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HTGR-GT SYSTEMS OPTIMIZATION STUDIES

by
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MASTER

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ABSTRACT

The compatibility of the inherent features of the high-temperature gas-cooled reactor (HTGR) and the closed-cycle gas turbine combined into a power conversion system results in a plant with characteristics consistent with projected utility needs and national energy goals. These characteristics are (1) plant siting flexibility; (2) high resource utilization; (3) low safety risks; (4) proliferation resistance; and (5) low occupational exposure for operating and maintenance personnel. System design and evaluation studies on dry-cooled intercooled and nonintercooled commercial plants in the 800-MW(e) to 1200-MW(e) size range are described, with emphasis on the sensitivity of plant design objectives to variation of component and plant design parameters. The impact of these parameters on fuel cycle, fission product release, total plant economics, sensitivity to escalation rates, and plant capacity factors is examined.

INTRODUCTION

The design of the high-temperature gas-cooled reactor closed-cycle gas turbine (HTGR-GT) has evolved over the past seven years (1), and a reference design was developed during 1977 and 1978. This design consisted of a 3000-MW(t) nonintercooled plant with three power conversion loops (2). An updated version of this plant was developed in 1979. This version formed the basis for the computer design and cost algorithms used in the plant optimization studies reported herein.

The nonintercooled plant was selected for the reference plant in 1977 because of its compatibility with cogeneration (bottoming cycle or process heat) and dry cooling and the desire to keep the nuclear heat source portion of the plant as simple as possible. The intercooled plant offered some performance advantages, but because of anticipated parasitic pressure losses, the performance gain was not of sufficient magnitude to compensate for its increased cost and complexity. Intercooling was therefore not selected or fully evaluated during the period when the reference design was being developed (1972 to 1978). With the increased emphasis on the gas turbine in 1979 and the apparent lack of utility interest in the bottoming cycle, it was felt that the intercooled cycle should be studied in more detail.

During the course of the 1979 effort, Gas-Cooled Reactor Associates (GCRA) made major contributions to the program. Of significance were their recommended plant size [approximately 800 MW(e)] and requirement that the turbine be capable of removal without displacement of the electric generator. These have a large impact on the intercooled/nonintercooled evaluation. In particular, the turbine removal requirement dictated a two-loop configuration. The layout characteristics of the intercooled plant are such that it is only competitive in the two-loop configuration, particularly from the viewpoint of scaling. The 1979 plant design configuration studies for dry-cooled intercooled and nonintercooled plants are reported elsewhere (3). This paper describes the parameter-related studies (for defining the optimal plant parameters for the two cycles) conducted during 1979 to support individual cycle configuration development leading to selection of one cycle.

OPTIMIZATION AND DESIGN EVALUATION

The HTGR-GT program is presently in the initial stages of cycle selection and conceptual design. System parameter selection, accomplished during optimization, has a dramatic effect on conceptual design by influencing overall plant performance, availability, safety, and economics. It is therefore appropriate and necessary to determine at the earliest stages of the design the sensitivity of these factors to changes in plant parameters. As the design progresses, additional guidance will be required to maintain a cost-effective design. This will be done by evaluating performance margins, the probability of achieving rated performance, and design alternatives.

The function of systems optimization and design evaluations during the conceptual design stage are to identify (1) an optimized set of parameters for further design development, (2) sensitive areas where additional design definition or improvements would be effective, and (3) where margins can most effectively be applied to achieve the desired probability of performance success at minimum cost. To accomplish this, sensitivity evaluations must be conducted on plant capital and fuel cycle costs, availability/maintenance, plant siting flexibility, and safety. The influence of the latter three items on the direction of the design of the plant has increased as their importance to the bottom line costs of the plant has been recognized. In particular, siting flexibility and availability have received more attention in recent years.

A design and cost evaluation computer code, CODER, has been used for the sensitivity evaluations in which the variables being evaluated interact with one another. CODER was developed using generic subroutines where applicable and developing specific subroutines to achieve a model which represents the overall plant at various levels of detail. In general, the level of detail is representative of the influence of the component on the plant evaluation factors.

CODER is not intended to be used to prepare a detailed design configuration or a cost estimate for the HTGR-GT. Rather, it is used to guide and evaluate from a fixed reference point the changes in plant characteristics resulting from changes in design parameters. To accomplish this, the base case reference design and cost data or reference point for further CODER operations is developed by combining design and cost algorithms and available cost and design information. The data generated by the algorithms are then normalized to agree with the available design and cost information.

The subroutines having a major impact on the plant evaluation factors are those associated with providing size, configuration, thermal performance, fuel performance, and cost information for (1) the reactor core, (2) the precooler/intercooler, (3) the recuperator, (4) the PCRV, and (5) the overall plant thermodynamic cycle. The code is set up with 37 independent variables which define the design and performance. It has the capability to optimize several of these variables simultaneously within broad bounds. The dependent variables are computed from the algorithms, and both are used in the subroutines for determining plant costs.

CODER generates the associated sizing, performance, and cost information for alternate parameters based on the application of reference case component configurations. Evaluations of alternate component configuraitons must be accomplished by external input or manipulation of the algorithms, depending on how extensive the alternate component configuration is. Table 1 gives the base case parameters for CODER evaluations of the intercooled and nonintercooled cycles. Table 2 presents the factors considered for the evaluations.

COMPONENT DESIGN CONSIDERATIONS

Since the component design details have been described previously, they will not be discussed in this paper. However, because optimization of the plant (Fig. 1) is dependent on individual component configurations, the key component characteristics are summarized below.

Helium Turbomachine

The turbomachinery design work for the HTGR-GT power plant was done by the Power Systems Division and Pratt and Whitney Aircraft Division of United Technologies Corporation (UTC) (4-7). Turbomachine conceptual designs were developed for a range of sizes, and the 400-MW(e) intercooled and nonintercooled machines were used in support of the optimization studies. The key turbomachine features are (1) a single shaft, (2) a direct drive, and (3) two bearing rotors with a single gas inlet duct (Figs. 2, 3).

Heat Exchangers

Work on the conceptual design of the heat exchangers for the HTGR-GT (8) was done by General Atomic (GA) and Combustion Engineering (CE). The thermal-hydraulic sizing and analysis was performed at GA and the mechanical design at CE.

Table 1 Base case parameters for CODER evaluations

	Intercooled CODER7 Run G01	Nonintercooled CODER8 Run H01
Reactor rating [MW(t)]	2000	2000
Number of loops	2	2
Liner option	Conventional	Conventional
Compressor pressure ratio, each	1.75	N/A[a]
Compressor pressure ratio, overall	3.06	2.5
Compressor flow [lb/h x 10^6 (kg/h x 10^6]	3.98 (1.81)	4.65 (2.11)
High-pressure (HP) compressor outlet pressure [psia (kPa)]	1142 (78.7)	1150 (79.3)
Reactor inlet temperature [°F (°C)]	841 (449)	945 (507)
Reactor outlet temperature [°F (°C)]	1562 (850)	1562 (850)
Turbine inlet temperature [°F (°C)]	1560 (849)	1560 (849)
Low-pressure (LP) compressor inlet temperature [°F (°C)]	79.6 (26.4)	80.1 (26.7)
HP compressor inlet temperature [°F (°C)]	80.0 (26.7)	N/A
Minimum cycle helium temperature [°F (°C)]	79.0 (26.1)	79.0 (26.1)
Primary system pressure loss [psi (kPa)]	72.5 (5.0)	76.7 (5.3)
Recuperator HP ΔP [psi (kPa)]	12.9 (0.89)	15.5 (1.07)
Core ΔP [psi (kPa)]	12.5 (0.86)	16.6 (1.14)
Turbine ΔP [psi (kPa)]	701 (48.3)	613 (42.3)
Recuperator LP ΔP [psi (kPa)]	5.9 (0.41)	11.0 (0.76)
Precooler ΔP [psi (kPa)]	6.9 (0.48)	5.6 (0.39)
LP compressor pressure rise [psi (kPa)]	284 (19.6)	N/A
HP compressor pressure rise [psi (kPa)]	489 (33.7)	N/A
Total compressor pressure rise [psi (kPa)]	773 (53.3)	690 (47.6)
Recuperator effectiveness	0.898	0.898
Turbine isentropic efficiency (%)	92.2	91.8
HP compressor isentropic efficiency (%)	90.2	89.8[b]
LP compressor isentropic efficiency (%)	90.8	
Generator efficiency (%)	98.8	98.8
Turbine blade cooling flow (%)	0	0
Turbine disc cooling flow (%)	2.96	2.96
Ambient air temperature [°F (°C)]	59 (15)	59 (15)
Cooling water temperature [°F (°C)]	69 (20.6)	69 (20.6)
Precooler outlet water temperature [°F (°C)]	188.5 (86.9)	270 (132.2)
Intercooler outlet water temperature [°F (°C)]	150.0 (65.6)	N/A
Heat loss [MW(t)]	9.93	11.87
Auxiliary power [MW(t)]	8.0	8.0
Plant efficiency (%)	41.6	38.0
Net electrical output [MW(e)]	832	759

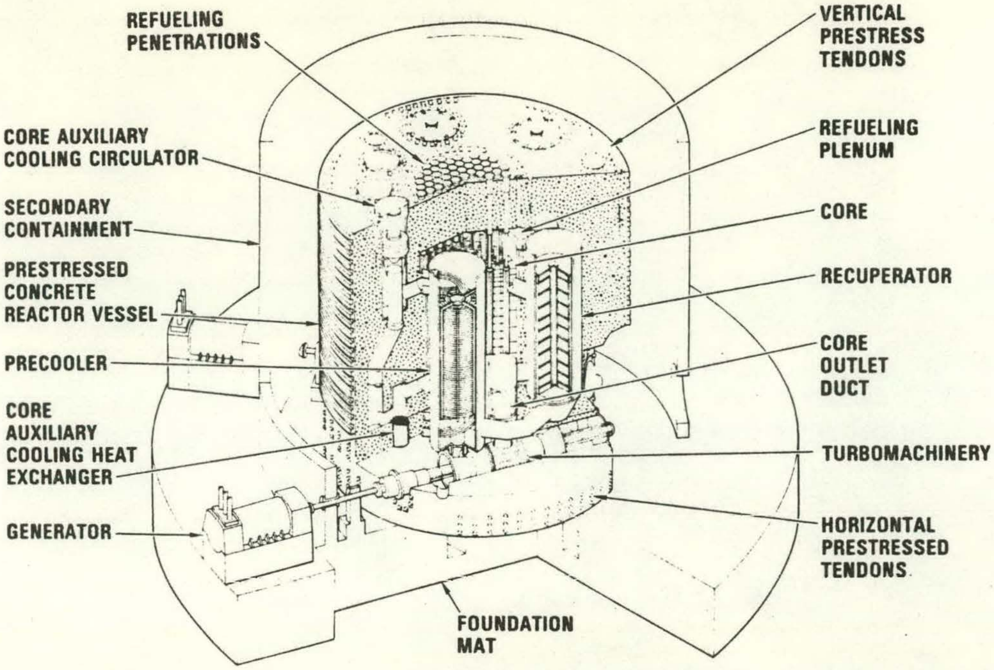
[a]N/A = not applicable.
[b]There is onle one compressor in the nonintercooled case.

Table 2 Factors used for CODER evaluation

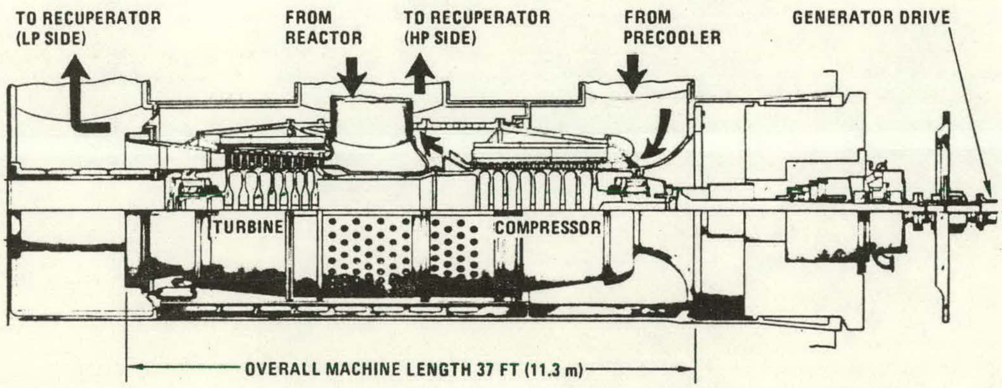
Configuration	Single unit, two power conversion loops
Schedule	
Project commitment	September 1982
Start of construction	October 1987
Commercial operation	June 1995
Construction site	Eastern Pennsylvania
Fuel cycle costs[a]	
U_3O_8 price	Variable \$50/lb (\$110/kg) to a high of \$100/lb (\$220/kg)
Inflation escalation rate	6%/yr
Enrichment price	\$100/standard work unit in 1979
Discount rate	10.2%/yr
Working capital rate	15.6%/yr
Tail assay	0.20%
Spent fuel shipping cost	\$1760/fuel element in 1979
Reprocessing cost	\$4550/fuel element in 1979
Waste disposal cost	\$2300/fuel element in 1979
Refabrication cost	\$7760/fuel element in 1979
Fuel manufacturing escalation indices	Graphite WPI-061-Ind Chemicals-Materials SIC-2819-labor
Capacity factor	80%
Plant capital	
Fixed charge rate	18%/yr
Interest during construction	9%/yr
Capacity factor	80%

[a]Levelized over 15 yr and dicounted to startup year.

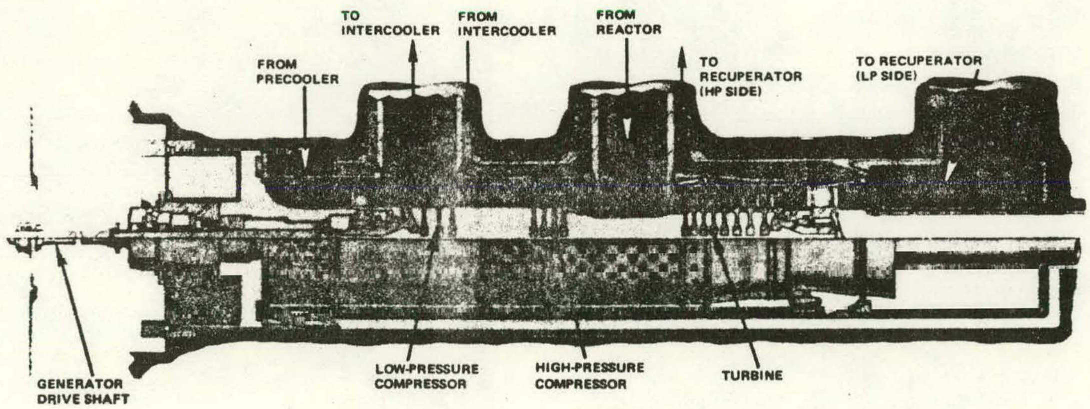
The helium-to-helium recuperator concept (Fig. 4) remains unchanged from that reported in Ref. 9, namely, a straight-tube, counterflow arrangement,



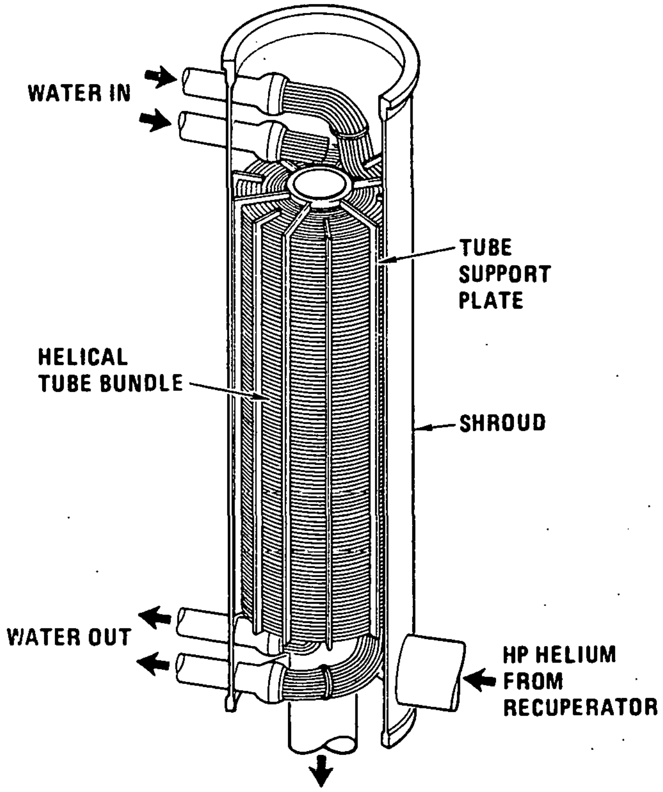
1. Two-loop 2000-MW(t) HTGR-GT



2. 400-MW(e) nonintercooled turbomachine for HTGR-GT



3. 600-MW(e) intercooled turbomachine for HTGR-GT



4. Straight tube, modular, counterflow recuperator concept for HTGR-GT

with the overall assembly made up by combining several individual hexagonal modules. The helium-to-water heat exchangers (precooler and intercooler) reported in Ref. 9 were modified to a cross-counterflow helical bundle configuration to provide higher water flow velocities with fewer tubes. This configuration is shown in Fig. 5.

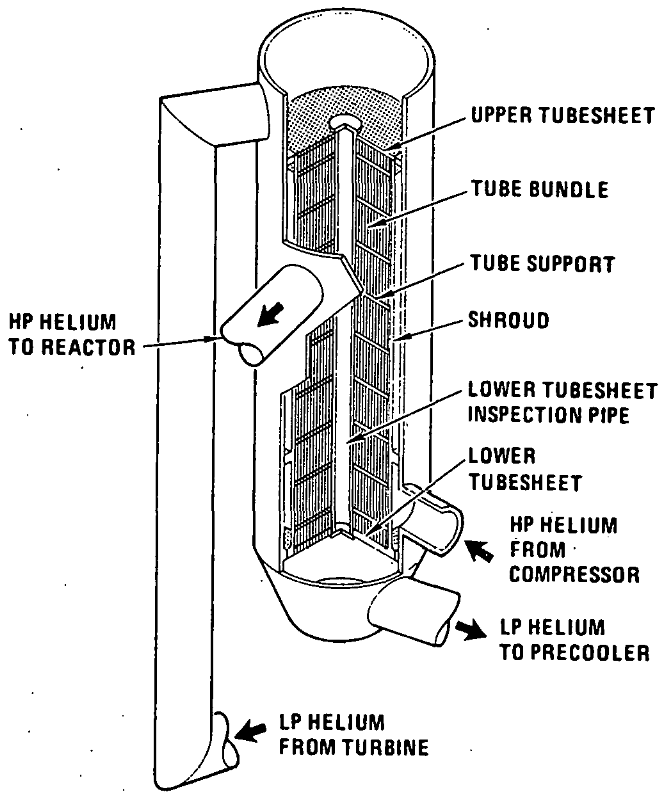
Prestressed Coincrete Reactor Vessel

The two-loop prestressed concrete reactor vessel (PCRV) arrangement was developed with the objective of obtaining a minimum size while providing for turbine removal without removal of the electric generator. The cant angle of the turbomachines was adjusted so that it is parallel to minimize the design problems at the containment interface. This arrangement induces a small size penalty on the PCRV compared with that obtained when the plant is optimized without this constraint. The nonintercooled PCRV configuration is shown in Figs. 6 and 7, and the intercooled version in Figs. 8 and 9.

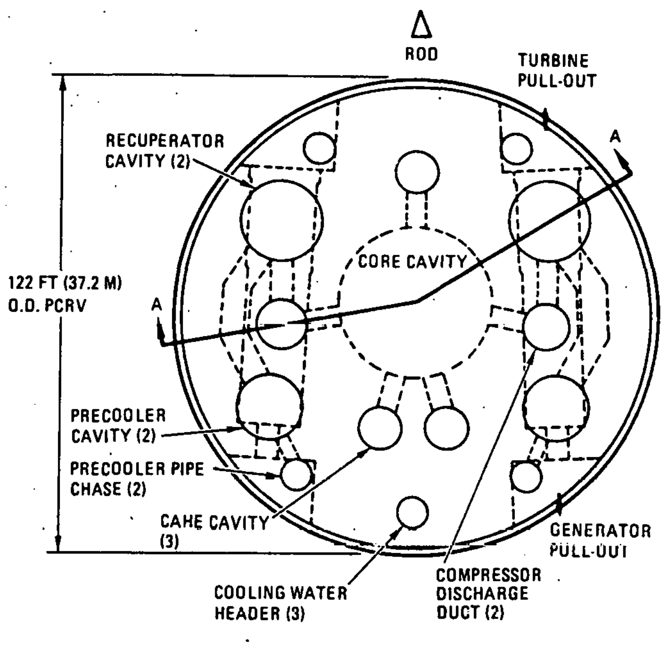
GENERAL PARAMETER SENSITIVITY RESULTS

Turbine Inlet Temperature Variations

The turbine inlet temperature was set at 1560°F (850°C) over five years ago, and subsequent studies have used this value. Turbine inlet temperature is the most important single parameter because of its pronounced effect on cycle efficiency. In the recent evaluations, turbine inlet temperature was varied

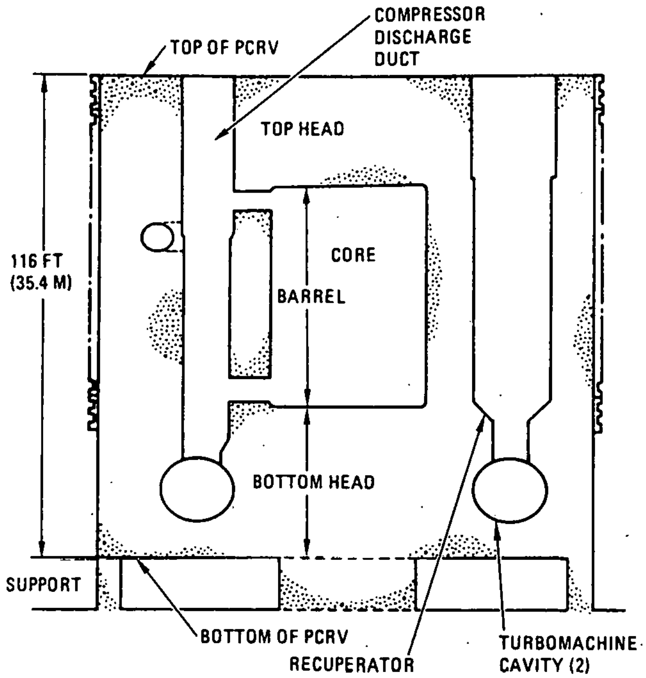


5. Helical bundle, cross-counterflow, helium-to-water heat exchanger (precooler and intercooler) for HTGR-GT

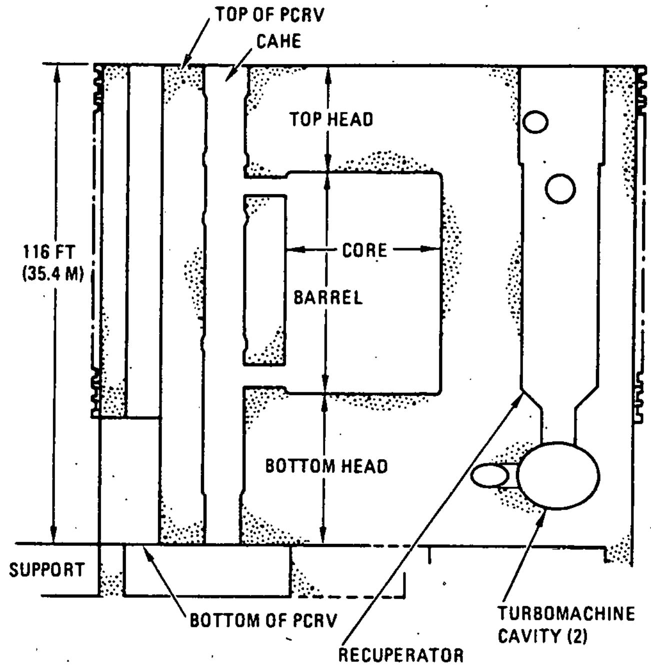


6. Nonintercooled top head plan

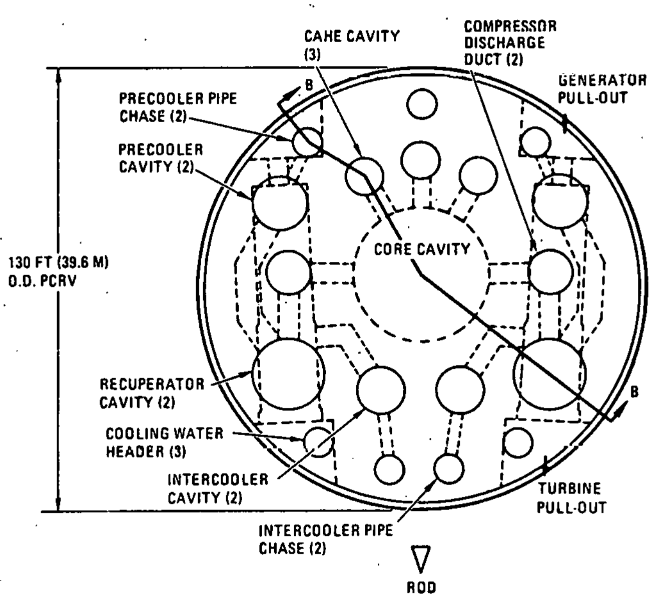
between 1472° and 1652°F (800° and 900°C). Figure 10 shows the impact of this variation on total power generation costs. This curve was developed for the nonintercooled cycle; however, the intercooled cycle shows the same general trend and sensitivity.



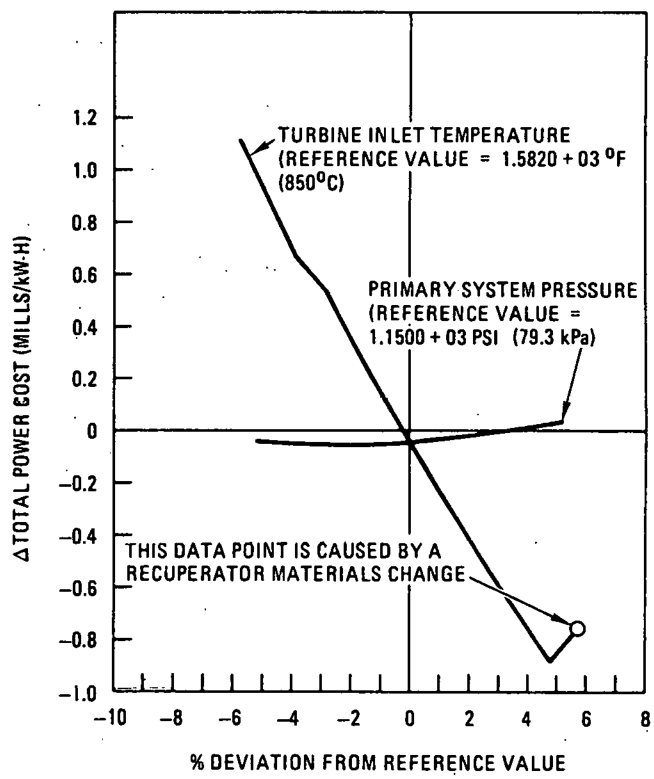
7. Nonintercooled vertical section



9. Intercooled vertical section



8. Intercooled top head plan



10. Turbine inlet temperature and system pressure variations vs change in power cost at 80% capacity factor

To gain a true perspective on the trade-offs associated with increased temperatures, costs must be weighed against increased performance benefits in terms of availability and maintenance penalties (effects of higher operating temperatures on lifetime degradation). An evaluation of these factors combined with the technical risks of higher temperatures led to the selection of 1560°F (850°C) as the design turbine inlet temperature for the nonintercooled and intercooled plants.

Primary System Pressure Variations

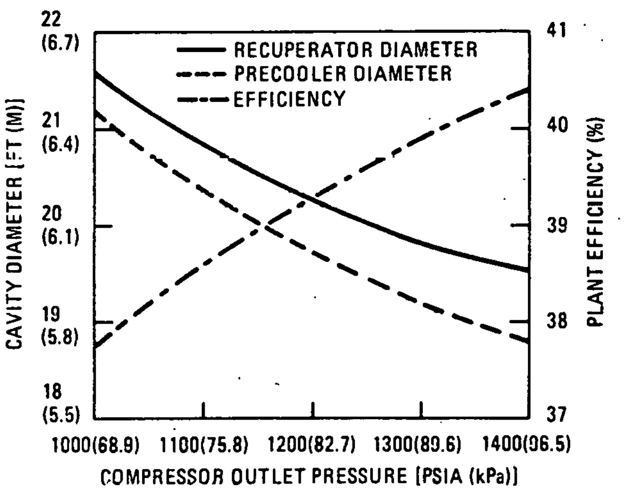
The primary system compressor discharge pressure was varied from 1090 psi (75.1 kPa) to 1210 psi (83.4 kPa), with the nominal value being 1150 psi (79.3 kPa). Figure 10 shows the impact of this variation on total power generation costs. There is little incentive in terms of total power cost to increase pressure above the nominal design value. The penalty for reduced pressure is also small. Figures 11 and 12 show the impact of pressure changes on total capital cost. As pressure increases, optimum recuperator and precooler diameters decrease and plant efficiency increases (Fig. 11). The PCRV diameter responds to the decreasing heat exchanger cavity dimensions until the ligament required to contain the pressure starts to govern. Figure 12 gives the PCRV diameter and plant cost. The present status of plant design, in which significant uncertainty is associated with parasitic pressure drop calculations, suggests that the design pressure be slightly higher than the pressure at the minimum of the capital cost curve.

Compressor Pressure Ratio Variations

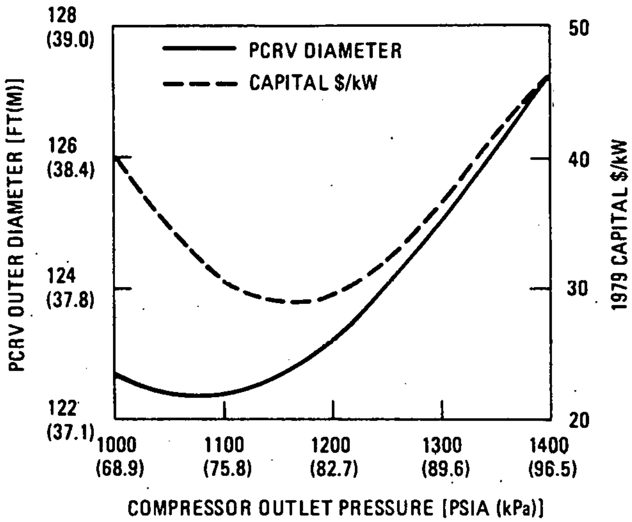
For the nonintercooled cycle, compressor pressure ratio was varied between 2.3 and 2.7, with a nominal value of 2.5. The pressure ratio of the individual compressors of the intercooled cycle was varied between 1.5 and 2.0, with a nominal value of 1.75. With the top system pressure established at 1150 psi (79.3 kPa), the compressor pressure ratio determines the system low-pressure conditions and thereby influences cycle efficiency as well as PCRV ligament sizing requirements. Figure 13 identifies the cost sensitivity of the compressor pressure ratio for the nonintercooled cycle. The intercooled cycle trends are similar.

Recuperator Effectiveness Variations

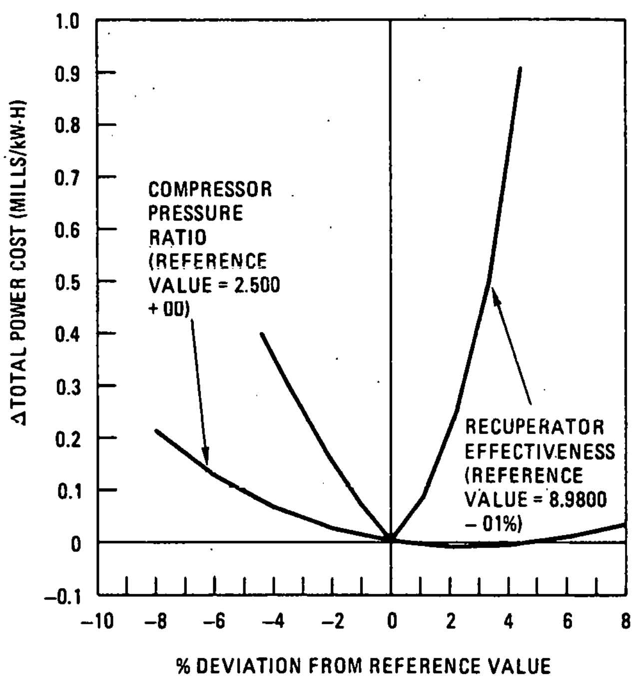
Recuperator effectiveness was varied between 85.8 to 93.8, with a nominal value of 89.8. Figure 13 shows the cost sensitivity of recuperator effectiveness. The cost penalty associated with increases in effectiveness are significant because of the exponential heat transfer surface relationship to heat duty. Recuperator effectiveness is a very influential parameter for plant efficiency, and therefore high recuperator effectiveness is desired.



11. HTGR-GT two-loop optimized plant; dry cooling tower. These curves represent CODER optimizations of capital mills/kWh for fixed compressor outlet pressures. The dry cooling tower approach temperature was 10°F (5.56°C) with a dry bulb temperature of 59°F (15°C).



12. HTGR-GT two-loop optimized plant; dry cooling tower. These curves represent CODER optimizations on capital mills/kWh for fixed compressor outlet pressures. 1979 capital dollars include direct and indirect capital costs.



13. Compressor pressure ratio and recuperator effectiveness variations vs change in power cost at 80% capacity factor

Fuel Cycle Variations

The fuel cycle influences parameter selection because of its influence on plant performance, power generation costs, and fission product release. Compared with the 4-yr cycle, the 3-yr fuel cycle has a greater fuel cycle cost; however, fission product release is less and thermal performance tends to be higher. Thermal performance is influenced by the lower radial peaking factor with the 3-yr cycle which reduces parasitic pressure loss. The 3-yr cycle decreases the residence time for the fuel elements, thereby decreasing the fuel particle failure fraction. Power density was varied between 6 and 8 W/cm^3, with a nominal value of 7 W/cm^3. Figures 14 and 15 show power generation cost versus power density for the 3- and 4-yr fuel cycles, respectively.

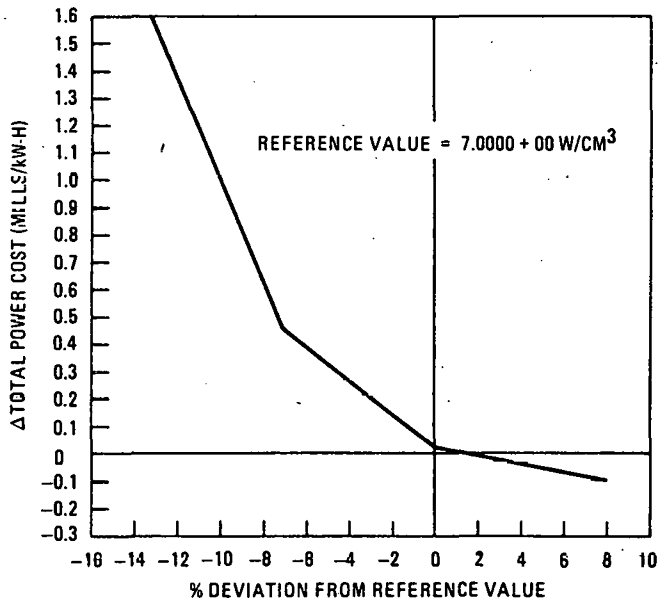
Performance Influence Coefficients

Influence coefficients can serve a valuable function, but they can, and often are, misused. They provide guidance for initial designs and consequences of contemplated design changes. However, the influence coefficients are only valid over a short range, assuming that the particular parameter is varied with all others remaining constant. The influence coefficients (Δ efficiency/Δ variation) can easily be developed from Table 3, which shows how varying certain parameters affects performance.

Parameter Optimization

CODER was used in the optimization mode to evaluate multiparameter variations. Table 4 gives optimized parameters for intercooled and nonintercooled design studies.

A major difference between the plant designs and resultant optimized parameters is due to the changes in the pressure drops within the primary coolant system. Table 5 gives the pressure drops and the changes relative to the base case design.



14. Power density variations for a 3-yr fuel cycle vs change in power cost at 80% capacity factor

ACKNOWLEDGMENTS

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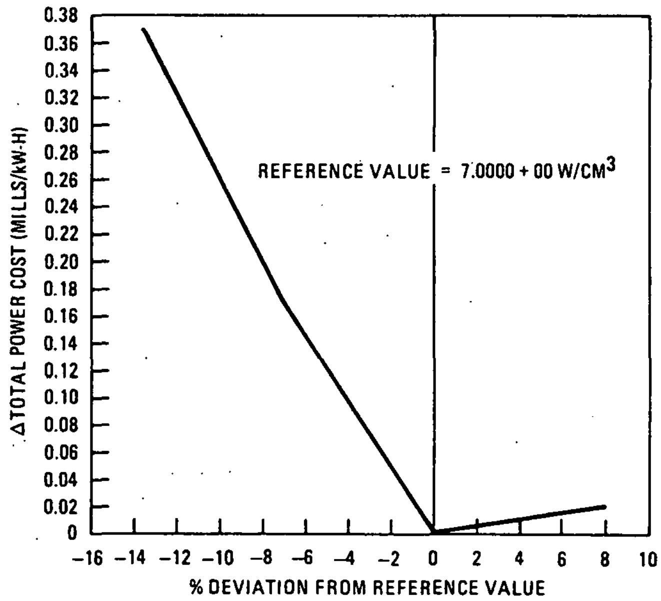

15. Power density variations for a 4-yr fuel cycle vs change in power cost at 80% capacity factor

Table 3 Impact of varying certain parameters on performance

Parameter	Variation Point	Δ Point of Efficiency
Compressor efficiency	+1%	+0.46
Turbine efficiency	+1%	+0.62
System pressure loss	+1%	-0.51
Maximum system pressure	+1%	+0.09
Compressor pressure ratio	+1%	-0.01
Reactor outlet temperature	10°F (5.56°C)	+0.22
Precooler outlet temperature (ambient air temperature)	10°F (5.56°C)	-0.73
Recuperator temperature efficiency	+1%	+0.17
Front turbine cooling (blade cooling)	+1%	-0.54
Rear turbine cooling (disc cooling)	+1%	-0.64

Table 4 Optimized parameters for intercooled and nonintercooled design studies

	Intercooled Case	Nonintercooled Case
Reactor rating [MW(t)]	2000	2000
Number of loops	2	2
Liner option	Conventional	Conventional
Compressor pressure ratio, (LP/HP)	1.75/1.80	—
Compressor pressure ratio overall	3.15	2.6
Compressor flow [lb/h x 10^6 (kg/h x 10^6]	3.93 (1.78)	4.47 (2.03)
High-pressure (HP) compressor outlet pressure [psia (kPa)]	1250 (86.2)	1220 (84.1)
Reactor inlet temperature [°F (°C)]	833 (445)	1922 (494)
Reactor outlet temperature [°F (°C)]	1562 (850)	1562 (850)
Turbine inlet temperature [°F (°C)]	1560 (849)	1560 (849)
Low-pressure (LP) compressor inlet temperature [°F (°C)]	79.6 (26.4)	80.2 (26.8)
HP compressor inlet temperature [°F (°C)]	80.0 (26.7)	
Minimum cycle helium temperature [°F (°C)]	79.0 (26.1)	60.9 (4.20)
Primary system pressure loss [psi (kPa)]	64.3 (4.43)	60.9 (4.20)
Recuperator HP ΔP [psi (kPa)]	7.1 (0.49)	9.3 (0.64)
Core ΔP [psi (kPa)]	11.1 (0.77)	14.5 (1.00)
Turbine ΔP [psi (kPa)]	792 (54.6)	690 (47.6)
Recuperator LP ΔP [psi (kPa)]	13.1 (0.90)	10.5 (0.72)
Precooler ΔP [psi (kPa)]	1.9 (0.13)	1.6 (0.11)
LP compressor pressure rise [psi (kPa)]	301 (20.8)	751 (51.8)(b)
HP compressor pressure rise [psi (kPa)]	556 (38.3)	
Recuperator effectiveness	0.898	0.905
Turbine isentropic efficiency (%)	92.2	91.8
HP compressor isentropic efficiency (%)	90.2	89.8(b)
LP compressor isentropic efficiency (%)	90.8	
Generator efficiency (%)	98.8	98.8
Turbine blade cooling flow (%)	0	0
Turbine disc cooling flow (%)	2.96	2.96
Ambient air temperature [°F (°C)]	59 (15)	59 (15)
Cooling water temperature [°F (°C)]	69 (20.6)	69 (20.6)
Precooler outlet water temperature [°F (°C)]	188.5 (86.9)	270 (132)
Intercooler outlet water temperature [°F (°C)]	150.0 (65.6)	—
Heat loss [MW(t)]	9.79	17.33
Auxiliary power [MW(t)]	8.0	8.0
Plant efficiency (%)	43.5	40.0
Net electrical output [MW(e)]	869	800

(a)Conditions: no blade cooling for turbomachine; standard day ambient air temperature.
(b)There is only one compressor in the nonintercooled case.

Table 5 Changes in pressure drops within primary coolant system

	Base Case ΔP [psi (kPa)]	Optimized Case ΔP [psi (kPa)]	Change [psi (kPa)]
Nonintercooled case			
Core	16.6 (1.14)	14.5 (1.00)	-2.1 (-0.14)
Core to turbine duct	7.6 (0.52)	7.0 (0.48)	-0.6 (-0.04)
Turbine to recuperator duct	2.9 (0.20)	2.6 (0.18)	-0.3 (-0.02)
Recuperator	15.5 (1.07)	10.5 (0.72)	-5.0 (-0.35)
Recuperator to precooler duct	1.3 (0.09)	1.3 (0.09)	0 (0)
Precooler	5.6 (0.39)	1.6 (0.11)	-4.0 (-0.28)
Precooler to compressor duct	1.6 (0.11)	1.5 (0.10)	-0.1 (-0.01)
Compressor to recuperator duct	9.0 (0.62)	8.1 (0.56)	-0.9 (-0.06)
Recuperator	11.0 (0.76)	9.3 (0.64)	-1.7 (-0.12)
Recuperator to core duct	5.4 (0.37)	4.6 (0.32)	-0.8 (-0.05)
Total	76.5 (5.27)	61.0 (4.20)	-15.5 (-1.07)
Intercooled case			
Core	12.5 (0.86)	11.1 (0.77)	-1.4 (-0.09)
Core to turbine duct	6.9 (0.48)	6.4 (0.44)	-0.5 (-0.04)
Turbine to recuperator duct	2.3 (0.16)	2.1 (0.14)	-0.2 (-0.02)
Recuperator	12.9 (0.89)	13.1 (0.90)	+0.2 (+0.01)
Recuperator to precooler duct	0.8 (0.06)	0.8 (0.06)	0 (0)
Precooler	6.9 (0.48)	1.9 (0.13)	-5.0 (-0.35)
Precooler to low-pressure compressor (LPC) duct	1.2 (0.08)	1.1 (0.08)	-0.1 (0)
LPC to intercooler duct	5.5 (0.38)	5.4 (0.37)	-0.1 (-0.01)
Intercooler	2.2 (0.15)	0.7 (0.05)	-1.5 (-0.1)
Intercooler to high-pressure compressor (HPC) duct	2.7 (0.19)	2.6 (0.18)	-0.1 (-0.01)
HPC to recuperator duct	8.8 (0.61)	8.3 (0.57)	-0.5 (-0.04)
Recuperator	5.9 (0.41)	7.2 (0.50)	+1.3 (+0.09)
Recuperator to core duct	4.0 (0.28)	3.8 (0.26)	-0.2 (-0.02)
Total	72.6 (5.03)	64.5 (4.45)	-8.1 (-0.58)

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