. The forced outage factor (essentially the complement of the reliability) has no overwhelming dominant component. Removal of combustion-related systems for direct relevance to the MHR would not have a large effect on the forced outage factor and, consequently, the reliability. However, it must be recalled that reliability is already indicated to be above 99%.

An examination of the turbine system component RAM statistics provided in Reference 10-1 indicates two major contributors to the scheduled outage factor: hot gas path inspection and major overhaul. A close third contributor (10-15%) to the scheduled outage factor is the category of general/preventive maintenance.

The aero-derivative gas turbines, primarily twin-shaft, aerodynamically coupled units, were also examined in support of the twin-shaft GT-MHR being considered for the indirect cycle. Data for two different designs (designated as Fleet A and Fleet B) is provided in Reference 10-1 for 1991 and 1992. For Fleet A the data indicates a reliability of 99.1-99.2% for the most part. For Fleet B it is mainly in the range 97%-98%. Availability varied between 92.4% and 96.5% for Fleet A and 89.5% and 94.2% for Fleet B. However, there appears to be no historical trend. Service factor varied between 90.3% and 95% for Fleet A and 85.7% and 93.7% for Fleet B, also with no definitive historical trend. For Fleet A, the major system categories of "gas generator" and "controls and accessories" essentially provide the entire contributor to the forced outage factor. For Fleet B the "gas generator" provided more than half the forced outage factor and "controls and accessories" provided another 25%. The "gas generator" was the single major contributor to the scheduled outage factor.

Table 10-2
 OPERATIONAL RELIABILITY ANALYSIS PROGRAM (ORAP)
 MODERN, BASELOADED GAS TURBINES (>50 MW)
 SYSTEM CONTRIBUTORS TO UNAVAILABILITY (1988 - 1992)

	UNAV (%)	FOF (%)	SOF (%)	SUR (%)	FR/MFH (\$/mfh)	FR/YR (\$/yr)	MTTR (hrs)	MTBF (hrs)
COGENERATION AND UTILITY APPLICATIONS								
Gas Turbine (Flange-to-flange)	3.31	0.17	3.14	0.34	34.65	0.28	11.42	28866.53
Gas Turbine (Controls and Accessories)	0.76	0.39	0.37	4.00	364.37	2.84	8.96	2744.36
Generator-Gas Turbine	0.19	0.12	0.07	0.34	81.24	0.64	14.17	12310.73
Balance-of-Plant-Gas Turbine	0.25	0.14	0.11	0.00	70.48	0.55	20.72	14188.63
Simple-Cycle Plant Totals	4.51	0.82	3.69	4.68	550.74	4.31	11.39	1815.90
COGENERATION APPLICATIONS								
Gas Turbine (flange-to-flange)	3.13	0.19	2.94	0.45	33.40	0.26	10.99	29948.51
Gas Turbine (Controls and Accessories)	0.65	0.38	0.27	3.75	355.30	2.88	7.88	2814.71
Generator-Gas Turbine	0.18	0.12	0.06	0.41	77.47	0.63	15.28	12908.84
Balance of Plant-Gas Turbine	0.28	0.17	0.11	0.00	78.80	0.64	20.72	12690.05
Simple-Cycle Plant Totals	4.24	0.86	3.38	4.61	544.97	4.41	10.98	1835.08
UTILITY APPLICATIONS								
Gas Turbine (flange-to-flange)	4.43	0.04	4.39	0.00	45.24	0.26	14.15	22104.15
Gas Turbine (Controls and Accessories)	1.39	0.50	0.89	4.80	441.09	2.55	16.26	2267.09
Generator-Gas Turbine	0.16	0.12	0.04	0.13	113.10	0.65	7.74	8841.66
Balance-of-Plant Gas Turbine	0.05	0.00	0.05	0.00	0.00	0.00	--	--
Simple-Cycle Plant Totals	6.03	0.66	5.37	4.93	599.43	3.40	14.49	1668.24

Operational Reliability Analysis Program (ORAP)
Modern, Baseloaded Gas Turbines (>50 MW): Cogeneration and Utility Applications
System Contributors to Unavailability (1988 - 1992)

	UNAV (%)	FOF (%)	SOF (%)	SUR (%)	FR/MFH (#/mfh)	FR/YR (#/yr)	MTTR (hrs)	MTBF (hrs)
<u>Gas Turbine (Controls and Accessories)</u>								
Fuel Oil System	0.15	0.02	0.13	0.06	59.73	0.47	4.52	16742.59
Purge Air System	0.10	0.10	0.00	0.03	4.78	0.04	226.30	209282.33
General Gas Turbine	0.09	0.07	0.02	3.13	10.75	0.08	1.43	93014.37
Fuel Gas System	0.07	0.04	0.03	0.09	29.86	0.23	14.08	33485.17
Starting System	0.07	0.03	0.04	0.16	1.20	0.01	139.00	837129.31
Fuel Oil Forward/Treatment	0.03	0.00	0.03	0.00	11.95	0.09	0.69	83712.93
NOx Injection System	0.02	0.01	0.01	0.06	35.84	0.28	3.76	27904.31
Atomizing Air System	0.01	0.00	0.01	0.06	7.17	0.06	1.92	139521.55
Control System	0.04	0.03	0.01	0.16	94.37	0.73	2.25	10596.57
Cooling Water System	0.03	0.00	0.03	0.00	1.20	0.01	14.30	837129.31
Exhaust System	0.03	0.01	0.02	0.00	5.97	0.05	11.66	167425.86
Hydraulic Control and Trip	0.03	0.02	0.01	0.22	34.64	0.27	7.10	28866.53
Lube Oil System	0.03	0.02	0.01	0.00	11.95	0.09	19.59	83712.93
Accessory Drive Systems	0.02	0.02	0.00	0.00	2.39	0.02	82.10	418564.66
Inlet System	0.02	0.00	0.02	0.00	0.00	0.00	—	—
Auxiliary Power System	0.01	0.01	0.00	0.00	11.95	0.09	4.90	83712.93
Fire Protection System	0.01	0.01	0.00	0.03	34.64	0.27	2.97	28866.53
Cleaning and Washing System	0.00	0.00	0.00	0.00	0.00	0.00	—	—
Cooling and Sealing Air System	0.00	0.00	0.00	0.00	2.39	0.02	5.15	418564.66
Enclosures	0.00	0.00	0.00	0.00	2.39	0.02	1.25	418564.66
Heating and Ventilation System	0.00	0.00	0.00	0.00	1.20	0.01	1.00	837129.31
Turning Gear	<u>0.00</u>	<u>0.00</u>	<u>0.00</u>	<u>0.00</u>	<u>0.00</u>	<u>0.00</u>	<u>—</u>	<u>—</u>
Subtotals:	0.76	0.39	0.37	4.00	364.37	2.84	8.96	2744.36

Note: The systems that are bolded represent systems that would not be included in the MHTGR-GT design.

Table 10-3 SYSTEM CONTRIBUTORS TO UNAVAILABILITY - CONTROLS AND

TABLE 10-4
OPERATIONAL RELIABILITY ANALYSIS PROGRAM (ORAP)
MODERN, BASELOADED GAS TURBINES (>50 MW)
SYSTEM CONTRIBUTORS TO UNAVAILABILITY (1988-1992)

	UNAV (%)	FOF (%)	SOF (%)	SUR (%)	FR/MFH (#/mfh)	FR/YR (#/yr)	MTTR (hrs)	MTBF (hrs)
COGENERATION AND UTILITY APPLICATIONS								
Turbine Systems	2.23	0.14	2.09	0.00	5.97	0.05	6.04	167425.86
Combustion System	0.88	0.02	0.86	0.34	25.09	0.20	9.95	39863.30
Compressor System	0.19	0.00	0.19	0.00	1.20	0.01	8.00	837129.31
Bearings	0.01	0.01	0.00	0.00	2.39	0.02	42.05	418564.66
Gas Turbine (flange-to-flange) Subtotals	3.31	0.17	3.14	0.34	34.65	0.28	11.42	28866.53
COGENERATION APPLICATIONS								
Turbine Systems	2.02	0.16	1.86	0.00	6.68	0.05	6.04	149742.54
Combustion System	0.88	0.02	0.86	0.45	22.71	0.18	8.97	44041.92
Compressor System	0.22	0.00	0.22	0.00	1.34	0.01	8.00	748712.71
Bearings	0.01	0.01	0.00	0.00	2.67	0.02	42.05	374356.36
Gas Turbine (flange-to-flange) Subtotals	3.13	0.19	2.94	0.45	33.40	0.26	10.99	29948.51
UTILITY APPLICATIONS								
Turbine Systems	3.45	0.00	3.45	0.00	0.00	0.00	--	--
Combustion System	0.95	0.04	0.91	0.00	45.24	0.26	14.15	22104.15
Compressor System	0.03	0.00	0.03	0.00	0.00	0.00	--	--
Gas Turbine (flange-to-flange) Subtotals	4.43	0.04	4.39	0.00	45.24	0.26	14.15	22104.15

10.2 SC-MHR CAPACITY FACTOR ASSESSMENT AND EXTENSION TO THE GT-MHR

This section provides a preliminary capacity factor comparison of the gas turbine and steam cycle designs (450 MWt SC, 475 MWt SC, 450 MWt DC, and 450 MWt IDC). They are all assumed to be base-loaded, with the evaluated plant capacity factor for the target plant averaged over the lifetime of the plant.

10.2.1 Scheduled Outage Assessment

10.2.1.1 SC Plant Scheduled Outage Assessments

As the differences between the 450 and 475 MWt SC-MHR with respect to maintenance and refueling activities are small, these two concepts are assumed to have the same scheduled outage rate. Major assumptions in the SC scheduled outage assessment are:

1. Refueling schedule is once every 18 months.
2. Refueling time (Shutdown and Depressurize, Fuel handling equipment movements, Refueling sequence, Repressurize and Startup) for one module is 15.5 days.
3. The SC plant's NI and ECA scheduled outage activity's duration, required periodicity, and ability to be performed coincident with the actual refueling process are consistent with prior assessments of SC plant capacity factor.

Scheduled outages over the life of the plant are shown in Table 10-5 for one SC-MHR module/unit. Columns three and four provide estimates of the duration of the nuclear island (NI) and energy conversion area (ECA) outages, respectively. The NI and ECA scheduled outages are based on a list of activities requiring reactor module or turbine-generator shutdown. These activities are allocated to the various planned outages associated with every refueling. The overall duration of each refueling is determined by summing the duration of NI and ECA activities that are on the critical path. The NI outage types are consistent with those used in prior assessments of the scheduled outage contribution to unavailability in SC plants. Since the NI and the ECA scheduled outage activities can be carried out concurrently, the plant scheduled outage at each of the refuelings is the maximum of the NI and ECA scheduled outage numbers. The scheduled outage limiting activity is shown in column six of Table 10-5. Most of these limiting activities are ECA-related turbine-generator set inspection, warranty, major overhaul, or rebuilding activities. However NI outages of type A', C, G, and K are also limiting.

The NI scheduled outage rate of 391.4 hours per year is the sum of the NI scheduled outages, divided by the plant life of 40 years. Similarly, the module scheduled outage rate is 528.3 hours per year. Since a loss of any one of the four modules/units causes a 25% plant capacity loss, the plant scheduled outage rate is also 528.3 hours per year, equivalent to a 6% unavailability. Note that the scheduled outage is equal to the planned outage, as no planned deratings have been identified for the SC-MHR.

Table 10-5
SC-MHR LIFETIME SCHEDULED OUTAGE ASSESSMENT

Refueling Outage Number	Months in Service	NI Scheduled Outage Duration (h)	ECA Scheduled Outage Duration (h)	Plant Scheduled Outage Duration (h)	SC Scheduled Outage Limiting Activity
1st	18	650.8	504	650.8	NI outage type A'
2nd	36	726.8	1440	1440	T-G warranty
3rd	54	474.8	504	504	T-G inspection
4th	72	726.8	504	726.8	NI outage type C
5th	90	410.8	504	504	T-G inspection
6th	108	870.8	1344	1344	T-G overhaul
7th	126	410.8	504	504	T-G inspection
8th	144	726.8	504	726.8	NI outage type C
9th	162	474.8	504	504	T-G inspection
10th	180	698.8	1344	1344	T-G overhaul
11th	198	410.8	504	504	T-G inspection
12th	216	870.8	504	870.8	NI outage type G
13th	234	410.8	504	504	T-G inspection
14th	252	698.8	1680	1680	T-G rebuilding
15th	270	410.8	504	504	T-G inspection
16th	288	698.8	504	698.8	NI outage type K
17th	306	410.8	504	504	T-G inspection
18th	324	870.8	1344	1344	T-G overhaul
19th	342	410.8	504	504	T-G inspection
20th	360	698.8	504	698.8	NI outage type K
21st	378	410.8	504	504	T-G inspection
22nd	396	726.8	1344	1344	T-G overhaul
23rd	414	474.8	504	504	T-G inspection
24th	432	870.8	504	870.8	NI outage type G
25th	450	410.8	504	504	T-G inspection
26th	468	698.8	1344	1344	T-G overhaul
End of life	480				

NOTE: The NI outage types are consistent with prior assessments of the scheduled outage contribution to unavailability for SC plants.

10.2.1.2 450 MWt DC Plant Scheduled Outage Assessment

In evaluating the DC scheduled outage rate, the following assumptions are made:

1. Except for steam generator tube bundle chemical cleaning and main circulator motor maintenance, all other NI scheduled outages for an equivalent SC plant continue to be applicable.
2. The turbomachine and its associated seals are removed from the power conversion vessel for inspection and maintenance every seven and half years, and it takes ten days to remove the turbomachine and replace it with the one spare turbomachine that exists on site. The ten-day period assumed for removal and replacement of the turbomachine is based on twice the five-day period required for removal and replacement of large gas circulators in AGRs in the UK.
3. Maintenance/ISI activities on other items like frequency converter, precooler, and intercooler can be done concurrent with refueling. The recuperator is not inspected and is built for the plant life of 40 years. Inspection of control valves is performed in-situ, and is concurrent with the vessel leak/pressure test. Balance of plant maintenance activities can be performed concurrently with the NI scheduled outages.

An examination of the time-scaled schedule logic diagrams of the SC's NI scheduled outage types show that, for all of them, the steam generator tube bundle chemical cleaning and main circulator motor maintenance activities are not on the critical path. So the duration of the NI scheduled outage types do not change for the DC plant.

The turbomachinery removal and replacement will be done with the building crane. The building crane is on a different set of rails than the fuel handling machine and, by design requirements, there is no interference between the two. So the two activities, turbomachinery removal and refueling, can be done concurrently as long as the reactor is shutdown and depressurized. An examination of the NI outages indicate that the smallest amount of time the reactor is shutdown and depressurized in any of the outage types is about 15.5 days. This is conservatively larger than the required ten days. This implies that the critical path schedule is governed by the NI scheduled outages as described for the SC. Hence, the DC plant scheduled outage rate is assessed to be identical to the SC's NI scheduled outage rate of 391.4 hours per year, or 4.5% unavailability.

10.2.1.3 450 MWt IDC Plant Scheduled Outage Assessment

In evaluating the IDC scheduled outage rate, the following assumptions are made:

1. Except for steam generator tube bundle chemical cleaning, all other NI scheduled outages for an equivalent SC plant continue to be applicable.
2. The turbomachinery and its associated seals have a five day inspection every odd-numbered refueling and a twenty day inspection every even-numbered refueling.

3. Inspection of control valves is performed in-situ and is concurrent with repressurization of the secondary loop, which is concurrent with vessel pressurization. Maintenance activities on other items like the precooler and intercoolers can be done concurrently with NI activities. The recuperator is not inspected and is built for the plant life of forty years. Balance of plant maintenance activities can be performed concurrently with the NI scheduled outages.

Therefore, the SC classification of scheduled outages continues to be applicable, and the durations of the NI scheduled outage types remain unchanged. Assumption (2) is based on experience with gas-fired turbomachinery units, as modified for the MHR operating conditions. An examination of the NI outages in Table 10-5 shows that for odd-numbered refuelings the minimum NI scheduled outage duration is 410.8 h and for even-numbered refuelings the minimum NI scheduled outage duration is 698.8 h. These times are conservatively larger than the inspection durations of the IDC. The two activities, turbomachinery inspection and NI scheduled activities, can be done concurrently, so the critical path schedule is governed by the NI scheduled outages as described for the SC. Hence, the IDC plant scheduled outage rate is assessed to be identical to the SC's NI scheduled outage rate of 391.4 h per year, or 4.5% unavailability.

10.2.1.4 Qualitative Uncertainty Comparison

The 450 and 475 MWt SC-MHR are judged to have a "reference" uncertainty on its unavailability of 6.0%. The "reference" uncertainty is due to system designs that are relatively well established and due to a previous history of developing scheduled outage plans that provide some confidence in the results. The 475 MWt IDC concept is judged to have a higher uncertainty than the SC concept because: (a) The system designs are not well established and, with design evolution, additional components may be required that will result in additional scheduled outages; (b) The scheduled outage evaluation of IDC unique components, most of which are evaluated based on available data, need to be substantiated. The 450 MWt DC concept is judged to have a slightly higher uncertainty than the IDC concept because, apart from the uncertainties shared with the IDC, the removal and replacement time for DC turbomachinery needs to be better substantiated. In addition, the plans for DC seals inspection and its duration need to be known better.

10.2.2 Forced Outage Assessment

The plant forced outage assessments for the various concepts build upon the results of earlier work in support of the 450 MWt SC-MHR. The plant forced outage assessment for all concepts, including the 450 MWt SC-MHR, have been updated for the current evaluation and based conservatively on service hours of 8322 h per year (95% of the year) for systems and components performing residual heat removal and 8060 h per year (92% of the year) for all other systems and components. This reflects the expected higher capacity factors for most of the MHR concepts. The results of the forced outage assessment are summarized in Table 10-6.

Table 10-6
GAS TURBINE AND STEAM CYCLE EFOH ASSESSMENT RESULTS

Systems	450 & 475 MWt Steam Cycle	450 MWt Direct Cycle GT	450 MWt Indirect Cycle GT
Reactor System	30.8	30.8	30.8
Vessel System	108.2	120.0	108.2
Heat Transport System	123.8	79.3	121.0
Shutdown Cooling System	6.8	6.8	6.8
Shutdown Cooling Water System	0.2	0.2	0.2
Reactor Cavity Cooling System	0	0	0
Nuclear Island Cooling Water System	0.2	2.4	0.2
Helium Services System	24.7	24.7	24.7
Radwaste and Decontamination System	3.4	3.4	3.4
Plant Protection Systems	54.4	54.4	54.4
Plant Control, Data, and Instrumentation System	43.3	43.3	43.3
Plant Monitoring System	3.2	3.2	3.2
Turbomachinery System	105.6	127.1	45.0
Secondary Loop	112.9	N/A	29.4
Circulating Water System	14.8	14.8	14.8
Energy Conversion Area Cooling Water System	0.2	N/A	2.2
HVAC Systems	7.6	7.6	7.6
Instrument and Service Air System	0.2	0.2	0.2
Essential AC Electrical System	7.5	7.5	7.5
Plant AC Electrical System	15.2	15.2	15.2
Essential DC Electrical System	7.6	7.6	7.6
Operator and maintenance errors	9.3	9.3	9.3
Other mechanical services	3.9	3.9	3.9
Total Forced Outage Rate (hours per year)	683.8	561.7	538.9
	7.8%	6.4%	6.2%

10.2.2.1 450 and 475 MWt SC Plant Forced Outage Assessment

The 475 MWt SC-MHR is assumed to have the same capacity factor as the 450 MWt SC-MHR concept. The EFOH of the SC systems has been updated to reflect the increased service hours. Two categories, operator and maintenance errors and failures due to other mechanical services, are expected to be independent of this variation in service hours and, so, are assumed to be the same as for the 450 MWt SC. The second column of Table 10-6 describes the SC forced outage assessment results.

10.2.2.2 EFOH Update of GT Systems Similar to SC

In the systems of the MHR that are presented in Table 10-6, shaded systems denote those that have major differences with the SC. Unshaded systems are expected to be similar to the SC, and are assumed to have the same EFOH for all concepts. The following systems have only small EFOH differences between the GT and SC concepts.

Vessel Systems: Primary coolant leak frequency is assessed to increase by 11% over the SC — resulting in an EFOH of 120 h per year. This is due to the DC's normal control and fast opening bypass valves, which, along with their maintenance block valves, are located on the outside of the power conversion vessel. Although the IHX vessel is much larger and may have different size penetrations for the secondary helium loop piping, the number of valves and bolted seams at the primary coolant boundary are expected to be comparable to that in a steam generator vessel's boundary. So it is assumed that the forced outage rate due to leaks in the primary coolant boundary for the IDC is similar to that evaluated for a SC — 108.2 h per year.

NI Cooling Water Systems: With the limited detail available for the intermediate cooling water loop of the DC, its EFOH is assumed to be proportional to the heat load, resulting in an EFOH of 2.4 h per year for the NI Cooling Water System. The IDC's EFOH is assumed to be similar to that evaluated for the SC, equal to 0.2 h per year.

IDC's ECA Cooling Water System: With the limited detail available for the IDC's ECA Cooling Water System, its EFOH is assumed to be proportional to the heat load, resulting in an EFOH of 2.2 h per year.

HVAC Systems: The HVAC systems for both the DC and IDC gas turbine plants are assumed to be similar to the SC plant, all of them having similar heat loads. So with the limited detail available, it is assumed that the EFOH of the DC and IDC is equal to 7.6 h per year.

10.2.2.3 Heat Transport System

Direct Cycle: A weld failure model was used to assign a tube leak frequency of 0.035 per module-year. The outage time is dominated by the time it takes to detect and plug a leaking precooler or intercooler tube which, because it is a leak into the secondary side and because of a larger number of tubes, is assumed to be 1.5 times that for steam generator tube detection and plugging. This results in an EFOH of 29.9 h per year for the DC precooler and intercooler leaks.

Based on a previous HTGR-GT availability evaluation, a failure rate of 0.08 failures per valve year is assumed for the two normal control bypass valves and the four redundant fast opening bypass valves. The mean time to repair is assumed to be four days, two days for startup/shutdown and two days for replacement of the bypass valve. This results in an EFOH due to bypass valve failures of 43.8 h per year.

The DC concept has four graphite compressor seals around the two stages of the compressors. There are no turbine inlet seals, as the shrouds are welded in that location. The four compressor seals' failure could cause bypass flows that will reduce power production. It was assumed that complete failures are highly unlikely and typical failures are 1% of complete failures, resulting in a average derating of 2.5%. Except for some graphite dust, the turbomachinery seals are expected to be static seals in a debris-free inert environment and so their failures are assumed to be 1×10^{-7} per hour. Based on these assumptions, the EFOH is 3.4 h per year.

HTS internals consist of the hot duct, elbows and shrouds. Although these are somewhat different in the DC, their EFOH is assumed to be same as that of the SC — 2.2 h per year. Recuperator failures, their frequency, and their effects were not assessed in this report; however, initial discussions with the recuperator vendor indicate that similar components operating in conventional gas turbine cycles have experienced very low failure rates. The latter operate under more demanding conditions in terms of temperature, thermal cycling and coolant chemistry, but with much lower pressure differentials. In addition, recuperator failures leading to immediate forced outages are considered unlikely, since leakage is primarily manifested in the form of efficiency degradation. Overall, the DC Heat Transport System EFOH is assessed to be 79.3 h per year. The lack of a projected contribution from the recuperator must be acknowledged as a deficiency in this estimate, although it is not expected to be large.

Indirect Cycle: The IHX leak rate, due to similar weld failures along with mechanical damage, fretting, and wear failures, was evaluated to be 0.2 per plant year. This is a factor of two smaller than the steam generator tube leak rate. In case of the IHX, it has twice as many tubes as the steam generator and so the mean time to repair of the IHX tube is assumed to be twice that of the SC. Thus, the assessed EFOH is the same as that of the SC — 57 h per year. An assumption inherent in this analysis is that IHX leak-related contamination has a negligible effect on future scheduled or unscheduled outages.

Although the main circulator power went up from the SC to the IDC design, the EFOH is assumed to be the same, 61.8 h per year. Similarly, HTS internals failures are assumed to be the same as that in a SC, 2.2 h per year. Hence, the IDC Heat Transport System EFOH is 121 h per year.

10.2.2.4 Turbomachinery System

With the limited detail available for the Turbomachinery System for both DC and IDC, its EFOH is assessed from the Operational Reliability Analysis Program data base, described in Section 10.1. As noted earlier, the analysis addressed existing modern, base-loaded gas turbines larger than 50 MWe, that are used in both cogeneration and utility applications.

Direct Cycle: The generator unavailability due to load equipment, control/protection, and auxiliaries from the data base amounts to an EFOH of 10 h per year. As the reactor startup times are not included in the gas turbine plant data base, it is assumed that the system EFOH is double that of the data base, equal to 20 h per year. The EFOH due to generator outages internal to the pressure boundary is obtained from the forced outage rate of 0.01 failures per year and an outage time of seven days that it takes to replace the generator with a spare (one day for shutdown, five days for generator replacement, and one day for startup) — an EFOH of 1.6 h per year. The generator coolers are assumed to have the same failure rate as that of SC's main circulator motor heat exchangers and a mean time to repair of nine days, resulting in an EFOH of 6.2 h per year.

For a turbomachinery removal and major overhaul program of once every seven and a half years and one spare, the turbomachinery is removed from one module and replaced in another on the average at 22.5 month intervals. In this analysis, it is assumed that, when a turbomachine is removed from a DC reactor module, it is held in storage for a ten month period to allow for plateout radionuclide decay and then it is overhauled, which takes another two months. If a gas turbine fails it is usually replaced by the spare turbomachine when it is available. If, however, a forced outage occurs while the spare is being held in storage for plateout radionuclide decay, then it is assumed that, with special effort, the spare can be readied for installation in three months. When the spare is available, the outage time is twelve days (one day for shutdown, ten days for gas turbine replacement, and one day for startup). The DC gas turbine failure rate of 0.08 failures per year leads to an EFOH of 97.3 h per year. Rare events such as turbine deblading or disk failures are assumed to contribute very little to the forced outage rate because the frequencies of these events are small and the turbomachinery casings are designed to handle these occurrences at overspeed conditions. Clearly the data base is limited, so a deblading failure rate of 1×10^{-4} per unit year is assumed for the DC turbomachinery system with an outage time of 10,000 hours. While this assumed failure rate is lower than that derived in Section 9.2.2.3, it is believed justified on the basis of the relatively moderate operating environment in the GT-MHR (i.e., lower temperatures, milder transients and reduced potential for corrosion), the elimination of major contributors to failures in conventional machines (e.g., combustor failures resulting in blade damage) and the inherent on-line diagnostics provided by active magnetic bearings. This assumption resulted in an EFOH of 1 hour per year.

The frequency converter's EFOH is estimated from its major components — thyristors, transformers and control systems. It is estimated to be 1 hour per year. So the DC turbomachinery system EFOH is estimated to be 127.1 h per year.

Indirect Cycle: With the limited detail available for the IDC's Turbomachinery System, its EFOH is assumed to be similar to those of the data base, an EFOH of 22.5 h per year. As the reactor startup times are not included in the data base, it is assumed that the system EFOH is double that of the data base, equal to 45 h per year.

10.2.2.5 IDC's Secondary Helium Loop

Helium leaks from the secondary coolant pressure boundary to the atmosphere are assumed dominated by small leaks in valves and bolted seams. Assuming four maintenance block valves

around the bypass valves and two pressure relief valves in the secondary coolant boundary, a total leak frequency of 0.04 per module-year is obtained. The detection and repair time is assumed to be two days, with an additional day to start up the reactor. This results in an EFOH of 2.7 per year. However, due to additional components in the secondary coolant boundary, the EFOH is assumed to be twice the value calculated, equal to 5.4 h per year.

Leaks in the precooler or in the two stages of intercoolers cause secondary helium to leak into the cooling water loops. For very small leakage, the plant may continue to operate since the escaping secondary helium is not contaminated; only for large leakage would the secondary loop be shutdown for immediate repair. Conservatively, the PRA data for coolers is used to estimate an EFOH of 4 h per year. The seal leakage causes bypass flows that reduce the efficiency of the IDC, that is, in effect, a derating. The EFOH is assumed to be the same as that evaluated for the DC — 3.4 h per year. As in the case of the DC, recuperator failures, their frequency, and their effects were not assessed in this report. Again, the contribution to EFOH is expected to be small.

Assuming two normal control bypass valves and the two fast opening bypass valves, and a failure rate of 0.08 failures/valve year with a mean time to repair of two days, one day for replacement of bypass valve and one day for startup, the EFOH is 14.6 h per year. Spurious operation of the four isolation valves results in forced outages. Assuming hydraulic isolation valves with a failure rate of 1×10^{-6} per hour and a repair time of 36 h and an additional day for reactor startup, the EFOH is 2 h per year. Hence, the IDC's secondary helium loop is assessed to have an EFOH of 29.4 h per year.

10.2.2.6 Plant Forced Outage Rate

The forced outage rate over the life of the plant is shown in Table 10-6 for each of the four concepts. The forced outage rate for the steam cycle plant is assessed to be 683.8 h per year or 7.8% unavailability — this is considerably higher than the gas turbine options, and possesses only a small margin from its requirement of 8% unavailability. The differences between the two gas turbine plants is small, because the disadvantage of having the turbomachinery in the main loop (in the DC) is not quite offset by the disadvantages of the intermediate heat exchanger and main circulator (in the IDC). The additional penetrations in the power conversion vessel also cause a small increase in the forced outage rate of the DC due to primary coolant leaks. The difference between the IDC and DC forced outage unavailabilities is essentially explained by these two reasons.

10.2.2.7 Qualitative Uncertainty Comparison

The 450 and 475 MWt SC-MHR are judged to have a "reference" uncertainty on their unavailability of 7.8%. The reference uncertainty is due to system designs that are relatively well established and due to a previous history of developing forced outage assessments providing some confidence in the results. The 450 MWt IDC concept is judged to have a higher uncertainty than the SC concept because: (a) Its system designs are not well established and, with design evolution, additional components may be required that will result in additional scheduled outages; (b) The forced outage evaluation of IDC unique components, most of which are evaluated based on available data, need to be substantiated; (c) The effect of IHX leaks

contaminating the secondary loop and the turbomachinery have been ignored. The 450 MWt DC concept is judged to have a slightly higher uncertainty than the IDC concept because, apart from the uncertainties shared with the IDC, the removal and replacement time for the DC turbomachinery needs to be better substantiated. For both the DC and IDC, the possibility of seal leakages and their effects have only been roughly estimated and recuperator failures have not yet been taken into account.

Table 10-7
MHTGR PLANT OUTAGE ASSESSMENT SUMMARY

Type of Outage	Assessed Unavailability (%)			Utility/User Requirement (%)
	450 & 475 MWt Steam Cycle	450 MWt Direct Cycle GT	450 MWt Indirect Cycle GT	
Scheduled Outage	6.0	4.5	4.5	8
Forced Outage	7.8	6.4	6.2	8
Plant Capacity Factor	86	89 ^(a)	89 ^(a)	84

^(a) Assumes relaxation of 18 month refueling interval, currently specified in Utility/User Requirements.

10.2.3 Capacity Factor

The capacity factor results are summarized in Table 10-7 and compared against the Utility/User requirements. All the concepts meet the overall plant requirements on capacity factor, scheduled and forced outages. The gas turbine concepts are similar and show more availability than the steam cycle concepts. These results are dependent on one key assumption: the reactor core is not limiting, i.e., the fuel can be designed to meet the capacity factors evaluated on an eighteen month refueling cycle. At present, the reactor core design for the steam cycle can meet an 86% capacity factor on an eighteen month refueling cycle. The core designs for the gas turbines are yet to be developed. A judgment on the uncertainty of these capacity factors shows the DC to have a slightly higher uncertainty than the IDC, which has a higher uncertainty than the SC.

If the reactor core for the gas turbine concepts can only meet an 86% capacity factor with an eighteen month refueling cycle, then the plant capacity factors for the DC and IDC would also drop to 86%. One alternative is to design the core to achieve 86% capacity factor on an eighteen month refueling cycle. Another alternative is to drop the eighteen month refueling cycle requirement — then the gas turbines can achieve higher capacity factors. By going to 27 refuelings with a refueling cycle of ~17.25 months, instead of the usual 26 on an eighteen month refueling cycle, the gas turbine concepts will achieve at least the same equivalent full

power months with only a 0.1% increase in its scheduled outage unavailability. This means that 89% capacity factors for the gas turbines are achievable by relaxing the eighteen month refueling cycle requirement.

A sensitivity analysis of the DC turbomachinery removal and replacement time indicates that the assumed time of 10 days can be extended to 15.5 days without increasing the DC scheduled outage unavailability. Increasing removal and replacement time to twenty days would only marginally increase the scheduled outage unavailability so that it remains at 4.5%. However, the forced outage unavailability would increase by 0.1% for a 20-day removal and replacement time for the DC turbomachinery.

A sensitivity analysis of the IDC turbomachinery inspection time indicates that the assumed time of a five day inspection every odd-numbered refueling and a twenty day inspection every even-numbered refueling can be extended to a 14.5 day inspection every odd-numbered refueling and a 26.5 day inspection every even-numbered refueling without changing the IDC scheduled outage unavailability.

REFERENCES

- 10-1 "Availability and Reliability Assessment: Existing Gas Turbine Technology," Report Number 2609301, prepared by Strategic Power Systems, Inc.®, Albany, New York, March 26, 1993.

SECTION 11

MAINTAINABILITY

The maintainability of the direct cycle (DC) and indirect cycle (IDC) gas turbine designs was assessed in comparison with the reference steam cycle (SC) concept. First, the impact of higher steady-state fuel temperatures on system plateout and occupational doses were evaluated for the gas turbine concepts and compared with the reference SC plant. Second, maintenance considerations were addressed for key components that are either unique to the DC or IDC, or which differ significantly from their steam cycle counterparts. Finally, a qualitative summary of maintainability differences was made, based on the above. The results are summarized in the following subsections.

11.1 OCCUPATIONAL DOSE

The impact of higher steady-state fuel temperatures on occupational doses was evaluated for both of the 450 MWt gas turbine concepts. The bases for this comparative evaluation are the results of prior 350 MWt SC-MHR plateout estimates and associated estimates of operation and maintenance (O&M) doses.

The best estimate occupational dose for the 350 MWt SC-MHR is 139 man-rem/year, which is less than the user goal of 154 man-rem/year. The user goal is based on meeting 10% of the 10CFR20 requirement of 5 man-rem/year, for a staff of 308. The important tasks which contribute to the 350 MWt SC O&M doses are identified in Table 11-1. The tasks are grouped based on the work area (e.g., reactor cavity or steam generator cavity) in which they are performed. Tasks performed in the reactor cavity and steam generator (SG) cavity contribute 60% and 16% of the total dose, respectively. These same tasks, reactor and SG vessel inspections and leak tests, support member surface exams, and visual exams, will also be performed for the gas turbine concepts.

The major source for the reactor and SG cavity doses is plateout of Ag-110m, Cs-137, and Cs-134, with half-lives of 0.8, 30, and 2 years, respectively. For the helium purification train area, the major source of dose is Kr-88, with a half-life of 3 h, which does not plate out in the primary circuit. The 350 MWt SC O&M doses in the reactor and SG cavities can be scaled, based on the relative plateout, to estimate the 450 MWt SC and gas turbine doses. Doses in the helium purification train area can be scaled by the relative amounts of krypton.

The 450 MWt SC doses have been estimated by scaling the 350 MWt SC doses by the power ratio to account for a larger nuclide inventory and, accordingly, more plateout at the higher power level. The fractional release of radioactive nuclides from the fuel is assumed to be the same as in the 350 MWt SC, since the steady state fuel temperatures are similar. The resultant 450 MWt SC dose is [179] man-rem/year which exceeds the user goal of 154 man-rem/year. However, the 350 MWt SC doses, from which the 450 MWt SC are scaled, are judged to be conservative, based on assumptions made in the early design stage. Also, it is likely that more remote maintenance will be practiced than assumed for the 350 MWt SC.

Table 11-1
SUMMARY OF 350 MWt SC O&M DOSES

Work Area	Dominant Tasks	Dose ^(a) (man-rem/year)	Major Source
Reactor cavity	Bolted seam and weld inspections, leak tests, surface and visual exams, O-ring and seal replacement	84 (60%)	Ag-110m, Cs-137, Cs-134 plateout
SG cavity	Weld inspections, leak tests, surface exams, torque bolts, tube plugging (4 man-rem/year)	22 (16%)	Ag-110m, Cs-137, Cs-134 plateout
Helium purification train area	Process equipment, compressor, regeneration equipment	11 (8%)	Kr in helium purification train
Other	Miscellaneous	22 (16%)	Based on reactor zone dose
Total		139 (100%)	

^(a)User goal is 154 man-rem/year.

The gas turbine plateout for each of the three dominant nuclides can be similarly estimated by scaling the 350 MWt SC plateout by the power ratio, a release factor, and a plateout distribution factor. Of these factors, the release factor is dominant and is mostly due to increased metallic nuclide release from standard particles, as the result of the higher gas turbine steady-state fuel temperatures. The plateout surface concentration for each nuclide is weighted by the nuclide's gamma energy and then summed. The resulting dose factor is the ratio of this weighted plateout concentration for the gas turbine, relative to the SC.

Overall, the DC and IDC dose factors are estimated to be 10 times that of the 450 MWt SC at the same power. Dose factors related to tasks performed in the turbine (DC) or IHX (IDC) and reactor cavities are estimated to be a factor of 40 and 10, respectively, higher than the doses in the 450 MWt SC. The gas turbine dose factors are less than previously estimated, due to revision of the time dependent fuel temperature distribution assumptions. The relative dose factors for the gas turbine and SC designs are summarized in Table 11-2, with the 450 MWt SC dose as reference. Additional, detailed plateout and dose calculations are needed for the gas turbine concepts.

The maintenance approach selected for the DC turbomachine begins with removal and replacement with an on-site spare, using remote handling techniques and/or appropriate shielding. Inspection and repair of the removed turbomachine is then accomplished in an

Table 11-2
RELATIVE DOSE FACTORS^{(a)(b)}

	450 SC	475 SC	450 DC	450 IDC
SG/IHX/turbine cavity	1	1.06	40	40
Reactor cavity	1	1.06	10	10
Helium purification	1	1.06	5	5
Turbine maintenance	Best	Best	Worst	Best
Overall	1	1.06	10	10

^(a)Numbers in table are relative to the 450 SC. For example, the 450 DC overall dose factor is estimated to be 10 times larger than the 450 SC dose factor.

^(b)Dose factors are less than previous estimates due to time dependent fuel temperature assumptions.

appropriate maintenance facility. On this basis, the dose factor related to DC turbine maintenance is judged to be small compared to dose factors related to the tasks which must be performed in situ in the reactor and SG/IHX/turbine cavities. However, the fact that the turbine is in the primary coolant loop in the DC makes it somewhat less attractive than the IDC in terms of occupational dose.

Overall, the two SC concepts are considered the best in terms of the relative difficulty in meeting the user occupational dose requirements. The gas turbine designs are judged to pose a greater challenge due to the increased plateout resulting from the higher steady-state fuel operating temperatures. It is clear that a greater degree of remote handling during maintenance and in-service inspection tasks will be an important consideration in the further development of these concepts.

11.2 MAINTENANCE REQUIREMENTS FOR KEY COMPONENTS

The maintenance requirements of key components were assessed for those instances where the components are unique to the gas turbine designs or where their characteristics or requirements differ significantly from the steam cycle.

11.2.1 Circulator Maintenance Considerations

Inherent in the IDC plant design is the ability to remove and replace the main and shutdown cooling circulators. The main circulator is readily accessible at the top of the IHX vessel. The SCS circulator, which is common to all of the concepts, is installed in the lower end of the reactor vessel. In a study performed on the steam cycle plant (by GA, ABB-CE, ORNL, and BGI), details of a procedure (together with the identification of the required equipment) to remove and replace this component were documented.

The circulators will have comprehensive diagnostic systems which will give on-line monitoring of the machine health. The main circulator will be removed and refurbished at 8 year intervals. Spare circulators will be kept at the plant site to minimize down time. The estimated time for circulator removal and replacement is 5 days.

Studies performed in the past have been based on pulling the circulators into a shielded cask which would then be moved into the circulator maintenance facility. Depending on the actual dose rates (yet to be definitively computed) it may be possible to pull the units into a less substantial shielded arrangement (e.g., plastic bags used for the AGR machines in U.K. plants).

Following decontamination of the impeller, by steam and water washing, it is projected that direct hands-on maintenance will be possible. This is substantiated by Fort St. Vrain experience where a circulator was removed after 60,000 h of operation, and after water washing of the impeller, the machine was disassembled by hand.

Following the practice adhered to in the AGR plants, the circulator maintenance facility would be located at the plant site. With the same maintenance crew available, the learning curve would be rapid. During refurbishment, any faulted parts would be replaced and, in addition, the bearings, seals, and instrumentation would be replaced.

Advantage would be taken of the successful AGR program in terms of circulator maintenance facility design and operation. James Howden and Company have produced 112 circulators for the AGR plants and these have accumulated over 7.5 million hours of operation, with recent availability of over 99%. Because of good performance the removal/replacement interval for these machines is being increased from 8 to 12 years.

Circulator refurbishment time is 6 weeks, but this could be reduced to 3 to 4 weeks if double shifts were scheduled.

While the IDC circulator has a higher power rating, the expectation is that in a mature plant its performance/availability/maintainability would match the excellent record of the current AGR machines.

11.2.2 IHX (IDC)

Although the primary helium entering the IHX is at a high temperature, it is a rather benign, inert gas, which is not expected to cause any corrosion related problems on the primary shell

side of the IHX. Likewise, because the secondary medium also is helium, no corrosion related problems are anticipated within the tubeside of the IHX.

Fretting and wear between the tubes and tube supports, due to flow-induced vibration, are potential damage mechanisms because of the high flow velocities. Damage mechanisms related to the high temperatures involved, in conjunction with thermal cycling effects (e.g., creep, low cycle fatigue) are also of concern. However, experience shows that proper design can avoid or limit the effects of these phenomena. Special attention will be devoted to these subjects during IHX design.

The tube bundle is designed with individual tube circuits from the secondary inlet tubesheet, through the helical tube bundle, to the cylindrical tubesheet at the top of the central downcomer. Access to both of these tubesheet areas is available for inspection, leak detection and tube plugging, as necessary. Access to the secondary inlet tubesheet is gained through a manway in the secondary inlet plenum at the bottom of the IHX vessel. Access to the cylindrical tubesheet, at the top of the central downcomer, can be gained by removing the secondary piping 90-degree elbow at the bottom of the IHX vessel. A special, telescoping platform, placed below the opening, will provide access up the downcomer for robotic inspection or maintenance equipment, or for personnel, if the radiation field is not too severe.

11.2.3 Precooler and Intercooler (DC/IDC)

The precooler and intercooler are helium-to-water heat exchangers that, in the DC design, cool the primary helium at the entrance to the first stage and second stage compressors, respectively. In the IDC, two intercoolers are provided, one between the low speed compressor and the first stage of the high speed compressor, and a second between the high speed compressor stages. Normal operating temperatures are low [tube temperatures less than 100° C (212° F)]. The tube side (water side) flow remains in the liquid phase and, because the water is on the tube side, there are not any low flow regions or crevices where corrosion products can accumulate. In both the DC and IDC, the tube bundle arrangements are such that the tubes are drainable.

In the DC design, the precooler and intercooler are located in the lower end of the power conversion vessel, and their replacement would involve considerable difficulty. Given the relatively mild operating conditions, the presence of helium on the shell side and the provision of an isolation cooling loop (that facilitates chemistry control) on the tube side, the need for total replacement of these heat exchangers within the plant lifetime is evaluated to be remote. A helically coiled tube design has been selected for these heat exchangers, and individual tube circuits are incorporated into the designs, routed from the cooling water inlet tubesheets to the outlet tubesheets. Access to the tubesheets is available, from outside of the power conversion vessel, for in-service inspection and for leak detection and tube plugging, if required.

In the IDC, the precooler and intercoolers are placed in individual shells and are readily accessible for inspection, repair and replacement, as necessary. Given their location in the secondary helium circuit, the present design does not include an isolation cooling loop. Accordingly, chemistry control on the water side would be a more difficult issue, and a greater degree of maintenance may be required as a result.

11.2.4 Recuperator (DC/IDC)

Both the DC and IDC employ compact plate-fin recuperator designs. There are not major differences between the direct and the indirect cycle requirements for recuperator maintenance. Corrosion and deterioration of surface cleanliness are not considered problems for this application and the recuperators are not expected to require routine inspection or maintenance. For both plants, inspection, testing and repairing of any recuperator module will only be performed after diagnostic analyses have determined that a module is leaking and that its removal/repair is necessary. In the case of the DC, the diagnostic analyses will also be used to determine if the leaking module should be replaced by the spare, or if it is more economical to repair it.

In the case of the DC, it is assumed that the recuperator must be removed for repairs, or simply replaced. Provisions have been made in the design to remove the recuperator modules through the top of the power conversion vessel. Access to the recuperators is obtained by first removing the generator, top head of the power conversion vessel, the turbomachinery cartridge and the internal dome separating the turbomachine and generator cavities. The recuperator seals are then separated and the recuperator module is lifted out of the power conversion vessel and replaced with a new unit. Because the levels of radiological contamination may be high, removal and replacement of the recuperator are assumed to require remote handling techniques. Repair of the recuperator, if practical, would be accomplished in an appropriate maintenance facility.

In the case of the IDC, the design is intended to provide for removal and replacement of any one recuperator core with a minimum of cutting and rewelding of pressure retaining components. To remove a core from the vessel the following steps would be required. Sections of the high pressure piping would have to be cut to allow removal of the end plate. The expansion joint connecting the pipe to the end plate would have to be cut off the pipe projecting from the core. The shell would be cut circumferentially between the end plate and the low pressure nozzles. The end plate and the shell portion attached to it would be removed along with the high pressure piping sections cut out. Finally the thermal shield would be cut away from the core. The core could then be removed. The replacement core would be installed in a similar manner, with the steps being reversed.

After removal, location of leaking joints is accomplished through a leakage test. This test is performed by pressurizing the cold side (high pressure side) of the module with nitrogen, and recording the pressure decay in a certain time interval (e.g., 15 min). The leakage rate can be calculated from the cold side internal volume, the initial and final pressures, the nitrogen temperature and the decay time. For leakage rates exceeding a certain value, locating and weld plugging the leaks would be necessary. Locating the leaks could be accomplished by listening to the sound of escaping gas around the inlet and outlet areas, the manway manifold, and around the heat exchanger core brazed edges (internally and externally).

Outside leaks in the heat exchanger core brazed edges (on the cold and hot sides) are repaired using gas-tungsten arc welding (GTAW). Internal leaks between the high pressure side and the low pressure hot side may require repair by sealing off the entire high pressure cold passages within a section of the recuperator core (see Figure 11-1). The recuperator core is

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Figure 11-1 INTERNAL LEAK REPAIR WELDING

composed of a number of separately brazed sections (of approximately 30 in. height) which are welded together with appropriate fittings. When the leaking high pressure passages are away from the top or the bottom of the recuperator core, it will be necessary to split the core into two sections by sealing the leaking section (approximately 18% of the recuperator heat exchanger core will be lost).

11.2.5 Turbomachinery

11.2.5.1 Direct Cycle

A detailed maintenance study was not undertaken, but turbomachinery maintenance is recognized to be a major consideration in the direct cycle because of the assumed necessity for remote handling. Pending dose rate estimates of the levels of activity in the machine, particularly in the turbine area, the exact maintenance procedure cannot be defined. Suffice to say that the machine will be contaminated and is assumed to require a shielded cask (or bag/tent) to facilitate movement from the vessel location to the maintenance facility. At this stage it is being assumed, that after decontamination, hands-on operations for disassembly will be possible.

As in the case of the main helium circulator in the other MHR plant variants, it is planned to have a spare turbomachine at the plant site. The current interval for planned turbomachine removal/replacement is 7.5 years (approximately 60,000 h running time) and was selected so as to be compatible with planned refueling operations. The estimated time to remove/replace the turbomachine is 10 days, this value being twice that required to remove/replace the AGR circulators.

The turbomachine maintenance facility could be located at either the plant site or the suppliers factory. During machine refurbishment, the following items would likely be replaced: (1) turbine blading, (2) catcher bearings, (3) seals, and (4) instrumentation/diagnostic systems. The estimated time for machine refurbishment is 12 weeks; again the value assumes a time twice that required for an AGR circulator.

11.2.5.2 Indirect Cycle

For the indirect cycle, maintenance planning would not have to deal with the access or personnel dose issues that are found with the direct cycle. Normal maintenance could be done in-situ, and removal of the rotor would only be required for major maintenance. A detailed maintenance planning assessment would be required to determine if a spare turbomachine provides any significant reduction in down time in a given plant or if shared among several plants.

11.2.5.3 Maintenance Technology Base

Clearly, the aforementioned projections are speculative, and will be refined when maintenance studies are performed for the direct and indirect cycle plants. Prior to this being undertaken, advantage will be made of the following databases: (1) decontamination experience for the Fort St. Vrain helium circulators, (2) data from the German HHT program (particularly

involving turbine blading decontamination), (3) maintenance facility experience from the AGR circulator program in the U.K., and (4) experience from industrial and aircraft gas turbines.

The Turbomachinery Reliability Assessment, discussed in Section 10.1, concluded that the simple-cycle plant unavailability is dominated by the scheduled outage factor (SOF). This total SOF value is dominated by the contribution from the gas turbine (flange-to-flange) main systems. This value, in turn, is dominated mainly by the turbine system, with major contributions by the hot gas path inspection and major overhaul activities. The combustion inspection activity for the gas turbine combustion system is a second major contributor to the scheduled outage factor.

The maintenance intervals for major maintenance activities for heavy-duty, open-cycle combustion turbines depend on the manufacturers and the fuel type. Table 11-3 provides the recommendations of two manufacturers for natural-gas-fired and (diesel, crude, or heavy) oil-fired units. Manufacturer A recommends the same interval for both fuel types and would result in a SOF of, at least, 1.6%. Manufacturer B recommends a more frequent flange-to-flange inspection than manufacturer A for the gas-fired units, resulting in a minimum SOF in the range 2%-3.5%. For the oil-fired units, Manufacturer B recommends more frequent inspections than for its gas-fired units with projected minimum SOF in the range of 4%-6%.

Inspection A can be characterized as a small inspection with focus on the combustion system, concentrating on the combustion liner assemblies and fuel nozzles. The combustion chamber and its contained components, the fuel nozzles, and combustion liners are removed and visually inspected for wear, hot spots, cracks, and condition of welds; parts are replaced as required. The fuel nozzles and liners are cleaned and/or replaced with new or refurbished parts. The hot gas path to the turbine first-stage nozzle and buckets are visually or boriscopically inspected for wear and deterioration. This inspection guides the schedule for the hot-gas-path inspection (Inspection B). The exhaust gas system is also inspected. In addition, leakage checks on the fuel system, the lube oil system, and cooling/air-sealing systems are made, as well as inspections of the temperature controls and temperature protection, overspeed protection, and flame monitoring systems.

Inspection B, characterized as normal inspection, includes the scope of the combustion inspection and an inspection of the hot-gas-path. The turbine first-stage nozzles are removed, cleaned, and inspected for erosion, corrosion, cracks, trailing-edge bowing, and cooling-hole plugging. These nozzles could be replaced with new or refurbished ones allowing cleaning and inspection at a service facility and minimizing downtime. The first-stage buckets and the second-stage nozzles and buckets are visually inspected for damage, erosion, corrosion, and visual cracks. Bucket tip clearances are measured and recorded. The compressor forward section is inspected through the inlet plenum, and the aft end blading is inspected through the combustion wrapper. The auxiliary and fuel systems are checked for function as well as leakage. All controls and turbine protection/monitor systems are checked.

Inspection C is the major maintenance inspection. The scope includes the scope of Inspections A and B, but in this case the upper half casing and bearing covers are removed and both rotors are pulled. The turbine first-stage buckets are removed, cleaned, and non-destructively examined. The turbine wheel is cleaned and examined for cracks. The bearing seals and liners are cleaned and inspected for wear, fouling, leaks, scoring, and deterioration. Compressor

Table 11-3
Recommended Maintenance Intervals
for Heavy-Duty Open-Cycle Combustion Turbines

	Recommended Interval (fired hours)		Estimated Time to Perform Maintenance
	Oil-fired	Gas-fired	Downtime Hours
Manufacturer A			
Interval			
A Combustion Inspection	8000	8000	80
B Combustion/Turbine Inspection	24,000	24,000	160
C Flange-to-flange Inspection	48,000	48,000	320
Manufacturer B			
Interval			
A Combustion Inspection	4000	8000	90-120
B Combustion/Turbine Inspection	8000	16,000	120-140
C Flange-to-flange Inspection	16,000	24,000	300-600

blades are cleaned, and the blades and stators are inspected for damage, erosion, corrosion, and fouling. Inlet guide valves are non-destructively examined for cracks. Journals and seals are inspected for wear and scoring. Auxiliary and fuel system pumps, motors, and coolers are checked, as well as the leak and functional testing of the A and B inspections. All controls and turbine and generator monitoring/protection systems are checked out. The generator, its auxiliaries, excitations, and starter are also inspected at this major overhaul.

For the GT-MHR application, none of the combustor related inspections apply. However, turbine first-stage nozzles and buckets and hot-gas-path inspections must be performed. But, since the helium working fluid is much more benign than combustor gas and the temperatures of operation are significantly lower, a longer inspection interval of 7.5 years has been judged to be feasible, thereby minimizing the scheduled outage factor and unavailability associated with the turbomachinery.

11.2.5.4 Turbomachinery Servicing Equipment

There is a differential service equipment requirement in the Direct Cycle relative to both the Indirect and the Steam Cycles. The requirement is for special equipment and provisions to facilitate the removal and installation of the turbomachine. At this time, the requirement is not defined in terms of quantified parameters. However, depending on contamination levels on turbine surfaces, several scenarios may be envisioned for accomplishing this function. The worst case is that a steel cask may be needed with built-in lifting and alignment mechanisms to remove the turbomachine and provide radiation shielding. The disadvantage of this concept is that the cask would be heavy, probably in the 200 to 500 ton range, and costly. In addition, a substantial building crane would be required.

A more likely case would be the use of lifting and alignment mechanisms in a collapsible (folding) tent to retain particle contamination. Based on the relatively low levels of primary system contamination experienced at Fort St. Vrain and the improved performance specified for the MHR fuel system, the feasibility of this option is very real. As a possible variation, if necessary, narrow metal shielding belts may be used for turbine stages with high contamination levels in conjunction with a collapsible tent.

In addition, as previously noted in Section 5.10.2, removal of the turbomachine for maintenance leaves the hot duct and the cold helium return annulus in the cross vessel completely open to each other at the power conversion system end. This creates a coolant flow path of very low resistance around the core, which renders the SCS ineffective for core cooling during refueling. This consideration results in the need to provide a solid or inflatable plug at the hot duct mouth and an appropriate mechanism to remotely insert it. The design of the plug must take into account removal and replacement of the turbomachine, recuperator modules and other power conversion system internals, as well as seal maintenance. The specific requirements for these capabilities and the design of the plug and insertion mechanism must be addressed in the course of follow-on design and development.

In summary, the service equipment can be expected to be more complicated in the direct cycle. Remotely controlled lifting and alignment mechanisms for the removal and installation

of the turbomachine and a hot duct plug will be needed. Special radiation containment and shielding for personnel protection may also have to be included around the mechanisms.

11.3 OVERALL MAINTAINABILITY ASSESSMENT

Since the reactor system is essentially the same for all configurations, plant maintainability tradeoffs are focused on turbomachine, ducting, piping and related heat exchanger maintenance considerations. The two gas turbine concepts appear to be superior to the steam cycle concept because of the elimination of a substantial number of steam and water systems that are known to require significant maintenance efforts in steam and water plants. Between the two gas cycle concepts, the direct cycle has the advantage of superior simplicity: the number of static and moving components is significantly lower in the direct cycle.

The indirect cycle does have the advantage of having the turbomachine and supporting heat exchangers outside of the primary coolant circuit where they are more easily accessible. However, it still has a primary coolant circulator and a large intermediate heat exchanger in the primary circuit.

In the final analysis, the direct and the indirect cycle concepts are comparable. However, a case can probably be made that direct cycle simplicity outweighs indirect cycle accessibility, particularly if special accessibility provisions are incorporated in the design of the direct cycle.

11.3.1 Direct Cycle

Other than the reactor system, which is essentially common to all concepts, maintenance requirements in the direct cycle plant include surveillance and maintenance of the turbomachine, recuperator, precooler, intercooler and bypass valves, as well as inspection of the power conversion system vessel.

The level and distribution of radionuclide contamination represent the greatest uncertainties that must be accommodated in the maintenance design of the DC. Prior gas cooled reactors, such as Peach Bottom and Fort St. Vrain in the U.S., AVR and THTR in Germany and the AGR reactors in the U.K, have exhibited low levels of machine contamination, compared to light water reactors. The fuel specification for the GT-MHR would imply even lower levels of contamination, other factors being equal. However, the increased normal operating temperatures of the GT-MHR imply a greater degree of contamination than that which would be expected for the SC. Further, most of the contamination would be expected to be found in the turbomachine or recuperator. Given these uncertainties, the design of the DC has evolved on the basis that remote handling and maintenance of the turbomachine and recuperator would be required.

The maintenance approach for the turbomachine includes on-line monitoring of machine performance via magnetic bearing controls, plus removal, inspection and replacement every 7-1/2 years. A spare turbomachine is available at the plant site for this purpose. Features are included to facilitate replacement of turbine blading, catcher bearings, seals and diagnostic instrumentation. The estimated time for turbomachine removal and replacement is 10 days, which fits within the refueling time. The estimated time for turbomachine refurbishment is 12 weeks.

Consistent with the above, and as discussed in Section 11.2.5.4, the turbomachine is removed utilizing remotely controlled lifting and alignment mechanisms in a cask or in a collapsible (folding) tent, possibly with special purpose metal shielding belts to retain particle contamination and provide shielding around specific machine areas. Although significant details need to be developed on the removal and installation of the turbomachine, the risks associated with turbomachine maintenance are judged to be acceptably low. The generator, which is not expected to be contaminated, would be removed separately.

The maintenance approach for the recuperator also includes on-line monitoring of heat exchanger performance. Although replacement within the plant lifetime is not planned, this capability is provided on the basis of remote handling techniques similar to those used with the turbomachine. However, replacement of the recuperator is made easier by the fact that they are extracted as separate modules of much lighter weight than the turbomachine unit.

The maintenance approach for the precoolers also includes on-line monitoring. In addition, it is anticipated that the cooling tubes would be inspectable and pluggable, if necessary, at tube sheets directly accessible from outside the power conversion vessel. Since there is no boiling in these heat exchangers, they are expected to be significantly more reliable and require significantly less maintenance than steam generators.

The maintenance approach for the bypass valves also includes on-line monitoring of performance, including ultrasonic emissions analysis. The valves are located outside the turbomachine vessel where they are directly accessible. Based on Fort St. Vrain experience, hands-on maintenance will probably be feasible. Nevertheless, casks and remote control mechanisms will be available for removal and installation.

Inspection of the power conversion system vessel is expected to be similar to the inspection of the intermediate heat exchanger vessel in the indirect cycle, and similar to the inspection of the steam generator vessel in the steam cycle.

11.3.2 Indirect Cycle

Maintenance plans for the indirect cycle include the same functions as in the direct cycle plus more. This is discussed in the following paragraphs. However, in general, some of the functions that are similar are more easily accomplished in the indirect cycle because the components involved are not exposed to contamination. On the other hand, some of the additional indirect cycle maintenance functions do include components that are exposed to contamination, particularly the intermediate heat exchanger and the primary coolant circulator.

Consider first the maintenance functions that are similar to the direct cycle functions. These include surveillance and maintenance of the turbomachine, recuperator, precooler, intercooler and bypass valves, as well as inspection of the power conversion system vessel.

The maintenance approach for the turbomachine is similar to the direct cycle, in that it includes on-line monitoring of machine performance, and periodic inspections: there is a 5-day inspection and a 20-day inspection, which are alternatively performed concurrently during scheduled refueling operations. However, since no machine contamination is expected, these

are hands-on type inspections following opening of turbine casing. No spare turbomachine is maintained at the plant site.

The maintenance approach for the recuperator, precoolers, intercoolers, and bypass valves is also similar to the direct cycle. However, as in the case of the turbomachine, no needs are anticipated for special shielding.

Inspection and maintenance of the intermediate heat exchanger vessel in the indirect cycle is expected to be similar to the inspection and maintenance of the power conversion system vessel. Although the equipment housed is different, the vessels, the level of inspection, and the expected contamination levels are comparable.

Major additional maintenance functions are needed in the indirect cycle. These are related to the primary coolant circulator, the intermediate heat exchanger, the intermediate heat exchanger isolation valves on the secondary coolant side, the secondary piping, the high pressure turbine compressor, the secondary helium purification system (discussed in Section 5.10.4), and the separate recuperator, precooler and intercooler vessels.

The maintenance approach for the primary coolant circulator includes on-line monitoring of machine performance, plus removal, inspection and replacement every 8 years. A spare circulator is available at the plant site for this purpose. The circulator is removed utilizing remotely controlled lifting and alignment mechanisms into a collapsible (folding) tent, possibly with special purpose metal shielding belts to retain particle contamination and provide shielding around specific machine areas. This is similar to the method used to remove the turbomachine in the direct cycle. However, because of the smaller size and lighter weight of the circulator, removal and replacement is expected to take a shorter time of 5 days, based on experience with AGR circulators.

The maintenance approach for the high pressure turbine compressor is similar to the approach for the turbomachine. However, inspection and maintenance should take less time because of the smaller size and lighter weight.

The intermediate heat exchanger is subject to the same type of inspection requirements as the steam generator in a steam plant, since it constitutes the primary coolant boundary. Although the intermediate heat exchanger is expected to require less maintenance than a steam generator, because it is not subject to corrosion, it is a major component that will have some level of contamination and will require special attention.

The intermediate heat exchanger is expected to have isolation valves on the secondary coolant side to provide isolation for maintenance and tube leak containment purposes. Although no unsurmountable problems are anticipated in the surveillance and maintenance of these valves, the large diameters involved (on the order of 4 ft) will make the process time consuming.

The surveillance and maintenance approach for the secondary loop helium purification system is a simplified version of the surveillance and maintenance associated with the primary loop helium purification system. Since the secondary system is not expected to deal with

appreciable radionuclides, a hands-on approach is acceptable. Nevertheless, the system will require the usual care associated with low temperatures and chemical processing components.

The rest of the major additional maintenance functions needed in the indirect cycle are related to the separate recuperator, precooler and intercooler vessels. No special difficulties are expected in performing these functions. Nevertheless, as pressure vessels, they require periodic surveillance and occasional maintenance.

SECTION 12

PROSPECTS FOR FUTURE TECHNOLOGICAL GROWTH

The high temperature capability of the MHR offers a unique potential for technological growth beyond the level of the applications evaluated in this report. This potential is summarized in the following subsections, which address enhanced electricity production and other advanced reactor applications respectively.

12.1 ENHANCED ELECTRICITY PRODUCTION

The direct cycle (DC) and indirect cycle (IDC) gas turbine and the steam cycle (SC) designs were evaluated for "Growth Potential" in terms of enhanced electricity production. The baseline case for each design was the 450 MWt design. Specifically, each design was evaluated for the potential of expansion to a higher power level, while staying within the physical envelope of the baseline 450 MWt reactor vessel. All other components would be expanded to meet the new core power output level up to the highest power level feasible. In each case, potential feasibility limits, if any, were identified. The major systems and components evaluated were the reactor, circulator (in SC and IDC only), reactor vessel, steam generator (in SC only), and intermediate heat exchanger (in IDC only).

In each of the plant designs, the reactor system is basically the same. However, in the DC and IDC designs, the reactor is operated at higher core inlet and core outlet temperatures, compared to the SC design: 490°C (914°F) versus 288°C (550°F) and 850°C (1562°F) versus 704°C (1300°F), respectively. The baseline 450 MWt core design uses a 84 fueled column core design at 6.0 W/cc power density. The feasibility of expanding this design to 102 fueled columns at 6.6 W/cc, thereby reaching a power output of 600 MWt, was investigated. The 102 fueled core column layout has the same total number of core (total of fueled and unfueled) columns as the 84 fueled core column design. The 84 column core can be expanded into the 102 column core by removing one ring of outer graphite reflector blocks and moving the annular fueled ring out one row. This results in a larger diameter inner reflector. The plan views of the 84 and 102 fueled column cores are shown in Figure 12-1 for comparison.

Reactor physics calculations were performed using the GAUGE computer code, and the 102 column core at 6.6 W/cc was found to be radially and azimuthally stable using conservative assumptions. The axial stability was estimated to be similar to the 84 column 450 MWt design. Because of the increased fuel loadings and number of fueled columns, the 102 column core requires 36 operating control rods in the outer reflector, 12 startup controls rods in the core and 18 reserve shutdown control equipment channels. With this control rod configuration, hot and cold shutdown requirements are met. Refueling shutdown margin requires further study and may show the need for six additional startup rods in the inner reflector. Because of the increased fuel loading for 6.6 W/cc, the maximum core capacity factor is 84% to 86% for an 18 month fuel cycle. The core operating temperatures at 600 MWt are comparable to the corresponding design at 450 MWt since the core inlet and outlet temperatures are kept constant and the core helium flow rate is increased proportionally.

102

27

84

27

FUEL COLUMNS

REACTOR VESSEL
FLANGE DIAMETER
(FEET)

Figure 12-1 84 COLUMN VERSUS 102 COLUMN CORES

Both pressurized and depressurized conduction cooldown (PCC and DCC) temperatures were investigated for each design. In the 600 MWt DC, operating control rods (located in the outer reflector) slightly exceed the maximum operational temperature [1052°C (1925°F) versus the 1024°C (1875°F) limit] during PCC events. This would require inspection or replacement of these control rods. Various other design alternatives are possible to either lower expected PCC temperatures or to increase the temperature tolerance of the operating control rods. All IDC, DC and steam cycle designs up to 600 MWt meet all other PCC and DCC component temperature requirements, assuming the steam cycle design uses the 9Cr-1Mo-V vessel, which has higher strength at PCC temperatures than the standard 450 MWt SC vessel made of SA-533.

Removal of the one row of outer reflector blocks increases the neutron dose to the vessel and core barrel. It is estimated that the neutron fluence at the vessel sidewall will increase by a factor of 30 over the 84 column design, but will still stay below 10^{19} n/cm². It is assumed that 9Cr-1Mo-V steel alloy is used for all the concepts. Some data exists that indicates that 9Cr-1Mo-V retains adequate fracture toughness properties at fluences up to 10^{22} n/cm² at DC and IDC vessel temperatures. It is therefore assumed that the increased neutron fluence on the vessel is not a feasibility problem for the DC or IDC designs with a 102 column core at 600 MWt. For the steam cycle, however, it remains to be confirmed that adequate fracture toughness properties are retained at the lower vessel temperatures. If a problem is encountered, one possible option is to increase vessel neutron shielding by the addition of beryllium oxide ceramic rods within the permanent side reflectors.

The top-head forging for the 102 column core design is larger than for an 84 column core design because of the increased number of control rod penetrations. The vessel top-head forging manufacturing capability for 102 column core designs was investigated. Forgings in the required sizes are a challenge to existing manufacturing capabilities, but are feasible. Also, welding of smaller forgings to meet the required sizes is possible.

In summary, reactor core growth potential to 102 column core designs and 600 MWt appears very feasible. Reactor vessel neutron fluence properties at DC and IDC temperatures appear to meet strength requirements. Vessel neutron fluence properties at lower steam cycle temperatures require further confirmation, but are assumed to be adequate.

The required circulator power was investigated for the steam cycle and IDC designs for up to 600 MWt core power levels. The required circulator power was calculated at the design point and at the maximum required point, based on the thermal performance requirements methodology used in the baseline 450 MWt design. The results of this estimate are summarized in Table 12-1. The estimated upper feasibility limit for circulator power size is estimated at 18 MWe. Therefore, it is concluded that circulator power requirements would limit the IDC design to less than 600 MWt.

The steam generator required for a 600 MWt steam cycle plant and an intermediate heat exchanger required for a 600 MWt IDC plant were considered. It was determined that, even though the sizes required were beyond what presently has been designed, there are no fundamental feasibility issues in designing heat exchangers to that size.

Table 12-1
REQUIRED CIRCULATOR POWER INPUT

	Circulator Power Input (MWe)			
	450 MWt		600 MWt	
	Point Design	Maximum Required	Point Design	Maximum Required
SC	4	5	[<9]	[<9]
IDC	10	13	16	21

Assumption: 18 MWe limit on two motor configuration.

Conclusion: Circulator power limits IDC growth potential.

The potential for increasing plant efficiency was evaluated for the DC, IDC, and SC concepts. Since the SC plant uses the indirect Rankine cycle, its efficiency is limited by practical turbine inlet temperatures and turbine and main steam piping materials. As shown in Figure 12-2, the SC plant main steam temperature is near the practical limits of turbine inlet temperature. Therefore, the SC plant efficiency at 38% is also near the practical limit. The DC and IDC plant use the direct and indirect Brayton cycle. HTGR gas outlet temperatures have been demonstrated to 950°C (1742°F) and gas turbine machine materials have been demonstrated to even higher temperatures. Therefore, both the DC and IDC plants have significant potential for increased efficiency. The DC plant has the highest efficiency potential because it uses the direct Brayton cycle.

In summary, each of the designs has the potential for expansion to increased power output within the baseline 450 MWt reactor vessel envelope. The DC and steam cycle could probably be expanded to 600 MWt. The IDC is limited by maximum helium circulator power to about 550 MWt. The DC has the best potential for further increases in thermodynamic efficiency.

12.2 PROSPECTS FOR OTHER MHR APPLICATIONS

12.2.1 Background

Over the last three decades, many studies have been performed in the U.S., Japan, and Germany on non-electric applications for the HTGR. These can essentially be categorized by the processes that are compatible with a given reactor outlet temperature, namely, (1) at around 700°C, electricity plus high quality steam can be produced; (2) at 850°C, process heat becomes a reality; and (3) at 950°C, coal gasification and hydrogen production by thermochemical water splitting can be realized. The multi-purpose capability of the MHR is simply portrayed on Figure 12-3.

Plant Efficiency

Figure 12-2 THE DIRECT CYCLE HAS THE HIGHEST EFFICIENCY GROWTH POTENTIAL

Figure 12-3 MULTIPURPOSE CAPABILITY OF ADVANCED MHTGR

12.2.2 Potential Applications

The range of applications for the three candidate plants is summarized in Table 12-2. There has been no attempt to rank them, but rather to identify their capability.

In the steam cycle plant, very high quality superheated steam can be extracted before it enters the turbine. In the case of the two gas turbine concepts, the major heat exchangers in the cycle are located after the turbine expansion step and, hence, they only have the capability to provide low quality steam. However, this is adequate for many process energy users (e.g., paper production).

The capability to meet future high temperature process heat users' needs requires an intermediate heat exchanger. The indirect cycle is identified as having this capability, but it is noted that the nuclear heat source, reactor outlet area, hot duct and IHX would need to be uprated to operate at 950°C. These capabilities have been previously demonstrated in the AVR reactor in Germany and are currently being embodied in the Japanese High Temperature Test Reactor (HTTR) design.

12.2.3 21st Century Marketplace

Advanced combustion gas turbines are likely to dominate the utility market in the next two decades, based on low capital cost (e.g., \$600/kWe for combined cycles), an efficiency of 60% for combined cycles by the year 2000 burning natural gas, and an efficiency of over 50% on coal-derived gas. The next generation of nuclear plants for electricity production must be competitive, in order to penetrate the 21st century marketplace.

An emerging trend is high electricity demand growth in third world countries. In this context, the inherent and passive safety characteristics, modest size and simplicity of operation of the MHR concepts may provide a particularly important opportunity.

Nonelectric applications, such as cogeneration are not well defined, although the need for process steam and desalination are recognized. The ultimate "green market" involving the electricity/ hydrogen economy is also elusive.

High temperature process heat, encompassing coal gasification and liquefaction, is several decades away and generally recognized to be competitive when fossil fuel prices reach much higher levels (i.e., oil at \$50/barrel and natural gas at \$8/million cubic feet). The process heat plant may well be a second generation MHR (circa 2030).

Table 12-2
MHR ADAPTABILITY TO ALTERNATIVE MISSIONS

Application	Steam Cycle	Direct Cycle GT	Indirect Cycle GT
Utility Electricity Generation	X	X	X
Plutonium Burning with Power Generation	X	X	X
Cogeneration			
● Elec. + high quality steam	X		
● Elec. + low quality heat	X	X	X
● Elec. + desalination	X	X	X
High Temperature Process Heat*			
● Coal gasification			X
● Hydrogen			X
● Methanol			X

* Assumes upgrading of nuclear heat source to 950°C.

Section 13

COMPARATIVE ECONOMIC ASSESSMENTS

Initial economic projections were developed for the 450 MWt direct cycle (DC) and indirect cycle (IDC) GT-MHR designs and comparisons were made with the reference 450 MWt steam cycle (SC-MHR), as well as competing fossil and nuclear alternatives. An economic projection was also developed for a 600 MWt DC GT-MHR, as an indication of the growth potential of that concept. As previously noted in Section 12, the DC GT-MHR is not constrained by the circulator or heat exchanger limitations of the SC-MHR. A power level trade study has been initiated to confirm the feasibility of the higher power level option.

13.1 COST ESTIMATE GROUNDRULES

The GT-MHR cost estimates were developed, in terms of January 1992 dollars, in accordance with the Advanced Reactor Cost Estimate Guidelines (Ref. 13-1) developed by Oak Ridge National Laboratory for the U.S. Department of Energy (DOE). These guidelines have been applied to the gas-cooled reactor and liquid metal reactor programs and may be applied to other DOE advanced nuclear programs in the future. These advanced reactor guidelines were incorporated into more detailed cost estimate procedures and groundrules developed for the MHR by GCRA (Ref. 13-2). The purpose of establishing cost estimate groundrules is simple: to provide the basis for consistent and comparable advanced reactor cost estimates.

Major changes to the groundrules have taken place since the prior GT-MHR cost estimate (Ref. 13-3) including selection of the EPRI reference nuclear site, near Kenosha, WI. With the change in site, there were changes in labor costs and productivities, material costs, site conditions and ambient temperatures. The GT-MHR design was optimized on the basis of a siting envelope, rather than the Kenosha site, consistent with the standardized design and deployment philosophy for the GT-MHR. Table 13-1 lists a summary of the key groundrules adopted for the MHR cost estimates presented here. Default contingencies of 15% for conventional construction and 25% for nuclear safety-grade construction or innovative technologies were specified for use. Indirect costs for the nuclear island were based on an algorithm recommended by Bechtel. A Target Plant or Nth-of-a-kind (NOAK) plant was defined by the groundrules to be the plant that exceeds 4500 MWe installed electric generation capacity. For the GT-MHR, the Target Plant was the sixth plant or modules 21 to 24. The SC-MHR Target Plant was defined to be the seventh plant, modules 25 to 28.

The MHR cost estimates are expected to be more comparable with the recent USCEA report (Ref. 13-4) capital cost estimates, which are also based on the EPRI reference site. However, inconsistencies related to the application of contingency, indirect costs and site labor rates, productivities and site material costs may remain. These inconsistencies, where known, are expected to yield more conservative GT-MHR cost estimates relative to the evaluated USCEA alternatives.

TABLE 13-1

SUMMARY OF KEY COST ESTIMATE GROUND RULES

FINANCIAL PARAMETERS			SITE PARAMETERS	
REFERENCE COST DATE	JANUARY 1992		REFERENCE SITE	KENOSHA, WI
LWR FUEL PRICE	\$0.77/ MMBTU		WET BULB TEMPERATURES	
COAL PRICE	\$1.45/ MMBTU		MAXIMUM	80°F
NATURAL GAS PRICE	\$2.33/ MMBTU		BASE FOR ECONOMIC	52°F
UTILITY CAPITALIZATION	(%)	RETURN (%/YR)	SITE LABOR RATES	\$/HR
DEBT	50	9.7	BOILERMAKER	\$24.55
PREFERRED EQUITY	10	9.0	CARPENTER	\$24.01
COMMON EQUITY	40	14.0	ELECTRICIAN	\$27.09
ECONOMIC PARAMETERS	NOMINAL	REAL	IRON WORKER	\$27.48
INFLATION RATE	5.0	0.0	LABORER	\$20.94
CAPITAL/LABOR ESCALATION	5.0	0.0	MILLWRIGHT	\$22.47
NUCLEAR FUEL ESCALATION	5.0	0.0	OPERATING ENGINEER	\$25.49
COAL ESCALATION	6.0	1.0	PIPEFITTER	\$25.48
NATURAL GAS ESCALATION	7.3	2.2	TEAMSTER	\$16.15
FEDERAL INCOME TAX RATE	34.0	N/A	OTHERS	\$22.81
STATE INCOME TAX RATE	4.0	N/A	SITE MATERIAL COSTS	
PROPERTY TAX RATE	2.0	N/A	FORMWORK - NUCLEAR	\$2.00/SF
INTERIM REPLACEMENT RATE	0.5	N/A	FORMWORK - STANDARD	\$1.85/SF
COST OF MONEY	NOMINAL	REAL	DECKING - NUCLEAR	\$5.00/SF
BEFORE TAX	11.35	6.05	DECKING - STANDARD	\$3.00/SF
AFTER TAX	9.57	4.36	REINFORCING STEEL - NUC.	\$700/TN
TAX ASSUMPTIONS	NUCLEAR	FOSSIL	REINFORCING STEEL - STD.	\$450/TN
DEPRECIATION PERIOD (YRS)	15	20	EMBEDDED STEEL - NUC.	\$2.50/LB
BOOK LIFE	30	30	EMBEDDED STEEL - STD.	\$1.50/LB
DEPRECIATION METHOD	150% DECLINING BAL		CONCRETE - NUCLEAR	\$90/CY
ACCOUNTING METHOD	NORMALIZED		CONCRETE - STANDARD	\$60/CY
PRODUCTION ASSUMPTIONS			STRUCTURAL STEEL - NUC.	\$3100/TN
TARGET PLANT DEFINITION	PLANT EXCEEDS 4500 MWe		STRUCTURAL STEEL - STD.	\$1400/TN
LEARNING ASSUMPTIONS	94% FACTORY LEARNING		MISC. STEEL - NUCLEAR	\$6000/TN
			MISC. STEEL - STANDARD	\$3000/TN

13.2 CAPITAL COSTS

Table 13-2 provides a breakdown of MHR capital costs for the two GT-MHR concepts evaluated and the reference SC-MHR plant. Each plant consists of four 450 MWt reactor modules with common facilities. The DC GT-MHR offers the lowest direct plant cost (\$825 million) and the highest net electric power output (869 MWe). For all three MHR concepts, structural costs dominate field labor and construction costs. The MHR has less piping and electrical construction relative to other nuclear concepts due to the elimination of active safety systems and reliance on the inherent characteristics of the MHR. The DC GT-MHR, through elimination of the steam and feedwater systems and their auxiliaries, has significantly less piping and electrical system requirements as reflected in the Table 13-2 cost estimates. Total base construction costs range from \$1,219 million for the 693 MWe SC-MHR to \$1,455 million for the 806 MWe IDC GT-MHR and to \$1,192 million for the 869 MWe DC GT-MHR.

The updated MHR cost estimates presented herein have increased relative to prior published estimates, due to a combination of design changes and cost estimate guideline changes. The design related changes were dominated by higher structural costs primarily caused by stiffer and stronger reactor buildings, the addition of a vented low pressure containment, higher reactor plant equipment costs, and higher indirect costs. The updated costs are expected to be more conservative than prior estimates and incorporate more design definition.

Indirect costs are derived from field labor costs, using an algorithm provided by Bechtel. Owner's costs were estimated by GCRA for each MHR concept and are reported in Reference 13-5. The DC GT-MHR plant owner's cost includes \$31 million for a spare turbomachine deemed necessary to reduce planned outages and maintain high module availability. At this time, a spare turbomachine has not been included in the IDC costs.

Contingency is applied to each line item in the cost estimates and total contingency ranges from 19% for the SC-MHR to 24% for the DC GT-MHR. AFUDC adds approximately 12% to the total capital cost reported on Table 13-2. The IDC GT-MHR was evaluated to have the highest capital cost (\$1,981 million) and unit capital cost (\$2,457/kWe) resulting primarily from the inclusion of the intermediate heat exchanger and secondary helium loop. The SC-MHR capital cost was \$350 million lower than the IDC GT-MHR but the lower plant electrical output pushed unit capital costs to \$2,349/kWe. The DC GT-MHR had slightly higher capital costs at \$1,659 million than the SC-MHR but the higher electrical output dramatically reduced unit capital cost to \$1,910/kWe. As shown on Table 13-3, the evaluated capital contributions to busbar costs are 30.2 mills/kW-hr for the SC-MHR, 31.6 mills/kW-hr for the IDC GT-MHR, and 24.6 mills/kW-hr for the DC GT-MHR.

13.3 OPERATING AND MAINTENANCE COSTS

MHR operating and maintenance (O&M) costs have been estimated on a bottom up basis and the details are documented in Reference 13-6. The non-fuel O&M costs are estimated for the 30 year economic life of the plant. The estimated O&M costs vary among the SC-MHR, IDC GT-MHR, and the DC GT-MHR due to the fundamental differences in plant design and the population of equipment. The elimination of the steam turbine and associated equipment is expected to significantly reduce maintenance requirements for both GT-MHR concepts relative

TABLE 13-2
MHR CAPITAL COSTS ('92\$)
TARGET PLANTS – 2016 STARTUP

	STEAM CYCLE	INDIRECT CYCLE	DIRECT CYCLE
REACTOR THERMAL POWER (MWt)	4x450	4x450	4x450
NET EFFICIENCY (%)	38.5%	44.8%	48.3%
NET ELECTRIC RATING (MWe)	693	806	869
CAPACITY FACTOR	84%	84%	84%
DIRECT COST ACCOUNTS: (M\$)			
LAND & LAND RIGHTS	2	2	2
STRUCTURES & IMPROVEMENTS	150	160	129
REACTOR (BOILER) PLANT EQUIPMENT	423	523	460
TURBINE PLANT EQUIPMENT	156	263	123
ELECTRIC PLANT EQUIPMENT	52	53	53
MISCELLANEOUS PLANT EQUIPMENT	40	40	31
MAIN CONDENSER HEAT REJECTION	30	21	27
TOTAL DIRECT COST	853	1,062	825
INDIRECT COST ACCOUNTS: (M\$)			
CONSTRUCTION SERVICES	117	123	98
HO ENGINEERING AND SERVICE	60	63	62
FO SUPERVISION & SERVICE	57	60	46
OWNER'S COST	132	147	161
TOTAL INDIRECT COST	366	393	367
BASE CONSTRUCTION COST (M\$)	1,219	1,455	1,192
CONTINGENCY (M\$)	232	310	285
TOTAL OVERNIGHT COST (M\$)	1,451	1,765	1,477
AFUDC (M\$)	176	216	182
TOTAL CAPITAL COST (M\$)	1,627	1,981	1,659
UNIT CAPITAL COST (\$/kWe)	2,349	2,457	1,910

TABLE 13-3
MHR BUSBAR GENERATING COSTS ('92\$)
TARGET PLANTS – 2016 STARTUP

	STEAM CYCLE	INDIRECT CYCLE	DIRECT CYCLE
REACTOR THERMAL POWER (MWt)	4x450	4x450	4x450
NET EFFICIENCY (%)	38.5%	44.8%	48.3%
NET ELECTRIC RATING (MWe)	693	806	869
CAPACITY FACTOR	84%	84%	84%
TOTAL CAPITAL COST (M\$)	1,627	1,981	1,659
UNIT CAPITAL COST (\$/kWe)	2,349	2,457	1,910
FIXED CHARGE RATE	9.47%	9.47	9.47%
LEVELIZED CAPITAL COST (M\$/YR)	154	188	157
FIXED O&M COST (M\$/YR)	34.6	31.1	27.6
VARIABLE O&M COST (mills/kWh)	0.2	0.2	0.2
CONTROL ROD & REFLECTOR REPLACE (M\$/YR)	4.8	4.8	4.8
ANNUAL O&M COST (M\$/YR)	40.6	37.0	33.5
FUEL COST (\$/MBTU)	1.26	1.27	1.28
LEVEL FUEL CYCLE COST (M\$/YR)	56.7	57.6	58.0
DECOMMISSIONING COST (M\$)	194	199	199
LEVEL DECOMMISSIONING (M\$/YR)	5.2	5.4	5.4
REVENUE REQUIREMENT (M\$/YR)	257	288	254
BUSBAR COST (mills/kWh)			
CAPITAL	30.2	31.6	24.6
O & M	8.0	6.2	5.2
FUEL	11.1	9.7	9.1
DECOMM	1.0	0.9	0.8
TOTAL	50.3	48.4	39.7
BUSBAR COST RELATIVE TO TARGET SC-MHR	1.00	0.96	0.79

to the SC-MHR concept. The DC GT-MHR offers a more simplified, compact plant design, with the least number of systems and components to operate and maintain. While the IDC design simplifies some of the aspects of the power conversion function, the degree of simplification is not as great as in the DC, since the main circulator is retained and the IHX replaces the steam generator of the SC-MHR. An offsetting factor is improved access for power conversion system maintenance in the IDC.

Certain O&M costs, such as those for materials and supplies, are dependent on the amount of energy generated by the plant. These variable O&M costs, which include the periodic replacement of control rods and reflector blocks, are added to the fixed costs to arrive at total O&M costs. Fixed costs include on-site and off-site staff costs, pensions, benefits, insurance and administrative and general costs. The plant operating staff salaries used to determine the on-site staff costs were defined in the Reference 13-1 guidelines together with a 10% premium to cover payroll taxes and unemployment insurance and 25% premium to account for pension and benefits. An additional 1% was included in the MHR pension and benefits entry to account for premium time paid to shift employees for holiday work, night shift, and certain weekend shifts. Personnel costs dominate the estimated MHR O&M cost estimates and correspond to 46% of the DC GT-MHR Target Plant.

The on-site staff size projected for the MHR is significantly lower than current U.S. power plants varying from 305 personnel for the SC-MHR to 223 personnel for the DC GT-MHR. An additional 20 full time equivalent off-site personnel are assumed to be assigned full time to the plant and their costs are included in the O&M cost estimate. These include provisions for support from a Central Operational Support Organization that is described in Reference 13-6. In the U.S., efforts are underway to reduce O&M costs and some of the unnecessary burdens that have increased staff requirements. The MHR will benefit from these efforts but also benefits from standardization and certification of the MHR design, enhanced safety performance, and substantial reductions in equipment important to safety. Current U.S. experience deals with largely unique designs at each site, which continue to evolve to meet changing regulatory requirements and improve performance. The MHR's enhanced safety features will hopefully permit substantial reductions in technical specifications and the amount of equipment subject to regulatory purview and reporting requirements. Through standardization and the availability of the Central Operational Support Organization, interface with the NRC for any future modifications may be designed and amortized over an entire class of MHR plants rather than requiring on-site staff at each plant.

O&M costs for the three MHR concepts are presented on Table 13-3. The DC GT-MHR Target Plant O&M costs were estimated to be nearly \$33.5 million per year, including \$27.6 million fixed and \$5.9 million variable at 84% capacity factor. The IDC GT-MHR O&M costs were estimated to be \$37 million per year (\$31.1 million fixed and \$5.9 million variable) and the SC-MHR O&M costs were estimated to be \$40.6 million per year (\$34.6 million fixed and \$6 million variable).

Busbar O&M costs at the 84% capacity factor were 5.2 mills/kW-hr for the DC GT-MHR, 6.2 mills/kW-hr for the IDC GT-MHR, and 8.0 mills/kW-hr for the SC-MHR. The relative O&M busbar cost advantage of the DC GT-MHR over the SC-MHR may be broken down into two components, a 20% reduction due to increased power output and a 17% reduction in annual

O&M costs. The IDC GT-MHR maintains a 14% advantage in power output and a 9% reduction in annual O&M costs.

13.4 FUEL CYCLE COSTS

The fuel cycle costs presented are based on 30 year operation with 18 month operating periods between refuelings. Costs for periodic reflector block and control rod replacement costs are included in the operations and maintenance expenses reported in Section 13.3.

The reference MHR fuel cycle is a once-through, low enriched Uranium (LEU) cycle. Fissile particle enrichment is 19.9%. Average enrichment is 15.5% and burnup is designed to be 121,000 MWt-day/metric ton of heavy metal. Spent fuel waste disposal costs are assumed to be covered by the 1 mill/kW-hr charge, as provided for in the Nuclear Waste Policy Act. Spent fuel would be shipped in transport casks, without processing or reduction, to a government operated waste repository.

In accordance with the Reference 13-2 groundrules, the nuclear fuel cycle cost input assumptions used in the fuel cycle cost evaluation include uranium at \$25/lb U_3O_8 , conversion at \$10/kgU, and enrichment at \$125/kg-SWU under 10.5% and \$925/kg-SWU over 10.5%. The higher efficiency (lower heat rate) of the GT-MHR concepts increases the \$/MMBTU spent fuel disposal cost based on the 1 mill/kW-hr Waste Policy Act assessment.

Table 13-3 presents the MHR fuel costs which vary from \$1.26/MMBTU for the SC-MHR to \$1.28/MMBTU for the DC GT-MHR. Levelized busbar fuel costs were estimated to be 11.1, 9.7, and 9.1 mills/kW-hr for the SC-MHR, IDC GT-MHR, and DC GT-MHR Target Plants, respectively.

13.5 DECOMMISSIONING COSTS

A MHR specific decommissioning cost estimate was developed during 1993 by Bechtel (Ref. 13-7) for the SC-MHR using actual quantities and commodities from the detailed cost estimate for a decommissioning scenario defined to remove all radioactive waste from the site and all construction material to a level of 1 m (3 ft) below grade. Based on the quantity takeoffs, the cost of decommissioning was estimated to be \$194 million. Decommissioning costs for the GT-MHR designs were estimated to be \$199 million for the reference four module power plant. Two alternative decommissioning scenarios were evaluated, ranging in cost from \$120 million for entombment of low level waste in-situ to \$260 million for removal of all structures and materials and return to "green field" site conditions. Consistent with the groundrules, funds for decommissioning are collected over the 30 year economic life and invested in an external sinking fund of high grade tax free municipal funds yielding a nominal 7%/year or 2% above inflation. Levelized busbar decommissioning costs contributed 1 mill/kW-hr or less to total busbar costs and, thus, the decommissioning method will have minor impact on total busbar costs.

13.6 TOTAL GENERATION COSTS

Table 13-3 presents the 30 year levelized busbar generation costs for the three MHR concepts. The SC-MHR plant has the highest evaluated busbar costs at 50.3 mills/kW-hr. A larger O&M staff associated with the steam/feedwater and water treatment systems increased O&M costs relative to the GT-MHR concepts, and plant efficiency affects both the O&M and fuel cost components of busbar cost. The IDC GT-MHR has a higher capital cost component than the SC-MHR, but benefits from reduced staffing assumptions and higher net thermal efficiency reduce the busbar cost estimate to 48.4 mills/kW-hr, a 4% improvement over the steam cycle. The DC GT-MHR combines the lowest capital cost and O&M staff requirements with the highest plant electrical output leading to a busbar cost estimate of 39.7 mills/kW-hr. The DC GT-MHR is 21% lower than the SC-MHR and 18% lower than the IDC GT-MHR busbar cost estimates.

13.7 COMPARISON WITH ALTERNATIVES

Table 13-4 provides a comparison of the three MHR concepts with the pulverized coal (PC) and natural gas combined cycle combustion turbine (CCCT) options prescribed in the Reference 13-1 guidelines and described in the USCEA study (Ref. 13-4). The breakdown of the busbar costs is presented graphically in Figure 13-1. In addition, an Advanced Light Water Reactor (ALWR) and an integrated coal gasification combined cycle (IGCC) plant were added from the USCEA Study. All cases presented are for the guideline fuel cost scenarios as presented in Table 13-1. The ALWR and IGCC cases were also adjusted for AFUDC, to be consistent with the approach identified in Reference 13-1. For the 2016 startup, all options are in the competitive range, given the amount of uncertainty in estimating operating costs between the years 2016 and 2045.

The DC GT-MHR Target Plant provides an advantage of nearly 18% over the three fossil options presented. The IDC GT-MHR Target Plant is competitive with the fossil options, and the SC-MHR is marginally higher in evaluated busbar generation costs. Clearly the DC GT-MHR provides a significant economic incentive and offers the opportunity to pursue higher temperature applications of nuclear power. The ALWR plant cost estimate is presented as a point of discussion only, as updated estimates for the ALWR, consistent with the Reference 13-1 Advanced Reactor Cost Estimating Guidelines, were not available for a direct comparison. Based on GCRA's understanding of the labor rates, productivities, and plant contingencies applied to the original ALWR cost estimate, an updated capital cost estimate consistent with Reference 13-1 groundrules and assumptions would be higher than that reported here, unless performance improvements or other cost reductions have been implemented.

Also included in Table 13-4 is a range of environmental externality costs which have been evolving throughout the U.S. on a state-by-state basis. Numerous studies have been completed and the range of values included in Table 13-4 are indicative of the diversity of results. Consistent with the USCEA Study, a \$500/ton Sulphur tax has been included in the O&M costs. This adds 1 mill/kW-hr to the reported busbar costs for the PC plant and 0.2 mills/kW-hr to the IGCC plant. The intent is to capture the full environmental cost of the respective option, at least in the decision process for option selection.

TABLE 13-4
SUMMARY GENERATION COST COMPARISON ('92\$)
2016 STARTUP, PLANTS UNDER 900 MWe

COST COMPONENTS	MHR TARGET PLANTS			ALWR	COAL		GAS
	STEAM	GT	GT	USCEA	PC	IGCC	CCCT
	CYCLE	IDC	DC	BASED	REF	REF	REF
o THERMAL RATING (MWt)	4x450	4x450	4x450	1828	1705	2x655	2x550
o NET RATING (MWe)	693	806	869	600	600	500	500
o NET EFFICIENCY (%)	38.5	44.8	48.3	32.8	35.2	38.1	45.4
o NET HEAT RATE (BTU/kWh)	8,868	7,620	7,070	10,400	9,700	8,950	7,514
o CAPACITY FACTOR (%)	84	84	84	80	80	84	84
o # OF TURBINES	4	4	4	1	1	2	2
o SCHEDULE (OVERALL)	60	63	63	60	42	42	24
o SCHEDULE (CONSTRUCTION)	32	36	36	42	30	30	18
o TOTAL CAPITAL (M\$)	1,627	1,981	1,659	1,140	836	862	282
o UNIT CAPITAL (\$/kWe)	2,349	2,457	1,910	1,900	1,394	1,723	565
o ANNUAL O&M (\$/kWe)	58.6	45.9	38.5	64.2	63.2	55.8	12.9
o FUEL COST (\$/MMBTU)	1.26	1.27	1.28	0.77	1.45	1.45	2.33
o REAL ESCALATION (%/YR)	0.0	0.0	0.0	0.0	1.0	1.0	2.2
BUSBAR COST (mills/kWh)							
o CAPITAL	30.2	31.6	24.6	26.0	19.4	22.9	7.5
o O&M	8.0	6.2	5.2	9.2	9.0	7.6	1.8
o FUEL CYCLE	11.1	9.7	9.1	8.0	20.3	18.7	39.3
o DECOMMISSIONING	1.0	0.9	0.8	1.0	0.1	0.1	0.0
TOTAL	50.3	48.4	39.7	44.2	48.8	49.3	48.6
OTHER FACTORS (mills/kWh)							
o ENVIR. EXTER. RANGE	~0-1	~0-1	~0-1	~0-2	~2-40	~1-20	~1-8

FIGURE 13-1

**BUSBAR GENERATION COST BREAKDOWN ('92\$)
2016 STARTUP, TARGET PLANTS UNDER 900 MWe**

It is realized that the levelized generation cost comparison is but one indicator of merit. In today's "least-cost, integrated-resource-planning", environment, many other factors of business risk, system reliability and infrastructure capability come into play. However, the generation cost serves as a convenient indicator for judging the worthiness of development priority among competing alternatives.

13.8 MHR STRETCH POTENTIAL

As previously noted in Section 12, consideration of the GT-MHR has led to a reexamination of factors limiting reactor thermal output, as the prior SC-MHR design was limited primarily by steam generator and main helium circulator considerations. To explore this potential, an on-going study is evaluating the DC GT-MHR at reactor thermal powers up to 600 MWt through consideration of higher power density and/or increasing the active core volume. This power level trade study is examining higher thermal powers without increasing reactor vessel diameter or sacrificing any of the MHR safety characteristics.

The increased plant thermal and electrical output (up to 33%) is offset by relatively small increases in plant capital costs and fuel cycle costs leading to substantial improvements in projected busbar economics. Table 13-5 compares three 600 MWt reactor modules coupled with three turbomachines with the same electrical output as the reference four module DC GT-MHR plant design. Even though plant facilities and plant staff are shared less effectively, the 3x600 DC GT-MHR plant busbar cost is projected to be nearly 6.6% lower (37.1 versus 39.7 mills/kW-hr). Considering the larger plant electrical output, the busbar generation cost of a 4x600 DC GT-MHR plant is projected to be 34.8 mills/kW-hr or 12.3% lower than the reference 4x450 DC GT-MHR power plant, as station output increases from 869 MWe to 1,159 MWe. Although the selection of a reference core thermal output will not be made until the power level trade study is completed, a recommendation to increase the reference core thermal output for the DC GT-MHR plant design is expected.

Table 13-5 presents three DC GT-MHR plant configurations with two ALWR plant alternatives of 1200 MWe, and 1000 MWe IGCC and CCCT alternative power generation options. The 4x600 MWt GT-MHR busbar cost of 34.8 mills/kW-hr compares favorably with the 47.1 mills/kW-hr and 47.8 mills/kW-hr estimated for the 1000 MWe IGCC and CCCT, respectively. The GT-MHR is also projected to be competitive with 1200 MWe ALWR station alternatives.

Figure 13-2 provides a bar chart presentation of the large power plant busbar costs presented in Table 13-5, comparing the 3 large GT-MHR plants with 1000-1200 MWe nuclear and fossil alternatives. One interesting observation is that the IGCC capital cost component is comparable to the nuclear plants. As natural gas prices increase in the future, nuclear alternatives to IGCC may provide less expensive, more environmentally sensitive options at a comparable capital cost. This would be an appropriate economic goal for advanced reactors.

Figure 13-3 provides a graphic comparison of the four DC GT-MHR configurations with the mid-size and large power generation alternatives presented in Tables 13-4 and 13-5. Three different plant configurations for the 600 MWt are included, a two, three and four module plant design identified by the solid triangles on Figure 13-3. As shown, the GT-MHR is competitive

TABLE 13-5
SUMMARY GENERATION COST COMPARISON ('92\$)
2016 STARTUP, PLANTS OVER 800 MWe

COST COMPONENTS	DC GT-MHR TARGET PLANTS			ALWR		COAL	GAS
	4x450	3x600	4x600	2X600	1X1200	IGCC 4X250	CCCT 4X250
o THERMAL RATING (MWt)	4x450	3x600	4x600	3,657	3,586	4x655	4x550
o NET RATING (MWe)	869	869	1,159	1,200	1,200	1,000	999
o NET EFFICIENCY (%)	48.3	48.3	48.3	32.8	33.5	38.1	45.4
o NET HEAT RATE (BTU/kWh)	7,070	7,070	7,070	10,400	10,200	8,950	7,514
o CAPACITY FACTOR (%)	84	84	84	80	80	84	84
o # OF TURBINES	4	3	4	2	1	4	4
o SCHEDULE (OVERALL)	63	60	63	78	72	42	24
o SCHEDULE (CONSTRUCTION)	36	33	36	60	60	30	18
o TOTAL CAPITAL (M\$)	1,658	1,490	1,830	2,034	1,860	1,611	531
o UNIT CAPITAL (\$/kWe)	1,910	1,710	1,580	1,695	1,550	1,611	531
o ANNUAL O&M (\$/kWe)	38.5	36.9	32.8	54.5	45.5	50.9	11.2
o FUEL COST (\$/MMBTU)	1.28	1.31	1.31	0.77	0.77	1.45	2.33
o REAL ESCALATION (%/YR)	0.0	0.0	0.0	0.0	0.0	1.0	2.2
BUSBAR COST (mills/kWh)							
o CAPITAL	24.6	22.1	20.3	23.2	21.2	21.4	7.0
o O&M	5.2	5.0	4.5	7.8	6.5	6.9	1.5
o FUEL CYCLE	9.1	9.3	9.3	8.0	7.9	18.7	39.3
o DECOMMISSIONING	0.8	0.7	0.7	0.6	0.6	0.1	0.0
TOTAL	39.7	37.1	34.8	39.6	36.2	47.1	47.8
OTHER FACTORS (mills/kWh)							
o ENVIR. EXTER. RANGE	~0-1	~0-1	~0-1	~0-2	~0-2	~1-20	~1-8

FIGURE 13-2
BUSBAR GENERATION COST BREAKDOWN ('92\$)
2016 STARTUP, TARGET PLANTS OVER 800 MWe

FIGURE 13-3
DC GT-MHR VS. ALTERNATE BASELOAD OPTIONS
2016 STARTUP, REFERENCE GROUND RULES, 30 YR LEVEL

at all power levels from 600 to 1200 MWe. The reference 4x450 GT-MHR design is also competitive with the ALWR and maintains nearly a 20% advantage over the larger fossil alternatives presented.

REFERENCES

- 13-1 Cost Estimate Guidelines for Advanced Nuclear Power Technologies, ORNL/TM-10071/R3, Oak Ridge National Laboratory, Oak Ridge, TN, March 1993.
- 13-2 MHTGR Cost Estimating Groundrules & Procedures, HP 10701, Revision 4, Gas-Cooled Reactor Associates, San Diego, CA, May 1993.
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- 13-4 Advanced Design Nuclear Power Plants: Competitive, Economical, Electricity, U.S. Council for Energy Awareness, Washington, DC, June 1992.
- 13-5 Estimate of Owner's Cost for Modular HTGR Plants, GCRA 90-003, Revision 1, Gas-Cooled Reactor Associates, San Diego, CA, August 1993.
- 13-6 Utility Projections of Operation and Maintenance Costs for Modular HTGR Plants, GCRA 90-005, Revision 1, Gas-Cooled Reactor Associates, San Diego, CA, July 1993.
- 13-7 MHTGR Decontamination and Decommissioning Cost Estimate, Attachment to BNI/GCRA:93-746, Bechtel National, Inc., San Francisco, CA, August 31, 1993.

SECTION 14

DEVELOPMENT REQUIREMENTS

Development requirements supporting the direct cycle (DC) and indirect cycle (IDC) gas turbine (GT-MHR) designs were evaluated and compared with the reference steam cycle (SC-MHR) development requirements. It was judged that the development requirements for the enhanced (475 MWt) SC-MHR design would not appreciably vary from the 450 MWt reference SC-MHR. Therefore, the enhanced SC-MHR was not specifically addressed. In the following subsections, development efforts supporting the DC and IDC are discussed in further detail. Technology development cost projections are then presented for all three concepts, with the SC-MHR serving as a baseline.

14.1 DIRECT CYCLE DEVELOPMENT

14.1.1 Overview

Of the two gas-turbine concepts, the DC poses the more difficult development challenge. However, the more significant issues relate to systems integration, rather than the feasibility or performance of individual components. DC components are, for the most part, enveloped by the existing state of the art. There are notable exceptions, however, that are identified in Section 14.1.2. The particular challenge of the DC is the integration of these individual components into a compact, highly reliable, operable and maintainable system that consistently attains the high performance that has been projected for it. Some of the issues and tradeoffs that must be made, and ultimately confirmed through the development program are:

- The need for a compact, highly integrated system, that must also:
 - Provide access to individual components for inspection and maintenance
 - Facilitate remote maintenance operations to reduce personnel exposure
 - Minimize pressure losses and helium bypass throughout the system
- Demonstration of adequate safety margins for events related to turbomachinery failures
- Verification of the integrated performance and operability of the power conversion system

In defining a development strategy for the DC, the use of existing technology bases and the potential for international participation will be maximized in the following areas: (1) experience with helium systems (e.g., FSV, Oberhausen, HHV, HTTR, etc.), (2) turbomachinery technology from the gas turbine industry (aircraft and industrial engines), (3) heat exchanger technology (with emphasis on compact units), and (4) state-of-the-art control, instrumentation, and simulations. Given this input, the development strategy involves both component tests and an integrated power conversion system test.

14.1.2 Component Development

The following DC components are evaluated to have significant development needs.

Turbomachine

There are gas turbines, currently operating in utility systems, that are comparable in physical size or larger than the turbomachine for the DC GT-MHR. In some respects, these turbomachines operate under more demanding conditions than those proposed for the DC. To illustrate this point, Table 14-1 compares the DC turbomachine with a large combustion open cycle machine manufactured by ABB Power Systems. The gross power outputs of the two machines are 222 MWe and 110 MWe, respectively. Note that the turbine inlet temperature of the open cycle machine is some 190°C (340°F) above that proposed for the DC. Reflecting the differences in the properties of helium versus air, the pressure ratio of the DC turbomachine is 2.8, compared to 13.8 for the open cycle GT. The number of stages is greater for the DC and the rotational speed higher, resulting in a longer, but smaller diameter rotor. The weight of the rotor is estimated to be about 30% less for the DC. It is also worth noting, that the inert helium working fluid avoids the Mach number limitations of air systems, and is considered less difficult from a chemistry viewpoint than the corrosive combustion gases of conventional gas turbines.

The open cycle machine employs conventional oil bearings, while magnetic bearings have been selected for the DC. Magnetic bearings are an emerging commercial technology, with over a million hours of accumulated operating experience in turbomachinery. The majority of this experience is centered in gas pipeline compressors. Magnetic bearings are also being developed for marine and aircraft gas turbine applications by the Department of Defense. While the advantages of magnetic bearings are particularly relevant to the DC design (See Section 5.8.2), the size and vertical orientation of the DC turbomachinery require a substantial extension of the current state of the art. This must be achieved through appropriate development.

In the context of this current experience, a number of issues remain to be addressed within the DC development effort. These include the following:

1. Magnetic Bearings and Associated Catcher Bearings

While there is extensive experience with active magnetic bearings, there are no vertically oriented machines with magnetic bearings operating in this size range. Input received from a leading manufacturer of magnetic bearings, however, indicates that the magnetic bearings themselves should not be a major development issue. Relative to horizontal applications, the steady state loads on the journal bearings will be reduced by the elimination of asymmetric gravitational forces in the vertically oriented machine. Gravitational loads on the thrust bearing will be offset by differential thrust loads during operation, although the full weight of the machine must be supported during startup. Dynamic loads, given the higher specific power of the DC turbomachine, represent an uncertainty that must be resolved during design and development so that the bearings can be appropriately specified. Once properly understood, such loads have not been found limiting in other applications.

Table 14-1 GAS TURBINE COMPARISON

TURBINE	ABB GT11N2	DC GT-MHR
TYPE	COMBUSTION OPEN CYCLE	HELIUM CLOSED CYCLE
MWe	110	222
Thermal Efficiency, %	34.1	50.3
Single Rotor	Yes, Welded	Yes
Blade Materials	High Chromium, Cooled	Nickel-Based Alloy, Uncooled
Generator	Air Cooled	He Submerged
Shaft Speed, rpm	3600	5165
Starting Device	Motor by Generator	Motor by Generator
Frequency Converter	Yes, to feed Generator with variable frequency and constant current	Yes, for Conversion to 60 Hz
Pressure Ratio	13.8	2.8
Turbine Inlet Temperature °C (°F)	1038 (1900) (calc)	850 (1562)
Exhaust Gas Temperature °C (°F)	528 (982)	510 (950)
Mass Flow, kg/sec (lbs/sec)	375 (827)	240 (529)
No. of Stages, Turbine	4	9
No. of Stages, Compressor	16	12 (LP), 16 (HP)
Rotor Largest diameter, mm (in)	1445 (56.9)	965 (38.0)
Length, m (ft)	8.14 (26.7) (T/C)	9.63 (31.6) (T/C)
Mass, kg (lbs)	52,520 (115,786) (T/C)	~ 36,300 (~ 80,000) (T/C)

Table 14-1 GAS TURBINE COMPARISON (CONTINUED)

TURBINE	ABB GT11N2		DC GT-MHR	
TYPE	COMBUSTION OPEN CYCLE		HELIUM CLOSED CYCLE	
Turbine				
Blade Height, mm (in)	Row 1	173 (6.8)	Row 1	119 (4.7)
	Row 4	559 (22.0)	Row 4	193 (7.6)
Tip-to-Tip Diameter, mm (in)	Row 1	1791 (70.5)	Row 1	1214 (47.8)
	Row 4	2563 (100.9)	Row 4	1346 (53.0)
Compressor				
Blade Height, mm (in)	Row 1	386 (15.2)	LP (Row 1)	132 (5.2)
	Row 16	76 (3.0)	HP	99 (3.9)
Tip-to-Tip Diameter, mm (in)	Row 1	2009 (79.1)	LP	1115 (43.9)
	Row 16	1504 (59.2)	HP	996 (39.2)
No. of Bearings				
Journal Bearings		2 Oil		3 Magnetic
Thrust Bearings		1 Oil		1 Magnetic
Catcher Bearings		-----		Yes
Weights, kg (lbs)				
Gas Turbine Module		> 110,019 (> 242,550)		TBD
Generator-Exciter Module		> 126,022 (> 277,830) (w/ housings)		TBD
Overall Length, m (ft)				
Gas Turbine/Compressor/Generator		~ 27.4 (~ 90)		17.2 (56.4)
Overspeed Protection				
		Electronic Sensors 112%		Bypass Sensors 120%
Missile Retention				
		Nodular Cast Iron Housing		Laminated Inc 625 Shield

By contrast, the catcher bearings for the DC turbomachine will require a substantial development effort. In the case of the thrust catcher bearing, some previous development testing has been accomplished in support of the SC circulator. While the DC turbomachine is significantly larger than the SC circulator, offsetting thrust effects will tend to reduce the loads on the catcher bearing. The larger concern is with the journal catcher bearing which has yet to be tested for large applications in a vertical orientation.

2. Interface Supports and Seals

Structural supports and associated seals must be developed for the interfaces between the turbomachine and the power conversion vessel internals. The resulting designs must provide adequate vertical and lateral support for the turbomachinery under the full range of static and dynamic loads, must provide positive sealing during plant operation and must allow the turbomachine to be readily removed and replaced, using remote handling techniques. The potential impacts on plant efficiency (seals) and plant maintenance (ease of removal and replacement) imply the need for a thorough qualification program.

3. Flow Distribution

Tests will be required to verify the flow distribution and pressure losses in the hot gas duct, turbine inlet scroll, and all other power conversion system inlet/outlet transitions

4. Overspeed Response

Tests will be required to verify turbine integrity under overspeed conditions and also to verify burst shield integrity as a backup in case of failure.

5. Control

Testing will be required to verify control valve performance. A simulation of control system performance will also be required.

Generator

As with the turbomachine, operation of a large, high speed generator in a vertical orientation has not yet been demonstrated, but is evaluated to be feasible. The bearing and overspeed issues identified above for the turbomachine are also applicable to the generator. In addition, the following issues must be addressed.

1. Generator Cooling

Cooling requirements for the generator are increased at higher rotational speeds. The proposed use of helium to cool the generator must also be verified through development.

2. Generator Power Leads

The DC design avoids large rotating seals by submerging the generator in helium. This requires the development of large electrical penetrations through the power conversion vessel. Indications from a leading manufacturer indicate that such penetrations are feasible; however, this must be verified through testing.

3. Generator Operation in High Pressure Helium

Internal generator components and materials, such as diodes and insulation must be evaluated for use in a high pressure helium environment. This includes assurance that generator deterioration will not occur under accident conditions, such as rapid helium depressurization.

Recuperator

Plate-fin recuperator modules, similar in size to those contemplated for the DC, have been successfully employed in conventional gas turbine cycles. In such applications, the requirements related to temperature, thermal cycling and the chemistry of the working fluid (exhaust gases) are comparatively more challenging. One difference, which must be addressed, is the higher pressure differential across the DC recuperator.

The interfaces (supports and seals) between the recuperator and the remainder of the power conversion loop will require particular attention within the development program. The design of these interfaces involves the resolution of potentially conflicting requirements related to structural support, low leakage, flexibility for thermal expansion and ease of removal and replacement. Other development issues include materials compatibility with the helium coolant, flow distribution, particularly in the inlet/outlet areas and the ability to detect and locate leaks impacting efficiency.

Precooler and Intercooler

The precooler and intercooler operate at relatively low temperatures and employ a single (liquid) phase on the cooling water side. Thus, these heat exchangers are not expected to pose unusual difficulties. Tests would be performed on individual modules to verify flow distribution, performance and integrity.

Additional discussion regarding the development needs of individual DC components and systems may be found in Section 5.

14.1.3 Power Conversion System Test

A major requirement of the DC development program is the integrated testing of a prototypical power conversion system before operation in a nuclear environment. A non-nuclear heat source will be required, and the major goal will be to verify the performance and integrity of the overall system within the power conversion vessel. Recognizing that the helium turbo-machine is a key element of the test, the GT-MHR Evaluation Team recommended that the

integrated test facility have the following characteristics: (1) full speed operation, (2) design value of turbine inlet temperature [i.e., 850°C (1562°F)], and (3) system design for reduced helium pressure (~20% was suggested) for cost effectiveness. It is recognized that these proposed requirements for the integrated test facility must be verified by the turbomachine manufacturer in the course of future work and updated, as appropriate.

It is anticipated that a full spectrum of plant steady state, transient and upset conditions will be simulated in the integrated test; however, overspeed to failure will not be included, since data in this area will have already been generated from the component testing phase.

The integrated test facility would also be used to demonstrate all of the necessary handling equipment to facilitate component installation and removal. If desired, the test facility could subsequently be used to proof test production and refurbished turbomachines before installing them in nuclear facilities.

14.1.4 Integrated Test Facility Options

Two approaches to the design of the proposed integrated test facility were explored, an option employing a gas fired heater and an option employing a modified open cycle gas-turbine. The former was used as a basis for the cost estimates presented in Section 14.3, since estimates for the latter were not yet available. This is expected to provide a conservative basis for those estimates.

Gas-Fired Heater Option

The test facility, which is conceptually depicted in Figure 14-1, includes a gas fired helium heater, a test cell containing the DC power conversion loop, auxiliary systems providing helium services, cooling, etc., data collection and analysis systems and administrative facilities. A process diagram for the test facility is provided as Figure 14-2.

The gas heater, which is the key component of the test facility, has been sized to result in a heater bundle pressure loss which is less than 14 kPa (2 psi) (so that the $\Delta P/P$ will be similar to that of the DC cycle design). Tentatively, the heater is a refractory lined steel structure, about 9 m (30 ft) high (excluding stacks), with a heater surface area of about 27,900 m² (300,000 ft²), and a heater bundle frontal area of 1440 m² (15,500 ft²). Tentatively, finned 76 mm (3 in) tubes of 800H are used. Eight 30.5 m (100 ft) long tubes are connected in a serpentine array to make one heater pass. About 460 parallel passes are required to obtain the requisite 100 MWt capacity. Assuming that the tube spacing can be 3 passes per foot, the footprint for the heater will be about 30.5 m (100 ft) by 47 m (155 ft). Multiple burners will be necessary to provide adequate heat distribution. To avoid the need for a tall stack and to ensure turbulent flow over the heater bundle, the need for a forced draft air supply is anticipated. Assuming that the overall capacity factor for the heater is below 50%, an air preheater has not been included. The heater efficiency is estimated to be about 40% (energy transferred to helium/energy content of gas burned). It is possible that a more economical design could be developed, if desirable.

Figure 14-2 INTEGRATED TEST FACILITY PROCESS DIAGRAM

Gas Turbine Option

Subsequent to the development of the cost estimates provided in Section 14.3, an alternative approach to the design of the integrated test facility was proposed that appears to have both technical and cost advantages. As shown in Figure 14-3, this arrangement employs a commercially available natural gas fired turbine-generator from which compressed air is extracted and heated in an external combustor to the desired temperature. The heated air is ducted to a compact "zero ΔP " helium heater made of existing, qualified materials. After the air passes through this heat exchanger, transferring heat to the helium, it is reheated in a second combustor to the desired gas turbine inlet temperature and returned to the turbine for expansion and power delivery. This arrangement is capable of providing the full 1562°F helium design temperature at part pressure [1.48 MPa (215 psia)] and part load (about 35% load for the conditions of Figure 14-3) for performing full scale integrated testing.

This integrated testing scheme appears to have several advantages. It uses commercially available components or components using proven materials which do not require technology development. Most importantly it eliminates the need for a very high temperature, high pressure differential heat exchanger (the helium heater) which itself is potentially a major developmental item. The concept is also more efficient.

One issue of importance is the availability of a combustion turbine which could be adapted to the process in question. A survey of available large simple cycle gas turbines was performed to determine the capabilities of units that either have or could be modified to utilize external combustors. Two such units are the ABB model GT 11N (83.8 MWe) and the ABB model GT 11N2 (109.2 MWe). Two other possibilities are the Siemens/KWU models V84.2 (103.2 MWe) and V84.3 (152 MWe). The parameters of Figure 14-3 relate to the ABB GT 11N2 unit, which was selected for a first look.

14.2 INDIRECT CYCLE DEVELOPMENT

The IDC concept incorporates systems and components from the reactor to the electric generator in an arrangement similar to that of the SC-MHR. The primary coolant system relies on the Fort Saint Vrain (FSV) operational experience and conceptual and preliminary design developed after FSV through the current SC-MHR concept. Specific technical issues and development needs for the main circulator are addressed in Section 5.2. The steam generator is replaced with a helium-to-helium intermediate heat exchanger (IHX). The IHX development issues are discussed in Section 5.3.

The IDC turbomachinery design incorporates a conventional, horizontal layout of the compressor, turbine, and generator. The selected design also employs a twin-shaft turbomachine concept, with the high-speed shaft turbine-compressors dynamically coupled to the synchronous shaft speed power turbine by the helium working fluid. Three compressor sections with intercooling are employed, along with a recuperator that preheats the compressor effluent before it enters the IHX with energy recovered from the turbine exhaust. The generator is a conventional, off-the-shelf unit. The turbomachinery relies on the extensive European operational experience with closed-cycle air turbines using fossil-fueled heaters (Table 14-2). In particular, the Oberhausen experience provides a base for turbomachinery using helium as the

Figure 14-3 GAS TURBINE INTEGRATED TEST FACILITY

TABLE 14-2
CLOSED CYCLE GAS TURBINE EXPERIENCE (END OF 1981)*

EXPERIMENTAL					INDUSTRIAL																			
					0.7 - 2.3 MW					6.0 - 6.6 MW					12-50 MW									
Plant	Yr	Fuel	MW	Mf	Plant	Yr	Fuel	MW	Mf	Plant	Yr	Fuel	MW	Mf	Plant	Yr	Fuel	MW	Mf					
AK-36	39	O	2.0	E	Coventry ⁵⁾	49	H	0.7	J	Coburg [*]	61	B	6.6	G	Paris ⁶⁾	52	O	12	E					
Clydebank	50	B	1.0	J	Ravensburg [*]	56	BO	2.3	EG	Haus Aden [*]	63	MB	6.4	G	Dundee ⁶⁾	54	O	12	J					
Clydebank	51	P	0.4	J	Toyotomi [*]	57	N	2.0	EF	Phoenix ³⁾	66	N	6.0	EL	Oberhausen I [*]	60	BC	14	EG					
Tuco-52	55	O	1.5	E	Altnabreak	59	P	2.2	J	4) Thermodynamic heated test bed (HHT-Project) 5) Heat utilization from retortes of a gas work 6) Reheated twin shaft 7) He-turbine plant					Kashira	61	L	12	EK					
IN-10 ¹⁾	61	O	7.0	M	Roths	60	S	2.0	J						Nippon Kokan	61	F	12	F					
ML-1 ²⁾	62	R	0.4		1) Ship propulsion 2) Mobile nuclear power plant 3) He-cycle for N ₂ production										Gelsenkirchen ^{**}	67	F	17	G					
La Fleur ³⁾	62	N	2.0	L											Vienna	72	Q	30	EW					
HHV ⁴⁾	81	-	-40	B											Oberhausen II ⁷⁾	74	C	50	GS					

*100,000 - 150,000 operating hours

**Nearly 100,000 operating hours

Fuels:

O Light Oil

C Coke Furnace gas

L Lignite

Q Heavy fuel oil

F Blast furnace gas

P Peat

N Natural gas

B Bituminous coal

R Nuclear

M Mine gas

S Coal slurry

H Waste heat

Manufacturers: B Brown Boveri
E Escher Wyss
F Fuji Electric
G Gute Hoffnungs Hütte
J John Brown
K EVT (KSG)
L La Fleur
M Mitsui
S Sulzer Brothers
W Waagner Biro

*Courtesy of H.U. Frutschi,
ABB Power Generation

working fluid. This is supplemented with the design experience gained in the HHT project. Power plants using twin-shaft turbomachinery are supported by operational data from aero-derivative fossil-fired plants currently in service. Specific technical issues and development needs are discussed in Section 5.9. European experience with closed cycle plants using air and helium as working fluids also provides a basis for the power conversion loop heat exchangers. Sections 5.5 and 5.7 present development needs for the recuperator and intercooler/precooler heat exchangers.

Although specific design feature technical support is described in the individual component sections, prototype testing may be required for the main circulator and will be required for the IHX. The main circulator proof testing could be performed in a vendor facility, which may need expansion to provide the appropriate capacity and power. The IHX would require a special test set-up. Depending on the power level deemed necessary for proof testing the IHX concept, a vendor facility, an expanded vendor facility, or special facility will be needed. The most likely approach would employ a natural-gas-fired helium heater and a primary and secondary helium circulator. These circulators would be within power and capacity ranges that are within current technology. Test objectives would be performance confirmation and validation of computational tools. Leak tightness between the primary and secondary paths would be confirmed. Training and experience on special maintenance and inspection features would be gained.

It is not anticipated that the power conversion loop, in general, and the turbomachinery, in particular, would require integrated proof testing, such as that proposed for the DC. The turbomachinery vendor would include any required confirmations within their normal scope.

The IDC itself could be viewed as a stepping stone to the DC. The operation of an IDC gas turbine power plant would provide an experience basis for the reactor system and power conversion loop. A next step could be the elimination of the IHX, yielding a DC with a conventional layout of the turbomachinery and heat exchangers. The final step to an integrated power conversion loop could be most easily accomplished with this experience base.

14.3 DEVELOPMENT COST PROJECTIONS

First-of-a-kind (FOAK) design and technology development costs were estimated for the DC and IDC designs, using the 450 MWt SC-MHR costs as a basis. The results are summarized in Table 14-3 and discussed below.

Technology development costs for the reference 450 MWt SC-MHR (SC) are summarized in the first two columns of Table 14-3. The \$298 million technology development cost is dominated by the fuel, fission product behavior and fuel process development activities which comprise \$158 million, or more than 53% of the total technology program. An additional \$48 million or 16% is associated with graphite and high temperature metals development. Design verification and support (DV&S) for the SC comprises \$71 million or 24% of the technology program costs. Program management, \$9 million, and miscellaneous categories add the remaining \$12 million in SC technology development costs. The weighted 90% confidence factor for the SC is 136%, leading to a SC 90% confidence technology cost estimate of \$405 million.

TABLE 14- 3
MHR DEVELOPMENT COST SUMMARY
(1992 M\$)

COST CATEGORIES	Steam Cycle		Gas Turbine - IDC		Gas Turbine - DC	
	50%/50%	90% Conf.	50%/50%	90% Conf.	50%/50%	90% Conf.
	Estimate	Factor	Estimate	Factor	Estimate	Factor
TECHNOLOGY DEVELOPMENT						
FUELS DEVELOPMENT; FP BEHAVIOR	100	145%	100	145%	100	145%
GRAPHITE DEVELOPMENT	22	109%	22	109%	22	109%
METALS DEVELOPMENT	26	110%	26	111%	31	110%
FUEL PROCESS DEVELOPMENT	58	150%	58	150%	58	150%
REACTOR SYSTEM DV&S	19	150%	19	150%	30	150%
CIRCULATOR DV&S	23	130%	34	130%	0	N/A
HEAT EXCHANGER DV&S	8	130%	6	150%	5	150%
TURBOMACHINERY DV&S	0	N/A	5	150%	125	170%
SHUTDOWN COOLING SYS. DV&S	7	130%	9	130%	9	130%
REACTOR CAVITY COOLING SYS. DV&S	5	141%	5	141%	5	141%
FUEL HANDLING & STORAGE SYS. DV&S	5	175%	5	175%	5	175%
PLANT CONTROL/PROTECTION SYS. DV&S	4	140%	4	140%	4	140%
PROGRAM MANAGEMENT	9	134%	10	121%	10	129%
MISC. (SAFETY, INT'L TECH TRANSFER)	12	161%	18	146%	18	146%
SUBTOTAL TECH. DEVELOPMENT	298	136%	321	139%	422	149%
DESIGN AND LICENSING						
PLANT LEVEL DESIGN & ANALYSIS	75	144%	84	147%	88	149%
REACTOR SYSTEM	80	145%	94	150%	94	150%
VESSEL SYSTEM	19	130%	19	150%	21	150%
CIRCULATOR	12	140%	18	140%	0	140%
HEAT EXCHANGERS	25	130%	25	150%	25	150%
TURBOMACHINERY	0	N/A	41	150%	90	150%
SHUTDOWN COOLING SYSTEM	15	137%	15	143%	15	143%
REACTOR CAVITY COOLING SYSTEM	3	140%	3	140%	3	140%
FUEL HANDLING & STORAGE SYSTEM	17	174%	17	174%	17	175%
OTHER NI SYSTEMS	15	127%	16	129%	14	134%
PLANT INSTRUM., CONTROL & PROTECT.	21	129%	18	133%	17	134%
TURBINE PLANT EQUIPMENT	7	106%	5	97%	1	60%
NI BUILDINGS & STRUCTURES	20	126%	20	129%	20	126%
ECA BUILDINGS & STRUCTURES	4	100%	3	121%	1	157%
PLANT AIR, WATER & SERVICE SYS.	11	109%	9	118%	8	122%
PLANT ELECTRICAL & SECURITY SYS.	11	121%	10	121%	9	124%
DESIGN MGT. & COST DEVELOPMENT	56	130%	66	138%	65	143%
REQ'TS, PLANNING & CONTROL, QA	25	125%	30	137%	28	147%
LICENSING MANAGEMENT	20	157%	21	157%	33	157%
SUBTOTAL DESIGN & LICENSING	436	137%	514	144%	549	147%
TOTAL DEVELOPMENT COST	734	136%	835	142%	971	148%
FACTOR RELATIVE TO SC	1.00	100%	1.14	104%	1.32	108%

The GT-MHR concept estimates were developed based on specific differences in technology requirements. However, most of the SC technology program as it relates to fuel, fission product behavior, fuel process, graphite, metals, reactor cavity cooling system, and fuel handling are the same for all MHR concepts. The IDC technology development cost estimate is nearly 8% higher than the SC at \$321 million. Table 14-3 identifies increases in the circulator (\$11 million), turbomachine (\$5 million), and international technology transfer (\$6 million) development costs to deploy the IDC concept. A much larger circulator, a high temperature helium to helium intermediate heat exchanger replacing the steam generator, and the closed cycle helium turbomachine are the reasons for the IDC technology program cost increases. The uncertainty in the IDC estimate increases slightly to 139%, leading to a IDC 90% confidence technology cost estimate of \$446 million.

The DC represents further advancement in technology from the SC. While the reactor system is basically unchanged, significant changes in the power conversion system are included in the DC. Elimination of the main circulator and steam generator and the introduction of an intercooled, recuperated closed cycle helium turbine in the primary loop result in additional increases in technology program costs. The recuperator, intercooler, and precooler arranged with a vertical turbomachine running on magnetic bearings with a helium cooled submerged generator represents a significant departure from current commercial technology. As a result, a full scale test of the power conversion system has been recommended and the costs are included in the turbomachine line item. For the purposes of the technology development cost estimate, it is assumed that the power conversion loop components would be tested in a fossil fired test facility and subsequently removed and deployed in the DC Lead Module. Thus, the capital cost of the power conversion equipment, including the turbomachine, heat exchangers and generator is not included in the development costs presented. The DC turbomachine test facility capital cost is estimated to be \$70 million. Startup and two years of operation for testing are estimated to cost \$35 million. An additional \$10 million was included for the power conversion test vessel. The \$91 million increase in technology program costs relative to the IDC is dominated by the \$115 million cost for the turbomachine test facility. This is offset by elimination of the circulator design verification and support (DV&S) (\$34 million). Additional costs are also estimated for metals technology, \$5 million, and reactor system DV&S, \$11 million, due to complexities related to placing the turbomachine in the primary system. The DC technology development cost is estimated to be \$422 million, which is 42% higher than the SC and 31% higher than the IDC. The uncertainty in the DC estimate increases to 149%, leading to a 90% confidence estimate of \$629 million.

14.4 DESIGN COSTS

The design and licensing development costs are based on the development of a standardized plant design, preparation and submittal of the Preliminary and Final Standard Safety Analysis Reports (PSSAR and FSSAR) to the Nuclear Regulatory Commission (NRC), and interactions with the NRC to obtain Preliminary Design Approval (PDA) and Final Design Approval (FDA). Additional design efforts required to manufacture components and construct the Lead Module are included in the Lead Module capital cost estimate (Section 13).

Design development costs are estimated at \$436, \$514, and \$549 million for the SC, IDC, and DC, respectively. Turbomachinery design costs account for a large fraction of the design development cost increase relative to the SC, with the IDC turbomachine design costs estimated at \$41 million and the DC estimated at \$90 million. Plant level design and analysis activities, reactor design, and licensing activities are also envisioned to be higher for the GT-MHR concepts.

SECTION 15

EVALUATION

The direct and indirect cycle gas turbine options were evaluated against the reference steam cycle plant, using a variant of the Kepner-Tregoe methodology. An enhanced 475 MWt steam cycle plant was also evaluated; however, the level of technical detail available was less than that for the other three concepts. The approach to the evaluation and the results of the evaluation are outlined in the following subsections.

15.1 EVALUATION METHODOLOGY

The Kepner-Tregoe methodology is a semi-quantitative approach which begins with the identification of evaluation criteria. The evaluation criteria are structured within two categories, "Must" and "Want". The Must criteria are considered essential and must be met if the option is to be given further consideration. The evaluation of Must criteria consists simply of assigning a "Yes" or "No" response to each criterion. Examples of Must criteria selected for the MHR evaluation include:

- Essential Safety Characteristics
- Minimum Availability/Capacity Factors
- Minimum Economic Performance Factors
- Acceptable Levels of Development Cost, Schedule and Risk
- Acceptable Basis for Engendering the Support of Utilities/Users, Industry and Government

The "Want" criteria are used to evaluate desirable characteristics in those options that meet the minimum Must requirements. They may be extensions of the Must criteria or may address other issues. For example, one Want criterion for the MHR addressed additional margin for protection of the public above the already conservative Must requirement, while another related to the relative ease of plant expansion after the first module is placed in service. The Want criteria evaluation is semi-quantitative, with each criterion being assigned a weighting factor characterizing its perceived importance. The evaluation is conducted by assigning a raw score of 1 to 10 to each option for each criterion. A score of 10 is assigned to the option which best meets the criterion. A relative score between 1 and 10 is assigned to each of the other options. For each criterion and option, the raw score is multiplied by the weighting factor and the totals are added to obtain a total score for each option.

The third step in the Kepner-Tregoe method consists of evaluating the consequences of prospective decisions indicated by the first two steps, a "sanity check", in essence. This can be a particularly important step if the evaluation criteria are not appropriately selected and weighted.

It was recognized from the start that any formal decision methodology is easily misapplied. In the case of Kepner-Tregoe, for example, the selection and weighting of decision criteria are particularly critical. Despite these difficulties, formal decision methodologies, such as Kepner-

Tregoe, at least provide a basis for structuring the evaluation and for enhancing the prospects that the important issues will be considered.

15.2 EVALUATION CRITERIA

The Must and Want criteria selected for the gas turbine evaluation are provided in Tables 15-1 through 15-4. As can be seen from the tables, the criteria were structured within the following four categories:

- Compliance with Functional Requirements
- Power Cost Competitiveness
- Compatibility with Energy Generation Needs
- Acceptable Front-End Costs

The weighting factors selected for the Want part of the evaluation are summarized in Table 15-5.

15.3 RESULTS OF EVALUATION

In the case of the MHR evaluation, the Must criteria were deemed to be met by each of the four options. For some of the more subjective criteria, a definitive conclusion could not be reached, and so the concepts were given the "benefit of the doubt". In the subsequent quantitative evaluation, a clear advantage was indicated for the two gas-turbine options, relative to the two steam cycle options. The differentiation of the two categories was unambiguous in that it was consistent throughout the most important categories addressing safety, performance and economics.

In the case of the DC and IDC gas turbine concepts, the numerical results of the quantitative evaluation (Table 15-6) were not so easily interpreted. Overall, the evaluated difference separating the two concepts was negligible, approximately 1%, with the raw score indicating a slight numerical advantage for the IDC. At the detail level, however, the tradeoffs differentiating the DC and IDC concepts were more clearly evident.

In the category of "Functional Requirements", the DC was evaluated to have a small (3%) advantage in the "Plant Operations" area. This results from the perceived benefits of plant simplification, offset by the need for a higher degree of remote maintenance in the DC. The IDC advantage in the "Plant Investment Protection" (11%) and "Safety/ ALARA" categories (9%) reflects the reduced consequences of turbomachinery failure events in the secondary system. While these differences are viewed as significant, the following were noted in discussion:

- The minimum thresholds set by the Must criteria in the areas of Investment Protection and Safety/ALARA are very high.
- The gas turbine concepts retain the passive safety characteristics of the steam cycle MHR, and both the DC and IDC were rated equal or superior to the steam cycle concepts in these categories. A key factor is the elimination of the steam generator tube rupture event, the dominant consequence event for the steam cycle.

The largest differences between the two gas turbine concepts were found in the category of "Power Cost Competitiveness". The DC was given a 21% higher rating in the "Minimum Power Cost" area and a 9% higher rating in the "Minimum Economic Risk" area.

The flexibility provided by the IHX for serving other applications (e.g., process heat) resulted in a higher rating for the IDC in the area of "Compatibility with Energy Generation Needs". Although the relative difference was significant (13%), the weighting factor for this category was relatively low. With an evaluated advantage of 8%, the IDC was also acknowledged to have lower development costs in the "Acceptable Front-End Costs" category.

While the numerical results of the evaluation were somewhat ambiguous, the process was successful in that the tradeoffs were clearly evident to evaluators. It was concluded that the indirect cycle provides insufficient economic incentives, relative to the steam cycle, to warrant its development. The direct cycle, while posing the greater development challenge, offers convincing economic incentives, along with a potential for future growth as the technology evolves. As a conclusion of the evaluation, it was the unanimous recommendation of the Evaluation Team that the direct cycle GT-MHR be identified as the initial commercial objective of the HTGR Program. The Evaluation Team recommendation was formally endorsed by the participating contractors in the form of letters from corporate officers. The recommendation was also endorsed by the Gas-Cooled Reactor Associates Management Committee and approved by the U.S. Department of Energy in June 1993.

TABLE 15-1
COMPLIANCE WITH FUNCTIONAL REQUIREMENTS

MUSTS

M11 PLANT OPERATION

- Design capacity factor $\geq 84\%$
- Planned outages $\leq 8\%$

M12 PLANT INVESTMENT PROTECTION

- Forced outages $\leq 8\%$
- Long forced outages (≥ 6 months duration) $\leq 0.8\%$
- Probability of exceeding the limits imposed by safety related design conditions $< 10^{-5}$ per plant year

M13 CONTROL OF RADIONUCLIDE RELEASES

- Preclude need for sheltering or evacuation beyond EAB
- Likelihood of implementing off-site ingestion pathway emergency plan less than 10^{-5} . [Assume true for all options. Relative capability to be addressed in want criteria.]
- No reliance on control room, plant control system or auxiliary power supplies (except batteries)
- Operator action not required to prevent fuel damage or to mitigate accidents

WANTS

W11 PLANT OPERATION

- *a. Maximum capacity factor
- *b. Minimum planned outage time
- c. Ease of plant control
- d. Ease of monitoring system/component performance
- e. Operation at part load during repair or expansion

* Denotes primary evaluation measure

TABLE 15-1 (CONTINUED)

WANTS (Continued)

W12 PLANT INVESTMENT PROTECTION

- *a. Minimum forced outage time
- b. Minimum probability of extended forced outages
- c. Adequacy of design margins
- d. Maximum reliance on passive systems for plant protection
- e. Maximum response times available for plant protective actions

W13 SAFETY/ALARA MARGINS BEYOND REQUIREMENTS

- *a. Additional margin for protection of public (cumulative risk assessment)
- b. Maximum reliance on passive features to protect public
- c. Safety characteristics readily demonstrated by analysis, scale testing, or full scale demonstration testing
- d. Low man-rem exposure (occupational)

* Denotes primary evaluation measure