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HTGR GAS TURBINE PROGRAM

**SEMIANNUAL PROGRESS REPORT
FOR THE PERIOD APRIL 1, 1978
THROUGH SEPTEMBER 30, 1978**

**by
PROJECT STAFF**

**Prepared under
Contract DE-AT03-76SF70046
for the San Francisco Operations Office
Department of Energy**

DATE PUBLISHED: DECEMBER 1979

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ABSTRACT

This report describes work performed under the gas turbine HTGR (HTGR-GT) program, Department of Energy Contract DE-AT03-76-SF70046, during the period April 1, 1978 through September 30, 1978. The work reported covers the demonstration and commercial plant concept studies including plant layout, heat exchanger studies, turbomachine studies, systems analysis, and reactor core engineering.

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1. INTRODUCTION AND SUMMARY

This report describes the progress by General Atomic Company (GA) on the gas turbine HTGR (HTGR-GT) program, Department of Energy (DOE) Contract DE-AT03-76-SF70046, for the period April 1, 1978 through September 30, 1978. The report covers the demonstration and commercial plant activities including plant layout studies, heat exchanger studies, turbomachine studies, systems analysis, and reactor core engineering. Previous work on the HTGR-GT program has been reported in Refs. 1-1 through 1-9.

A summary of the demonstration plant configuration studies is given in Section 2. These studies supplement those previously reported in Ref. 1-9 and cover cooling schemes which utilize the warm liner concept in only the core and intercooler cavities. As a result of the design study, configuration VI was selected for further analysis by GA. This concept utilizes a total warm liner, full-size components, a chordal turbomachine, and an off-set core.

Based on the selected concept (configuration VI), demonstration plant structural studies were conducted as summarized in Section 3. The areas covered include prestressed concrete reactor vessel (PCRV) design, liner design, reactor internals design, and thermal barrier design. Of specific concern is the support of the core barrel in a seismic event, adequate liner cooling, and duct integrity in the annulus surrounding the core barrel.

Section 4 covers the steady-state and dynamic system analysis associated with the demonstration plant. The computer code CODER (cost optimization/performance) is in the process of being modified to cover the demonstration plant configuration. In addition, the transient code REALY2 is being modified. Transient events including total loss of cooling water, total loss of load, and bypass valve failure have been analyzed.

The reactor core engineering effort for the 1500-MW(t) demonstration plant core is summarized in Section 5. This core study is the first to investigate medium enriched uranium (MEU) fuel in detail. The peak fuel rod centerline temperature is 1205°C (2200°F). The overall thermal behavior of the core is good.

Section 6 summarizes the demonstration plant heat exchanger activities by Combustion Engineering under subcontract to GA. General arrangement drawings and design description documents were developed for the straight-tube recuperator, the helical precooler, and the helical intercooler.

Section 7 summarizes the work by United Technologies Corporation under subcontract to GA. This work covers the 620-MW(e) intercooled turbomachine for the demonstration plant studies utilizing the warm liner concept and also for the two-loop alternate commercial plant studies utilizing the conventional insulated liner concept.

Section 8 covers conceptual design studies related to a two-loop alternate commercial plant design which utilizes the same basic turbomachine and heat exchangers as the demonstration plant but which utilizes a conventional insulated liner rather than the warm liner. These studies led to the selection of a concept which was developed in more detail, as discussed below.

Section 9 covers the structural work accomplished related to the conceptual design of the two-loop alternate commercial plant design. As noted above the work was based on a conventional liner. However, a brief study was made of a plant with a conventional liner in the core area and a warm liner in the remaining cavities in an effort to minimize the thermal barrier because of the concern over oil saturation in the event of a turbomachine oil leak.

The systems work associated with the two-loop intercooled alternate commercial plant is covered in Section 10. The plant efficiency is 42.09% as compared with 39.55% for the reference commercial plant which is non-intercooled.

Section 11 briefly reviews the commercial plant auxiliary system specifications developed during the reporting period. These include the auxiliary circulator service system, the helium purification system, and the gas waste system.

The alternate commercial plant reactor core engineering is summarized in Section 12. A 3000-MW(t) prismatic core with a 6.5 W/cm^3 power density was studied.

Section 13 covers the alternate commercial plant heat exchanger effort. These heat exchangers are basically the same as those used in the demonstration plant. The only difference is caused by the utilization of conventional thermal barriers in the heat exchanger cavities.

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2. DEMONSTRATION PLANT CONCEPTUAL DESIGN STUDIES

2.1. GENERAL

A conceptual design study has been performed for the one-loop 600-MW(e) demonstration plant for the purpose of selecting the best demonstration plant arrangement with regard to PCRV design, position and size of components, maintenance capability, adaptability to a follow-on commercial plant, and plant performance and cost. Configurations I through XI of this study were reported in the last quarterly report (Ref. 2-1). Since then, additional configurations (XIIa, XIIb, XIII, and XIV) have been investigated; these are shown in Figs. 2-1 through 2-4 and described in Table 2-1.

Consistent with the project ground rules, all of the configurations evaluated embodied the warm liner feature in the core cavity. Incorporation of the warm liner principle (conceived in Germany) substantially complicates the plant primary system layout.

From the conceptual studies essentially two families of designs emerged: (1) configurations I through IX with elimination of the thermal barrier in the major component cavities (heat exchangers and turbomachine) and (2) configurations X through XIV with conventional thermal barrier in the major component cavities.

In order to select the best demonstration plant concept, all configurations were evaluated on the basis of a simplified rating system which included:

1. Primary system gas flow path complexity (strongly influenced by major cavity thermal barrier considerations).
2. Core cavity - bottom- or side-fed high-pressure gas inlet (flow distribution and core support considerations).

Fig. 2-1. Configuration XIIa - 600-MW(e) HTGR-GT one-loop demonstration plant (sheet 1 of 2)

Fig. 2-1. Configuration XIIa - 600-MW(e) HTGR-GT one-loop demonstration plant (sheet 2 of 2)

Fig. 2-2. Configuration XIIb - 600-MW(e) HTGR-GT one-loop demonstration plant

Fig. 2-3. Configuration XIII - 600-MW(e) HTGR-GT one-loop demonstration plant (sheet 1 of 2)

Fig. 2-3. Configuration XIII - 600-MW(e) HTGR-GT one-loop demonstration plant (sheet 2 of 2)

Fig. 2-4. Configuration XIV - 600-MW(e) HTGR-GT one-loop demonstration plant (sheet 1 of 2)

Fig. 2-4. Configuration XIV - 600-MW(e) HTGR-GT one-loop demonstration plant (sheet 2 of 2)

TABLE 2-1
DEMONSTRATION PLANT CONCEPTUAL DESIGN CONFIGURATIONS

Configuration	Component Size ^(a)			Turbomachine Location	PCRV			Coolant Flow Path	Remarks
	Recuperator	Precooler	Intercooler		Core Cavity Offset [m(ft)]	Diameter [m(ft)]	Height [m(ft)]		
I ^(b)	F	F	F	Chordal	2.59 (8.5)	37.19 (122)	34.14 (112)	Fig. 2-5	Slanted core inlet duct
II ^(b)	F	F	F	Chordal	2.59 (8.5)	35.97 (118)	34.14 (112)	Fig. 2-5	Slanted core inlet duct Turbomachine cavity protruding
III ^(b)	F	F	F	Chordal	2.04 (6.7)	35.66 (117)	35.05 (115)	Fig. 2-5	Vertical center core outlet duct Vertical center coaxial coolant inlet
IV ^(b)	F	F	F	Diametral	0.91 (3.0)	36.27 (119)	35.05 (115)	Fig. 2-5	Same core outlet and coolant inlet Hot duct access difficult
V ^(b)	1/2	1/2	F	Diametral	0 (0)	33.53 (110)	34.14 (112)	Fig. 2-5	Same as IV
VI ^(b)	F	F	F	Chordal	2.13 (7.0)	35.66 (117)	35.05 (115)	Fig. 2-5	Vertical center coolant inlet duct Side core outlet duct
VIIa ^(b)	F	F	F	Diametral	1.07 (3.5)	35.66 (117)	35.05 (115)	Fig. 2-5	Two turbine inlet ducts Recuperator over turbomachine Slanted core inlet duct
VIIb ^(b)	F	F	F	Diametral	0.76 (2.5)	35.97 (118)	35.05 (115)	Fig. 2-5	Recuperator at side of turbomachine Horizontal core inlet
VIII ^(b)	1/2	1/2	1/2	Diametral	0.61 (2.0)	32.30 (106)	35.05 (115)	Fig. 2-5	Two vertical core outlet ducts
IX ^(b)	F	F	F	Diametral	1.22 (4.0)	35.05 (115)	35.05 (115)	Fig. 2-5	One vertical core outlet duct
X ^(b)	F	F	F	Chordal	2.29 (7.5)	35.97 (118)	35.05 (115)	Fig. 2-6	Recuperator and precooler over turbomachine Side core outlet duct
XI ^(b)	1/2	F	F	Chordal	4.11 (13.5)	36.57 (120)	35.05 (115)	Fig. 2-6	Precooler over turbomachine
XIIa	1/2	1/2	1/2	Diametral	0.91 (3)	32.30 (106)	35.05 (115)	Fig. 2-6	Double turbomachine inlet, all 1/2 size heat exchangers
XIIb	1/2	1/2	1/2	Diametral	0.91 (3)	32.30 (106)	35.05 (115)	Fig. 2-6	Single turbomachine inlet
XIII	F	F	F	Chordal	3.96 (13)	36.88 (121)	35.05 (115)	Fig. 2-6	Precooler not over turbomachine
XIV	F	F	F	Chordal	3.96 (13)	36.27 (119)	30.48 (100)	Fig. 2-6	Separate PCRV for turbomachine

(a) F = full size, 1/2 = half size.

(b) Previously reported (Ref. 2-1).

Fig. 2-5. Coolant flow diagram for full cavity coverage

Fig. 2-6. Coolant flow diagram for partial cavity coverage

3. Adaptability of configuration to two-loop commercial plant.
4. Utilization of full-size or half-size heat exchangers (particularly the recuperator).
5. Maintenance and inservice inspection considerations.
6. Plant cost (estimated since cost data have not been generated).
7. Safety and licensing.
8. Feasibility.
9. Commonality with steam cycle HTGR (HTGR-SC).

For the family of concept variants I through IX, in which the thermal barrier was eliminated to the highest degree possible, configuration VI offered the best solution. For concepts embodying conventional water-cooled and insulated liners (configurations X through XIV), configuration X or XIV appeared to be the best. Due to the fact that the ground rules requested warm liners in all cavities, configuration VI was selected.

Conceptual details of configuration VI are shown in Fig. 2-7. The arrangement shown has an offset core cavity and a chordal turbomachine orientation. Both helium-to-water exchangers are positioned so that they are not over the turbomachine cavity. The gas from the high-pressure compressor can enter from below the core or through a side entry. The cold gas [62°C (80°F)] after leaving the precooler bundle is directed back up the precooler cavity, across to the recuperator cavity, and down an annulus in the recuperator cavity; after flowing axially through the turbomachine cavity annulus, it enters the low-pressure compressor. Routing the gas in this manner (similarly in the intercooler cavity) permits all of the major cavity liners to be swept by 62°C (80°F) gas, thus eliminating the need for thermal barrier.

Fig. 2-7. Configuration VI - 600-MW(e) HTGR-GT one-loop demonstration plant (sheet 1 of 2)

Fig. 2-7. Configuration VI - 600-MW(e) HTGR-GT one-loop demonstration plant (sheet 2 of 2)

Incorporation of the swept liner feature has enormously complicated the primary system gas flow paths and component interfaces (particularly the turbomachine). An initial assessment of the pressure losses showed a system loss ($\Delta P/P$) of 13%. A tentative layout of configuration VI was prepared to estimate material quantities and to obtain an initial evaluation for the PCRV layout.

An important aspect of the demonstration plant conceptual study was its adaptability to a two-loop commercial plant. A scoping study concluded that configuration VI is adaptable to the two-loop variant.

Conceptual details of configuration X are shown in Fig. 2-8. The arrangement shown has an offset core cavity and a chordal turbomachine orientation. It can be seen that the precooler is positioned over the turbomachine cavity and that the primary system gas flow paths are much cleaner when a conventional water-cooled and insulated liner is utilized in the major component cavities. This is particularly noticeable in the turbomachine cavity where the annular flow path is obviated and a much simpler transition between the precooler and low-pressure compressor can be realized. System pressure losses have not been estimated for this variant; however, it is projected that the losses ($\Delta P/P$) will be over 2 percentage points less than for configuration VI, which will show up in improved plant efficiency.

Configuration X is also adaptable to the two-loop commercial variant. Variants of configuration X, which may have improved maintenance and inservice inspection features, are configurations XIII and XIV (Figs. 2-3 and 2-4). Positioning of the precooler so that it is not over the turbomachine cavity permits utilization of simpler vertical water pipes that penetrate the bottom head of the PCRV. Configurations XIII and XIV are both adaptable to the two-loop commercial variant.

All demonstration plant conceptual designs for this study employ an upflow coolant duct which enters the core cavity annulus in the bottom center. It was anticipated that with this arrangement an even distribution of the coolant flow over the entire core cavity liner area would be achieved. Later

Fig. 2-8. Configuration X - 600-MW(e) HTGR-GT one-loop demonstration plant (sheet 1 of 2)

Fig. 2-8. Configuration X - 600-MW(e) HTGR-GT one-loop demonstration plant (sheet 2 of 2)

investigations showed, however, that by a correctly shaped flow restricting device in the annulus of the hot duct (high-pressure compressor outlet) an even distribution in the core cavity annulus can be achieved, and the vertical bottom inlet duct was thus eliminated.

The final conceptual design of the selected configuration VI is shown in Fig. 2-9. The PCRV design is described in Section 3.

2.2. DESCRIPTION OF CONFIGURATIONS

A description of configuration I through XI studies is given in Ref. 2-1. Subsequent configurations are briefly described below.

2.2.1. Configuration XIIa (Fig. 2-1)

This configuration has a diametral orientation of the turbomachine, and the half-size heat exchangers are arranged symmetrically on either side. The gas flow is as shown in Fig. 2-6, where the gas covers only the core and the intercooling cavities. The core outlet is through two vertical ducts into either side of the turbomachine. The hot ducts are cooled by a partial flow from the high-pressure compressor, while the main flow goes through the center vertical duct into the core cavity annulus. Because of the half-size components, the PCRV diameter is small [32.6 m (106 ft)]. In the top view the centers of the core cavity and the PCRV are offset 0.91 m (3 ft).

2.2.2. Configuration XIIb (Fig. 2-2)

This configuration differs from configuration XIIa only in that it has one turbine inlet duct rather than two, as shown in Fig. 2-2. The arrangement of the turbomachine and the heat exchangers, as well as the gas flow, is the same as for configuration XIIa.

2.2.3. Configuration XIII (Fig. 2-3)

This configuration has a chordal turbomachine orientation and full-size heat exchangers. The core outlet duct is through the side of the bottom

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SECTION A - A

Fig. 2-9. PCRV configuration VI for
HTGR-GT one-loop demonstra-
tion plant (sheet 2 of 4)

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SECTION 13 - 13

Fig. 2-9. PCRV configuration VI for HTGR-GT one-loop demonstration plant (sheet 3 of 4)

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Fig. 2-9. PCRV configuration VI for HTGR-GT one-loop demonstration plant (sheet 4 of 4)

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plenum. The duct is cooled by a partial flow from the high-pressure compressor discharge, while the main flow goes through the vertical duct into the core cavity annulus. The gas flow is the same as for configuration XII. The precooler and intercooler are arranged to either side of the turbo-machine, which provides for circulating water inlet and outlet through the bottom of the PCRV. In the top view the centers of the core cavity and the PCRV are offset 3.96 m (13 ft).

2.3.4. Configuration XIV (Fig. 2-4)

This configuration is a variant of configuration XIII but has the same chordal turbomachine and heat exchanger arrangement. The difference between these two configurations is the design of the PCRV. The prestressing system of the HTGR-GT PCRV with vertical heat exchanger cavities and horizontal turbomachine cavities in a vertically oriented cylindrical body has always been a difficult problem. In the present prestressing system the PCRV is conventionally prestressed by longitudinal tendons and circumferential wire winding for the vertical cavities and by a diagonal tendon system in the area of the horizontal turbomachine cavity where wire winding can not be applied. This is an inefficient PCRV design with excess concrete and reinforcing and tendon steel for that portion of the PCRV where the horizontal turbomachine cavities are located. The PCRV design for this configuration is an attempt to ease this problem by combining two separate prestressed pressure vessels, one for the vertical heat exchanger and core cavities and one for the horizontal turbomachine cavity. The vertical PCRV is conventionally prestressed by tendons for longitudinal prestress and by a wire-winding system for circumferential prestress. The horizontal PCRV for the turbomachine cavity is a separate entity and is prestressed by longitudinal tendons in the horizontal direction and by a hoop tendon system for circumferential prestress. This turbomachine PCRV is located between the vertical PCRV and the base mat and will be structurally connected to both by an appropriate system of reinforcing bars.

For all other PCRVs, the support consists of a central pedestal and a concrete ring at the outer edge of the PCRV with circumferential openings for the generator and for access to the tendon heads and the pipe connections of the heat exchangers.

REFERENCE

- 2-1. "Gas Turbine HTGR Program, Quarterly Progress Report for the Period Ending March 31, 1978," DOE Report GA-A15002, General Atomic Company, June 1979.

3. DEMONSTRATION PLANT STRUCTURAL STUDIES

3.1. PCRV AND LINERS

The conceptual studies of the PCRV and liners for the one-loop demonstration plant were completed. Preliminary general arrangement drawings with sufficient detail to support cost estimates were released. The conceptual PCRV design incorporated the concept of the warm liner. In the absence of appropriate provisions for warm liner in the ASME Code, Section III, Division 2, a layer of 0.3 m (1 ft) thick nonstructural insulation concrete was introduced around the core cavity where the liner/concrete interface temperature is expected to reach 110°C (230°F) in the conceptual PCRV design in order to maintain the structural concrete within the existing code allowables. In some local areas, such as the ducts between the core auxiliary heat exchanger (CAHE) and core cavity and the upper portion of the recuperator cavity, the concrete may be exposed to higher than allowable temperatures. The present PCRV conceptual design has not addressed these areas for the warm liner concept.

3.1.1. PCRV Design

The diameter of the offset core PCRV was established on the basis of the estimated minimum requirement for radial ligaments between the inter-cooler cavity and the core cavity and the orientation of the turbomachine cavity in plant view. The height of the PCRV is governed by the height of the recuperator cavity and the elevation of the turbomachine cavity in the PCRV bottom head. The PCRV layout provides for four CAHEs, three cooling-water headers, and other major heat exchanger cavities. The conceptual arrangement of the PCRV is shown in Figs. 3-1 and 3-2. The PCRV is 32.67 m (119 ft) in diameter and 35.36 m (116 ft) high.

3.1.2. Liner Design

The basis for the liner design is satisfying the German HHT criteria. To meet these criteria, flow restrictors are provided in those penetrations where closure failure is postulated. Shear anchorage systems have to conform to the ASME Code, Section III, Division 2 CB; penetration closure systems conform to the requirements of Section III, Division 1 NB; and component support flanges conform to the requirements of Section III, Division 1 NG.

The major component cavity liner sizes were established by adding liner fabrication and installation tolerances together with inservice inspection requirements to the component outline dimensions presented by Combustion Engineering and United Technologies Corporation.

Scoping calculations supporting the design of the liners, penetrations, closures, cooling tube requirements, and shear anchorage systems were prepared. Scoping calculations in support of the liner and PCRVR cooling water system designs were performed based on preliminary 100% power system conditions. Normal and abnormal thermal transients were not considered as they have not as yet been defined. The results of these calculations are summarized as follows:

1. A cooling tube pitch of 152.4 mm (6 in.) for the core cavity side wall region is acceptable. In other regions nuclear heating is less severe, and a 203.2 mm (8-in.) pitch is acceptable. To be consistent with the German design, an embedment depth of 203.2 mm (8 in.) from the liner was assumed. For the other cavities which are swept with 62°F (80°F) gas, no need for cooling tubes was identified except that temperature transients, which have not as yet been defined, may require some PCRVR cooling for these cavities.
2. Based on thermal analysis results, there will be no thermal expansion of anchor studs, which tends to push the liner away from the concrete.

GENERAL NOTES:

1. THE DUCTING BETWEEN CAVITIES SHOWN ON THIS DRAWING IS SCHEMATIC ONLY AND REQUIRES VERIFICATION BY THE INTERFACING ORGANIZATIONS.
2. MINIMUM LINEAR PRESTRESSING TENDON FORCE IMMEDIATELY AFTER ANCHORING SHALL BE 1735 KIPS.
3. ALL TENDON BEARING PLATES SHALL BE GROUTED IN PLACE AFTER CONCRETING OF THE PCRV.
4. PRECAST CONCRETE PANELS SHALL BE USED AS FORMS FOR PLACEMENT OF PCRV CONCRETE.
5. PCRV SIZING IS BASED ON THE CAVITY ENVELOPE INFORMATION DOCUMENTED IN MEMO NO. SED-PB-290-78.
6. ALL SPECIFICATIONS FOR THE 900 MW(2) HTGR-SC PLANT ARE APPLICABLE TO THIS DRAWING.
7. DIMENSIONS AND OPENINGS SHOWN IN THE PCRV SUPPORT STRUCTURE ARE SUGGESTED ARRANGEMENTS ONLY. FINAL CONFIGURATION AND LOCATION ARE PART OF THE BOP DESIGN.
8. BOTTOM HEAD HORIZONTAL TENDONS ARE NOT SHOWN ON THE PCRV TOP HEAD PLAN. INTERFERENCE OF THESE TENDONS WITH VERTICAL TENDONS AND OTHER CAVITIES AND PENETRATIONS WILL REQUIRE FURTHER INVESTIGATION.
9. NUMBER AND LOCATION OF SIDE ENTRY PENETRATIONS FOR TEMPERATURE MEASUREMENTS IN CORE REGION AND OTHER CAVITIES WILL BE DETERMINED LATER.
10. PROVIDE 3-16 IN. DIA. IN-SERVICE INSPECTION (ISI) PENETRATIONS IN TOP HEAD. SEE DWG 054477 FOR LOCATION.

Fig. 3-1. Conceptual arrangement of PCRV, plan view

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3. The movements of the junctions of the core cavity liner with the cross-duct liners can be controlled by means of suitably designed ring shear anchors.
4. Based on available information, the layer of warm concrete between the liner and embedded cooling tubes can have 90% of the strength of the 66°C (150°F) concrete, although more steel reinforcement may be necessary. Laboratory tests will be required to establish the anchor capacities and stiffness.
5. Buckling of the anchored liner does not appear to be a problem even for the recommended 12.7 mm (0.5 in.) liner thickness. For a 19.05 mm (0.75 in.) diameter stud, the area covered by one stud is limited to $32 \times 10^{-2} \text{ m}^2$ (50 in.²) as a result of the loading due to a postulated cooling tube leak.
6. The strains in the core cavity and cross-duct liners and the forces and displacements of anchor studs will apparently meet the code requirements for both static and loading conditions.

Several major design issues have been identified and are summarized as follows:

1. As a result of the turbomachine cavity being located in the PCRV bottom head, circumferential prestressing is provided by means of linear horizontal tendons. Due to space limitation in this region, an extension study of the tendon arrangement, including interference and anchorage locations in the periphery of the PCRV, will be required.
2. Sizing of the PCRV ligaments is generally based on past experience on similar multicavity vessels. Due to the complexity of the vessel configuration, the conceptual PCRV sizing requires confirmation by a three-dimensional finite element analysis.

3. No provision for the warm liner concept is currently included in the ASME Code, Section III, Division 2. The requirement of 0.3 m (1 ft) of nonstructural concrete adjacent to the core cavity liner has not been established. The implication of nonstructural concrete pertaining to liner design, temperature distribution in the PCRV, and interaction between nonstructural and structural concrete requires further study involving detailed thermal analyses. The evaluation of the acceptability of structural concrete to withstand warm liner temperature would entail material studies and possibly code changes.
4. The interface of the seismic support for the major components, recuperator, precooler, and intercooler with the liner requires further study. While the present design appears satisfactory, the effect of long-term component replacement is not clear.

3.2. REACTOR INTERNALS

The study of the feasibility of the warm liner concept for the demonstration plant was pursued based on a configuration in which the liner and core barrel coolant flow enters the bottom of the main cavity from the side, through the annular space around the hot duct. A comprehensive survey of the technical problems posed by this concept to the reactor internal structures was performed. Three major areas still need to be investigated:

(1) rotational restraint of the core, (2) sealing of the core barrel, and (3) the ability to withstand the abnormal design conditions. The more fundamental problem of providing an adequate supply of cooling gas to the liner and the core barrel under all plant conditions also remains to be solved by an appropriate plant system design. The preliminary conclusion pending completion of the technical and cost studies is that the proposed reactor internals design will probably be technically feasible, but that its disadvantages far outweigh its advantages when compared with the gas turbine plant with conventional liner in the core cavity.

As the design of the reactor internals evolved, the preliminary layout drawings were revised to incorporate changes. This effort resulted in a conceptual layout drawing (Figs. 3-3a through 3-3f). This drawing was also used to support cost estimates.

In support of the feasibility study and design revision, a structural evaluation of the metallic reactor internals was conducted.

An exchange of technical questions and comments on the reactor internals for the demonstration plant was begun between GA and the German and Swiss participants in the HHT project.

3.2.1. Core Barrel Cooling

The problem of supplying and maintaining an adequate mass flow of 110°C (230°F) coolant for conditions other than steady-state normal plant operation has not yet been solved. Solution of this problem rests with the design of the core auxiliary cooling system (CACS), which is still to be produced. Two of the problems to be overcome are:

1. Maintaining the core inlet temperature during certain normal conditions at a value high enough to keep the thermal stresses of core and graphite internals within allowable limits while still providing low-temperature coolant for the liner and the core barrel.
2. Maintaining the liner and core barrel cooling function during postulated accident conditions which affect the integrity of the coolant flow path.

In the meantime, preliminary plant transient analysis data seem to indicate that the temperature of the helium inside, as well as outside, the core barrel might considerably exceed the nominal values. Should this be confirmed when all transient and abnormal plant conditions have been analyzed,

design of the metallic reactor internals and insulation of the core barrel might have to be considerably altered.

3.2.2. Core and Core Barrel Lateral Restraint

It was originally planned to support the core and the graphite internals laterally by means of inner spring packs attached to the inside of the core barrel and to tie the core barrel to the PCRV core cavity liner with outer spring packs. After careful evaluation, a combination of the alternate solutions, as shown in Fig. 3-4, was chosen instead. The spring packs are attached to the PCRV liner and extend to the permanent side reflector and the core support floor through openings in the core barrel. The horizontal seismic loads generated in the core are thus transmitted directly to the PCRV without interference with the core barrel. The "soft" section of the spring packs is compressed by thermal growth of the graphite internals so that the latter are in turn pressed tightly against one another during reactor operation, which eliminates the need for preloading the spring packs at installation. This core lateral restraint design is very similar to the one used for the HTGR-SC projects. However, the effective transverse stiffness of the proposed spring packs is reduced because of their increased length, and it will be necessary to verify whether the spring packs will provide sufficient rotational seismic restraint to the core.

Another difficulty is restricting the passage of helium through the numerous spring pack openings in the core barrel while permitting free relative movement between the core barrel and spring packs. Sliding seals are proposed, but bellows might be an acceptable alternate.

The core barrel itself is laterally restrained by means of keys attached to the PCRV core cavity liner. The keys allow only radial and vertical movements of the barrel relative to the PCRV, thus facilitating the design of the components which are connected to both the barrel and the PCRV. Another advantage is that the barrel is subjected only to much smaller seismic loads caused by its interaction with the PCRV.

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Fig. 3-3. Internals of HTGR-GT one-loop demonstration plant
(b) core barrel top

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Fig. 3-3. Internals of HTGR-GT one-loop demonstration plant
 (c) core and reflectors

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Fig. 3-3. Internals of HTGR-GT one-loop demonstration plant
(d) core support floor

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Fig. 3-3. Internals of HTGR-GT one-loop demonstration plant
(e) lower core plenum

Fig. 3-3. Internals of HTGR-GT one-loop demonstration plant
(f) core barrel floor

Fig. 3-4. Principle of alternate core restraint through the core barrel

3.2.3. Core Barrel Design

The general adequacy of the proposed core barrel design was borne out by the structural evaluation mentioned above, which was based on the ASME Code, Section III, Subsection NG - Core Support Structures. The top edge of the barrel side wall was modified to increase the safety margin against buckling under external pressure (emergency condition). This also improved the layout of the barrel top and permitted elimination of any support in addition to the existing refueling penetrations. The supports of the barrel floor were moved to the edges of the floor sections and the joints between sections were designed to avoid buildup of thermal stresses. Since horizontal seismic movement of the barrel is prevented by keys, as discussed earlier, it was possible to use simpler core barrel supports of the flexible column type used in the Fort St. Vrain (FSV) HTGR, instead of the metallic posts and seats proposed originally.

3.3. THERMAL BARRIER

Preliminary general arrangement drawings (Figs. 3-5a through 3-5e) were completed and released. These drawings include a detailed list of materials with quantities required for each thermal barrier zone. A preliminary cost estimate for the thermal barrier will be made based on the above drawings.

3.4. HOT DUCTS

3.4.1. General Considerations

The primary effort during this reporting period was directed toward configuring the lower plenum, the lower main cross duct, and the duct extension to the turbomachine (Figs. 3-6 and 3-7). The design conditions involved in this effort included duct velocity, materials, and thermal and seismic movements.

Fig. 3-5. Thermal barrier general arrangement, Sheet 1

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- NOTES :
1. THICKNESS INCLUDES METALLIC DUCT WALL.
 2. AREA GIVEN IS TOTAL AREA PER REACTOR AND IS MEASURED AT LINER SURFACE.
 3. DOES NOT INCLUDE ATTACH. FIXTURE PROTRUSION.
 4. INSTALLATION PER G.A. SPEC. NO. 900008.

SECTION C-C SH 1
SCALE : 1/64 [3/16" = 1 FT]

SECTION B-B SH 1
SCALE : 1/64 [3/16" = 1 FT]

ZONE	CLASS	QTY	THICK.	AREA	DRAWING NO.	REMARKS	SHOWN ON :
1	B1	1	101.6 m.m. [4.00"]	23.3 m ² [250 FT ²]			SH3, Z66
2	B1	1	101.6 m.m. [4.00"]	25.8 m ² [278 FT ²]			SH2, ZE1
3	B1	1	101.6 m.m. [4.00"]	25.0 m ² [266 FT ²]			SH2, Z63
4	B1	1	101.6 m.m. [4.00"]	31.5 m ² [339 FT ²]			SH3, ZE4
5				NOT USED			
6				NOT USED			
7	B2	1	127 m.m. [5.00"]	76.1 m ² [819 FT ²]	024497		SH2, ZD3
8	C	1	378 m.m. [14.88"]	80.3 m ² [864 FT ²]	024497		SH2, ZC3
9	B2	1	203 m.m. [8.00"]	32.6 m ² [351 FT ²]	024392		SH2, ZD3
10	B2	1	203 m.m. [8.00"]	49.5 m ² [522 FT ²]	024392		SH2, ZC1
11	A	1	50.8 m.m. [2.00"]	53.9 m ² [580 FT ²]			SH2, ZC7
12	B1	4	101.6 m.m. [4.00"]	91.6 m ² [984 FT ²]			SH2, ZF4
13	B1	4	101.6 m.m. [4.00"]	36.6 m ² [393 FT ²]			SH2, ZF6
14	B2	4	203 m.m. [8.00"]	138.6 m ² [1492 FT ²]			SH2, ZE4
15	B1	1	76.2 m.m. [3.00"]	95.1 m ² [1024 FT ²]			SH4, ZB2

TOTAL AREA = 1038.9 METRES²
PER REACTOR [10,957 FEET²]

Fig. 3-5. Thermal barrier general arrangement. Sheet 3

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Fig. 3-5. Thermal barrier general arrangement, Sheet 4

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MAJOR THERMAL BARRIER COMPONENTS REQUIRED PER ZONE																
ZONE	CLASS	COVER PLATE		SEAL SHEET .036" THK		SEAL ANGLE .036" THK		INSULATION BLANKET SET		BLANKET SUPPORT		CENTRAL ATTACHMENT FIXTURE		MID-EDGE RETAINER		SPECIAL COMPONENTS
		SIZE	QTY	SIZE	QTY	SIZE	QTY	SIZE	QTY	SIZE	QTY		QTY		QTY	
1	SH2Z66	131	38 X 31 X .54 AVG TYPE 316 ST. STL	21	39 X .62 TYPE 316 ST. STL	25		5.0 X 31 X 27 KAO WOOL 1250 FT ² OF 1.0 THK	34				21		63	
2	SH2ZE1	131	38 X 31 X .56 AVG TYPE 316 ST. STL	26	39 X .44 TYPE 316 ST. STL	31		5.0 X 31 X 18 KAO WOOL 1390 FT ² OF 1.0 THK	42				26		78	
3	SH2Z63	131	38 X 31 X .58 AVG TYPE 316 ST. STL	21	39 X .66 TYPE 316 ST. STL	25		5.0 X 31 X 29 KAO WOOL 1480 FT ² OF 1.0 THK	34				21		63	
4	SH2ZE4	131	38 X 31 X .48 AVG TYPE 316 ST. STL	32	39 X .56 TYPE 316 ST. STL	38		5.0 X 31 X 24 KAO WOOL 1395 FT ² OF 1.0 THK	54				32		64	
5		NOT USED														
6		NOT USED														
7	SH2ZD3	132	1.0 X 24 X .29 CARBON/CARBON .50 X 18 X .29 AVG CARBON/CARBON	153 43	36 X .52 AVG MA 956 O.D.S.	168		3.576 FT ² OF 1.0 THK SAFFIL 2.532 FT ² OF 1.0 THK KAO WOOL	392			INTEGRAL WITH COVER PLATE	N/A		NONE	
8	SH2ZC3	C	38 X 38 X .44 38 X VARIOUS TYPE 316 ST. STL	48 66	46 X .52 VARIOUS HASTELLOY X 46 X .52 VARIOUS TYPE 316 ST. STL	48 66		4.920 FT ² OF 1.0 THK SAFFIL 4.920 FT ² OF 1.0 THK KAO WOOL								FUSED SILICA BLOCK: 3.0 THK X 38 X .44, 48 REQD; 3.0 THK X 38 X .44, 48 REQD; 3.5 THK X 38 X .44, 48 REQD; 3.5 THK X VARIOUS, 66 REQD; 02+437-003 PAD, 204 REQD; 02+444-004 REQD; 02+438-001 PLUG, 408 REQD; 02+434-001 RETAINER, 48 REQD; 022+442 RETAINER, 60 REQD; 022+434-001 SHIM, 204 REQD; 022+434-002
9	SH2ZD3	132	RIGHT CIRCULAR CYLINDERS APPROX 1.5 THK X 67 I.D. X 83 O.D. CARBON/CARBON	4				SAFFIL WASHER 6.0 THK X 67 I.D. X 83 O.D. 3.150 FT ² OF 1.0 THK	25							RETAINING RING, 4 REQD .25 THK X 67 I.D. X 83 O.D. INCOLOY 800
10	SH2ZC1	132	RIGHT CIRCULAR CYLINDERS APPROX 1.5 THK X 67 I.D. X 83 O.D. CARBON/CARBON	7 4				SAFFIL WASHER 6.0 THK X 67 I.D. X 83 O.D. 6.048 FT ² OF 1.0 THK	48 4							RETAINING RING, 7 REQD .25 THK X 67 I.D. X 83 O.D. INCOLOY 800
11	SH2ZC7	A	38 X 30 X .60 38 X 30 X .80 CARBON STEEL	35 21	38 X .68 38 X .88 CARBON STEEL	42 25		2.5 X 30 X 30 2.5 X 30 X 15 KAO WOOL 1160 FT ² OF 1.0 THK 1800 FT ² OF .5 THK	56 34				56		147	
12	SH2ZF4	131	38 X 32 X .45 TYPE 316 ST. STL	96	40 X .53 TYPE 316 ST. STL	114		5.0 X 32 X 23 KAO WOOL 4920 FT ² OF 1.0 THK	160				96		192	
13	SH2ZF6	131	38 X 29 X .45 TYPE 316 ST. STL	80	37 X .53 TYPE 316 ST. STL	95		5.0 X 29 X 23 KAO WOOL 17040 FT ² OF 1.0 THK	134				80		160	
14	SH2ZE4	132	RIGHT CIRCULAR CYLINDERS APPROX 1.5 THK X 67 I.D. X 83 O.D. CARBON/CARBON	16				SAFFIL WASHER 6.0 THK X 67 I.D. X 83 O.D. 7.136 FT ² OF 1.0 THK	136							RETAINING RING, 16 REQD .25 THK X 67 I.D. X 83 O.D. INCOLOY 800
15	SH41B2	131	38 X 30 X .38 38 X 32 X .51 38 X 31 X .53 38 X 31 X .51 TYPE 316 ST. STL	85 5 20 16	38 X .46 40 X .59 39 X .41 39 X .59	100 6 24 19		2.5 X 31 X 22 AVG KAO WOOL 2048 FT ² OF 1.0 THK 1024 FT ² OF .5 THK	210				126		278	

- NOTES FOR SH 5 ONLY
1. BLANKET SET CONSISTS OF LAYERS OF INSULATION MATERIAL AND IS NOT IN A COMPRESSED STATE.
2. THE FOLLOWING DETAIL PARTS COMPRISE THE CENTRAL ATTACHMENT FIXTURE:
FOR CLASS A : 022355 RETAINING SLEEVE
022356 POST
022362 RETAINER
022370 WASHER
- FOR CLASS B1 : SAME AS CLASS A EXCEPT MATERIAL IS TYPE 316 STAINLESS STEEL.
3. THE FOLLOWING DETAIL PARTS COMPRISE THE MID-EDGE RETAINER:
FOR CLASS A : 022357 SEAL SLEEVE
022363 NUT
022366 POST
022367 WASHER
- FOR CLASS B1 : SAME AS CLASS A EXCEPT MATERIAL IS TYPE 316 STAINLESS STEEL.
4. ELBOW WEDGES ASSUMED TO BE STRAIGHT FOR SIMPLICITY.
5. DIMENSIONS ARE IN INCHES, SH 5 ONLY.

Fig. 3-5. Thermal barrier general arrangement, Sheet 5

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Fig. 3-6. Hot duct layout, configuration 6

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Fig. 3-7. Lower plenum thermal barrier layout, Sheet 1

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Fig. 3-7. Lower plenum thermal barrier layout, Sheet 2

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3.4.1.1. Duct Velocity. There has been considerable concern in choosing a hot duct design velocity. Because of considerations for vibration, permeation, and material erosion, it has been deemed desirable at this stage to limit the gas velocity to approximately 61 m/s (200 ft/s). However, in view of the desire to minimize the size of the PCRV, as well as the desire for commonality with the HHT plant version, this duct has been designed conceptually for a velocity of 102 m/s (335 ft/s).

3.4.1.2. Materials. The GA HTGR-SC plant used Hastelloy X as the Class B (high temperature) thermal barrier cover plate, attachment fixture, seal sheet, and hot duct thermal shield material. Hastelloy X is not considered a viable candidate in the regime of predicted HTGR-GT core outlet temperatures (due to carburization and creep). It is also considered premature to state that with cast superalloys (such as IN 713LC or IN 100) carburization will not be a problem. Equally important is the lack of adequate friction-reducing coatings for application above 870°C (1600°F). There is also a question regarding the availability of an adequate seal sheet material. At the reference temperatures most nickel-based alloys lose chromium by depletion, which is frequently accompanied by loss of ductility. Since flexure of the seal sheet between cover plates is a design condition, this loss of ductility could be critical. Hence, there is a general concern about the possible employment of metallics in the Class B thermal barrier.

Although clearly lacking in design data, the use of nonmetallic materials such as a ceramic or a carbon/carbon (C/C) composite for the Class B insulation system and the core outlet duct is favored. One initial preference is for a ceramic material such as a slip-cast fused silica or slip-cast fused alumina (or a composite of the two). Ceramics are preferred to C/C composites because of a more extensive material data base and greater confidence in the data, as well as the manufacturing state of the art. However, by the very nature of the ceramic materials, a thicker thermal barrier, and thus a larger duct outside diameter, would be required. Also, because the system unit weight would be appreciably greater than for the fibrous insulation/cover plate type, a relatively rigid foundation would be necessary. This could lead to a liner duct outside diameter of perhaps as much as 3 m

(10 ft) in the case of a single hot duct. This is currently impractical with lower plenum and turbine inlet geometries. Current opinion is that the use of ceramics is not compatible with an annular counterflow duct configuration. Thus, for design studies at this state, a C/C structure will be referenced.

3.4.1.3. Thermal and Seismic Movements. Particular concerns with the demonstration plant warm liner concept are related to the thermal and seismic movements of the thermal shield in the core cavity. Seismic movements of approximately 51 mm (2 in.) are predicted in the vertical and radial directions. Thermal growths of the thermal shield at the duct location are approximately 19 mm (0.75 in.) in both the radial and vertical directions. These movements require complex flexible connections at the interfaces with the hot ducts.

3.4.2. Demonstration Plant Design

Preliminary design studies were completed for hot ducts having velocities of 61, 102, and 122 m/s (200, 335, and 400 ft/s). The principal concept is an offset annular duct designed for an operating velocity of 102 m/s (335 ft/s). The basis for this concept is that high-temperature, high-resiliency insulation can be made in segments to form a washer fitted between a C/C inner shield and a conventional steel liner. For ease of handling, the insulation would be at least partially covered with a high-temperature cloth such as quartz or woven insulation fibers, both of which are commercially available. The washer segments would be installed in approximately 1 m (3 ft) units and compressed to a predetermined amount by retaining rings. The rings would be attached to the liner by conventional bolts. The assembly of the unit would be completed by inserting and attaching a C/C cylinder to the ring. The next set of blanket washers would be placed into position and the process repeated progressively. Note that in this concept the insulation primary fiber orientation is in the radial direction. While this increases the solid fiber conduction by up to 50%, it also produces acceptable stiffness against the C/C cylinder while allowing compression without fiber damage during installation. This design has the advantage of allowing the fasteners to be installed after the thermal barrier is set up and the blankets are compressed. Component removal and replacement can be done with relative ease.

Critical design issues for the thermal barrier have been identified and are summarized as follows:

1. A C/C composite material has been identified for use as the cover plate material in high-temperature regions such as the lower cross ducts and the lower plenum. No data are available on the use of C/C materials in HTGRs. In particular, expected high sound power levels (up to about 170 db) and high duct velocities (up to about 100 m/s) could cause problems with this material.
2. Due to the elevated temperatures in the lower plenum, it is expected that hard ceramic materials will be required in the thermal barrier.
3. Removeability requirements for the hot duct will require development of flange coupling mechanisms and removal guide systems.
4. Thermal expansion and seismic movement of the core barrel require development of a reliable flexible seal at the duct to core barrel interface.
5. The rapid depressurization rates expected in the HTGR-GT will require design changes to the standard thermal barrier metallic-fibrous designs.

4. DEMONSTRATION PLANT SYSTEM ENGINEERING

4.1. STEADY-STATE CODES AND PERFORMANCE

The development of the CODER5 version of CODER (cost/performance code) has progressed to the final debugging stage. This version includes the intercooled cycle in a base case thermodynamic calculation for the demonstration plant HTGR-GT with either the conventional or warm liner. The program can now operate in a base case - alternate case mode whereby the reference design plant is defined for the program by stored input data and an alternate plant is defined by adjusting independent design variables. The intended purpose of CODER is to model the sizing, performance, and cost of the HTGR-GT plant as a function of the selected design variables which describe the plant design. Presently CODER can calculate plant performance but it will not have the capability to size the plant or optimize system costs until additional modifications are made to the PCRVR and DUCT subroutines and cost data and algorithms are updated. In preparation for updating the PCRVR and cost algorithm subroutines, preliminary programming changes were incorporated into CODER5. Further work in this area is not scheduled due to budgetary limitations until the second quarter of FY-79.

Performance calculations have been made to determine the demonstration plant output for variations in turbomachine efficiencies and ambient conditions. Table 4-1 compares the plant parameters as reported in Ref. 4-1 with those for a plant with updated turbomachine efficiencies on both a standard day and a cold [9°C (49°F)] day. For the study reported in Ref. 4-1, the isentropic efficiencies for the turbine and for the low- and high-pressure compressors were 91.8% and 89.8%, respectively. For the updated study the efficiencies for the turbine, low-pressure compressor, and high-pressure compressor were 92.2%, 90.2%, and 90.8%, respectively.

TABLE 4-1
COMPARATIVE PLANT PARAMETERS FOR VARIATIONS IN TURBOMACHINE
EFFICIENCIES AND AMBIENT CONDITIONS

	Reference 4-1 Study Standard Day	Updated Turbomachine Efficiencies	
		Standard Day	Cold Day
Reactor rating, MW(t)	1530	1530	1530
Number of loops	1	1	1
Liner option	Warm	Warm	Warm
Compressor pressure ratio (each compressor)	1.75	1.75	1.75
Compressor pressure ratio (overall)	3.0	3.0	3.0
Compressor flow, kg/h (lbm/h)	2,742,464 (6,046,100)	2,731,941 (6,022,900)	2,726,634 (6,011,200)
High-pressure compressor outlet pressure, kPa (psia)	7868.4 (1142)	7868.4 (1142)	7868.4 (1142)
Reactor inlet temperature, °C (°F)	445 (833.9)	444 (831.1)	443 (829.7)
Reactor outlet temperature, °C (°F)	850 (1562.2)	850 (1562.2)	850 (1562.2)
Turbine inlet temperature, °C (°F)	849 (1560.0)	849 (1560.0)	849 (1560.0)
Low-pressure compressor inlet temperature, °C (°F)	28 (82.6)	28 (82.7)	23 (72.6)
High pressure compressor inlet temperature, °C (°F)	27 (80.2)	27 (80.1)	21 (70.1)
Minimum cycle helium temperature, °C (°F)	26 (79.0)	26 (79.0)	21 (69.0)
Primary system pressure loss, kPa (psi)	459.6 (66.7)	458.2 (66.5)	460.9 (66.9)
Recuperator high-pressure ΔP , kPa (psi)	44.8 (6.5)	44.1 (6.4)	44.1 (6.4)
Core ΔP , kPa (psi)	69.6 (10.1)	69.6 (10.1)	69.6 (10.1)
Turbine ΔP , kPa (psi)	4878.8 (708.1)	4880.2 (708.3)	4878.1 (708.0)
Recuperator low pressure ΔP , kPa (psi)	37.2 (5.4)	36.5 (5.3)	36.5 (5.3)
Precooler ΔP , kPa (psi)	28.9 (4.2)	28.9 (4.2)	30.3 (4.4)
Intercooler ΔP , kPa (psi)	22.7 (3.3)	22.7 (3.3)	24.1 (3.5)
Low-pressure compressor pressure rise, kPa (psi)	1966.4 (285.4)	1966.4 (285.4)	1966.4 (285.4)
High-pressure compressor pressure rise, kPa (psi)	3372 (489.4)	3372 (489.4)	3372 (489.4)
Recuperator effectiveness	0.898	0.898	0.898
Turbine isentropic efficiency, %	91.8	92.2	92.2
High-pressure compressor isen- tropic efficiency, %	89.8	90.2	90.2
Low pressure compressor isen- tropic efficiency, %	89.8	90.8	90.8
Generator efficiency, %	98.8	98.8	98.8
Turbine disk cooling flow, %	3.8	3.8	3.8
Ambient air temperature, °C (°F)	15 (59.0)	15 (59.0)	9 (49.0)
Cooling water temperature, °C (°F)	21 (69.0)	21 (69.0)	15 (59.0)
Precooler outlet water tempera- ture, °C (°F)	87 (188.5)	87 (188.5)	87 (188.5)
Intercooler outlet water tem- perature, °C (°F)	66 (150.0)	66 (150.0)	66 (150.0)
CACS parasitic heat loss, MW(t)	1.8	1.8	1.8
Heat losses, MW(t)	7.0	7.0	7.0
Auxiliary power, MW(t)	8.0	8.0	8.0
Plant efficiency, %	40.6	41.33	41.96
Net electrical output, MW(e)	621.0	632.4	642.0

4.2. TRANSIENT CODE AND DYNAMICS STUDIES

The REALY2 transient computer code (Ref. 4-2) has been updated using specifications for the demonstration plant design. Specifically, new models for the core, turbomachine, and heat exchangers have been developed. The code was also revised to allow heat transfer modeling in the warm liner annuli. The plotting capability has also been improved. Plots can now be obtained on the Tektronix plotting system, which provides fast off-line plotting capability to program users. The warm liner heat transfer can be modeled by invoking the pseudo-plenum mode of the DUCT subroutine. The overall REALY2 model matches the PCRV geometry and the fluid conditions of the demonstration plant design as well.

The core model updates were the most recent revisions to the code. The neutron kinetics are changed only slightly; parameters describing core thermal properties are changed to a greater degree primarily because the model scaling has changed. The dynamics of the core are now representative of both the demonstration plant design and the alternate two-loop intercooled commercial plant. It appears reasonable to use the new model in forthcoming studies of plant control and dynamics.

The core geometry in REALY2 is referred to as fuel-centered modeling. The core model geometry is shown in Fig. 4-1 (from Ref. 4-2). The model dimension r_f shown in this figure is equal to the fuel rod radius in the demonstration plant design.

The model was previously based on the core design for a 3000-MW(t) plant. Selected differences between the past and present models are shown in Table 4-2. The model scaling as shown in Table 4-2 is based on the number of fuel rods as viewed in the actual core cross section.

Core components which are modeled in REALY2 are shown in Fig. 4-2. The new core data revise volumes, thermal resistances, surface areas, power fractions, and flow fractions consistent with the new core model. The core and

Fig. 4-1. Fuel-centered core geometry

TABLE 4-2
COMPARISON OF SELECTED CORE MODELS

	Old Core Model Based On 3000-MW(t) Design	New Core Model Based on Demonstration Plant Design [scaled to 3000-MW(t) for comparison]
Power density, W/cm ³	8.4	7
Fuel-centered model scaling	1:60988	1:145738
Model reference core flow rate at design power, kg/s (lb/h)	1.82×10^3 (14.45×10^6)	1.42×10^3 (11.27×10^6)
Core helium inlet temperature, °C (°F)	511 (951)	444 (831)
Core helium exit temperature, °C (°F)	827 (1521)	850 (1562)
Core helium inlet pressure, kPa (psia)	6757 (980)	7750 (1124)

Fig. 4-2. Active core model showing nodes and coolant flow

side reflector are divided for more computational detail, as in the previous modeling. The neutron kinetics are a point-source model which is described by an average mean lifetime and six delayed neutron groups with lambdas and betas for an MEU design.

In order to model the warm liner and plant dynamics more accurately, the helium circuit volumes for the warm liner demonstration plant were calculated and input to the REALY2 code. The thermal masses and heat transfer data of components associated with warm liner flows were also derived and input to the REALY2 code. When the code is used to compute transients, especially those involving cooling failures, heat storage in the warm liner annuli and the propagation of temperature changes through the warm liner regions of the helium circuit are now taken into account.

The modeling work has also included an improvement to handle the condition known as "compressor turbinizing." During rapid shutdown the low-pressure compressor picks up all of the load, leading to a condition equivalent to "turbinizing" of the high-pressure compressor. The MCURVE and COMPR routines in REALY2 have been revised to account for this condition in plants such as the demonstration plant which use two-stage compression. The compressors are described by performance maps in the code. The allowable search range for compressor parameters which are described by the tabular data in the code was expanded beyond the previous versions of the code. When the code searches for parameters corresponding to compressor pressure ratios which are below or only slightly above 1.0 (i.e., "compressor turbinizing"), the new extrapolated data allow the search to continue. When the code computes compressor energy balances which are actually based on the turbinizing condition, the turbinizing condition is recognized and the computation is made accordingly.

The code modifications have been tentatively based on recommendations by United Technologies to allow transition into the turbinizing region. The results, while acceptable, show the transition into and out of turbinizing to be abrupt. It is expected that refinement of the turbomachine tables by United Technologies can eventually alleviate this problem.

The model has been used to obtain a set of plotted transients for observing the extreme conditions of plant operation for a selected set of critical transient events. The set of selected events for a single-loop HTGR-GT plant are listed in Table 4-3, which also summarizes some of the major results. These transients are a basis for establishing controls, component, and plant protection system (PPS) requirements. (The licensing categories shown in Table 4-3 are as used formerly and are not the new Regulatory Guide 1.70 guide lines.)

Figure 4-3 was extracted from the plot package describing "total loss of cooling water." The figure shows many of the plant temperatures for one of the worst possible conditions of high temperature and pressure because both the precooler and intercooler are subjected to no-flow conditions at relatively high helium pressure. A total loss of cooling water affecting both the precooler and intercooler, although unlikely in the single-loop demonstration plant, would be more likely in the demonstration plant than in multiloop designs where separation of portions of the cooling water system may be possible. Further, it should be noted that the event analyzed assumes a concurrent failure of the non-PPS protective function to detect the loss of flow and/or to initiate shutdown. Requirements for the PPS set point for the core inlet temperature are established from this transient.

Figure 4-3 shows the transient from a point just prior to the cooling water flow being stopped. The cooling water flow is stopped in about 8 s. There is a tendency for the liner material and shrouding in the warm liner design to delay the rate of temperature rise as compared with previous plants analyzed for this transient. This is because the gas which leaves the precoolers, although rising in temperature, must pass through several warm liner annuli and transfers heat to the cooler metal in the process. Due to this effect, the core inlet temperature takes 10% to 15% longer than in conventional liner plants to reach the core inlet temperature set point of 510°C (950°F). In the process the liner is heated to nearly 204°C (400°F). The reactor and the turbomachine are tripped to shut down the plant when the core inlet temperature reaches the set point.

TABLE 4-3
SUMMARY OF WARM LINER MAXIMUM PLANT
PARAMETERS FOR TRANSIENT EVENTS AT 100% POWER

Event	Reactor Power (%)	Turbomachine Speed (%)	Side Cavity Pressure [kPa (psia)]	Liner Temperature, Maximum [°C (°F)]	Event Class
Total loss of cooling water	Nominal	Nominal	~6308 (~915) ^(a)	~199 (~390) ^(a)	Emergency
Rod runout	140	Nominal	Nominal	Nominal	Emergency
Bypass valve fails open	Nominal	Nominal	~5515 (~800)	Nominal	Upset
Emergency shutdown from design	Nominal	108	~820	29 (85)	Emergency
Generator shaft breaks	Nominal	117	~5584 (~810)	Nominal	Faulted
Loss of precooler water	103	Nominal	Nominal	~177 (~350)	Emergency
Plant loss of load	Nominal	108	~5377 (~780)	Nominal	Upset

(a) The transient values are short-term effects which do not include CACS cooling. Temperatures and pressures will decrease when the CACS is started.

Fig. 4-3. Total loss of cooling water, low-temperature transients (single loop - intercooled)

Figure 4-4 was extracted from the plot package describing "plant loss of load." The figure shows control system response during the most active portion of this transient. There is more volume associated with the helium circuit in the warm liner concept because the plant has considerable volume (>30%) in the warm liner annuli. Correspondingly, by comparison with the conventional liner demonstration plant, the warm liner model shows a difference in the response of system pressures and flows to the loss of load disturbance. There is a tendency for the pressure rise in the warm liner model to occur more slowly and to oscillate at higher amplitude. The controls response is also more oscillatory. These observations have been considered a subtlety and, other than the valve size, not a basis for revision of plant controls as modeled in REALY2. However, the controller compensation designs should be refined to reflect these differences and much of the oscillatory characteristic should then be eliminated.

The valve size that was previously selected for the conventional liner demonstration model was 0.79 m (2.6 ft); however, the valve required to hold the overspeed level below 110% in the warm liner demonstration plant was 0.85 m (2.8 ft), and this size valve was used in all demonstration plant transients. The maximum speed of 108% occurs in about 1.3 s as shown in Fig. 4-4 ("turbine transients").

The studies have helped to show areas of interest in developing the heat exchangers, the liner design, and the turbomachine. The major area where further model development is required is the turbomachine because of the compressor turbing effect. Further improvement of component representation, such as the heat exchanger models, will depend upon advancement of the design. Also, in order to continue the demonstration plant effort of completing control/protective system concepts, the controls will require additional development.

Fig. 4-4. Plant loss of load, turbine transients (single loop - intercooled)

REFERENCES

- 4-1. "Gas Turbine HTGR Program, Quarterly Progress Report for the Period Ending March 31, 1978," DOE Report GA-A15002, General Atomic Company, June 1979.
- 4-2. Croft, M. Z., "REALY2: The GT-HTGR Transient Performance Analysis Program," General Atomic Report GA-A13880, March 31, 1976.

5. DEMONSTRATION PLANT REACTOR CORE ENGINEERING

5.1. GENERAL

Preliminary physics design studies were completed for the 1530-MW(t) demonstration plant HTGR-GT core. This core study was the first to investigate in detail the behavior of MEU fuel. The study provided significant information relating to the use of MEU fuel, which can be summarized as follows:

1. U-238, which is the dominant (denaturing) nuclide in the fissile fuel, has a large resonance cross section in the resonance energy range. This has a significant impact on the core physics design and fuel cycle choice.
2. Large variations in the thorium loading are required in the buffer zone for controlling the power peaking and intraregion tilts due to the dominant effect of U-238.
3. In comparison with the HEU/Th fuel cycle, region power peaking and tilts are very sensitive to a small shift in the MEU/Th fuel distribution.

The results of the study show that the core operates at relatively low temperatures at steady-state full power. The peak fuel rod centerline temperature, which is the maximum temperature the core will experience at any time point during its life, is 1205°C (2200°F). The overall thermal behavior of the core is good. The time histories of fast neutron flux, fissions in failed fuel particles, and core-averaged gaseous fission product release as characterized by Kr-85m and Xe-138 isotope releases are well within limits established for the HTGR-SC. Significant core performance results are summarized in Table 5-1.

TABLE 5-1
CORE PERFORMANCE SUMMARY FOR 1530-MW(t) DEMONSTRATION PLANT HTGR-GT^(a)

Peak fuel centerline temperature, °C (°F)	1205 (2200)
Time-averaged fuel centerline temperature during 3-yr core residency (maximum), °C (°F)	1149 (2100)
Peak moderator (H-451 graphite) temperature, °C (°F)	1149 (2100)
Time-averaged moderator temperature during 3-yr fuel core residency (maximum), °C (°F)	1094 (2000)
Peak fast neutron fluence, n/m ²	5×10^{25}
Fast neutron fluence design limit, n/m ²	8×10^{25}
Peak burnup (FIMA fraction)	
TRISO UC ₂	0.25
TRISO ThO ₂	0.035
Total failed fuel particle fraction (core averaged), %	
TRISO UC ₂	0.14
TRISO ThO ₂	0.07
Fraction of fissions in failed particles (total, core averaged)	
Computed value	0.00124
Level A ^(b)	0.0025
Kr-85m R/B ratio	
80% confidence level	3.11×10^{-5}
Level A ^(b)	6.16×10^{-5}
Xe-138 R/B ratio	
80% confidence level	2.618×10^{-6}
Level A ^(b)	4.57×10^{-6}

(a) $T_{in} = 453^{\circ}\text{C}$ (848°F); $T_{out} = 850^{\circ}\text{C}$ (1562°F).

(b) Level A values refer to the design criteria values based on safety considerations.

The demonstration plant HTGR-GT core has excellent design features. The core has 10-row FSV-type fuel elements, a low power density of 6.5 W/cm^3 , a 3-yr fuel cycle with all TRISO fuel particles, an optimized axial power profile, and excellent fuel zoning. On a core-averaged basis, no fuel particle failure occurred due to kernel migration or SiC - fission product reaction.

While a review of results projects credible performance of the demonstration plant HTGR-GT core, a preliminary estimate of the metallic fission product release from the core shows a cesium release of $\sim 37,000 \text{ Ci}$ on a 40-yr plateout basis. Two following factors primarily contribute to this relatively high prediction:

1. Postulated pressure vessel failure mechanism for the MEU fuel.
2. High core thermal-power-to-coolant-flow ratio.

5.2. METHODS AND ASSUMPTIONS

The demonstration plant HTGR-GT core layout with one-third symmetry and fuel segment identification is shown in Figs. 5-1 and 5-2. Major core design parameters are shown in Tables 5-2 and 5-3. The refueling scheme is shown in Table 5-4. The reactor coolant enters the core at 453°C (848°F) and exits at 850°C (1562°F). The prismatic fuel element employed in the core is identical to the 10-row FSV element. A standard and a control element are shown in Figs. 5-3 and 5-4.

The demonstration plant HTGR-GT core uses medium-enriched-TRISO UC_2 as the fissile fuel particle design data in Ref. 5-1 were based on an estimated peak burnup of 16.5% FIMA with an end-of-cycle (EOC) reactivity of 1.015. Subsequently, the EOC reactivity was reduced to 1.008 for the purpose of controlling the intraregion power peaking in the core. This reduction in the EOC reactivity resulted in a decrease of approximately

NOTES:

1. FUEL REGION BOUNDARIES
2. CONTROL ROD COLUMN
(WITH FUEL REGION
IDENTIFICATION NUMBER)
3. FIXED-ORIFICE REGION

Fig. 5-1. Plan view of HTGR-GT demonstration plant

Fig. 5-2. Layout of symmetric one-third of HTGR-GT demonstration plant core

Fig. 5-3. Fuel element

Fig. 5-4. Control fuel elements

TABLE 5-2
CORE DESIGN AND PERFORMANCE FEATURES FOR THE HTGR-GT

Description	Conditions
Power density	Nominal 7 W/cc
Core layout	See Fig. 5-1
Fuel cycle	Design capable of changing between MEU and HEU
Fuel lifetime	3 year
Fuel element	10 row
Coated particle type	TRISO/TRISO
Region peaking factors (with tilts)	Choose same envelope as SC-HTGR
Core helium temperatures In Out	453°C (intercooled DC-HTGR) 850°C
Core ΔP Demonstration HTGR-GT	8.5 psi
Fuel contamination factor	Lower by factor of 5 to 2×10^{-5}
Manufacturing defect fraction	Lower by factor of 2 to 1×10^{-3}
Fission product basis for maintenance of HTGR-GT	80% confidence on gases 50% confidence on metals (Level A)
Fission product release model from fuel particle	Model the diffusive release of Cs and Sr

TABLE 5-3
BASIC CORE PARAMETERS OF HTGR-GT

Core thermal power (MW)	1520
Power density (kW/l)	6.5
Fuel block design	FSV - 10 row
Fuel in-core lifetime (yr)	3
Reload interval (yr)	1
Fraction of core reloaded annually (%)	~33
Capacity factor (%)	80
Carbon/thorium initial core/reloads	350/580
Core volume (m ³)	234.6
Number of fuel columns	331
Standard	282
Control	49
Number of fuel blocks (8/col.)	2648
Number of control rod pairs	49
Number of small control rods	49
Number of reserve shutdown hoppers	49
Number of flow regions (total)	55
Variable flow control	
7-column	37
5-column	12
Fixed orifice regions	6
Number of axial fuel zones	4
T _{inlet} (°C)	453
T _{outlet} (°C)	850
Fuel rod diameter (cm)	1.22

TABLE 5-4
SEGMENT AND FUEL LOAD IDENTIFICATION CHART FOR SURVEY ANALYSIS

Fuel Load	Year of Operation	Segment Positions		
		A	B	C
Initial core	1	Fuel load 1	Fuel load 1	Fuel load 1
Reload 1	2	Fuel load 2	Fuel load 1	Fuel load 1
Reload 2	3	Fuel load 2	Fuel load 2	Fuel load 1
Reload 3	4	Fuel load 2	Fuel load 2	Fuel load 2
Reload 4	5	Fuel load 3	Fuel load 2	Fuel load 2
Reload 5	6	Fuel load 3	Fuel load 3	Fuel load 2
Reload 6	7	Fuel load 3	Fuel load 3	Fuel load 3

5% in the fuel loadings and an increase in peak burnup to ~20% FIMA. For the sake of streamlining the fuel fabrication technique, it is recommended that all MEU cores use the fuel particles designed for the FSV core with a peak burnup of 26.5% FIMA. The design data of the FSV fuel particles are shown in Table 5-5. The FSV fuel particles were used in the demonstration plant HTGR-GT core since neither the fuel rod void fraction requirements nor the fuel loading increases were seriously compromised.

The methods and computer codes used in the demonstration plant HTGR-GT core performance study are described in Ref. 5-2. The interfacing physics code TSORT generates the three-dimensional power and flux distributions in the core, which include both initial and equilibrium core conditions covering up to 7 yr of core operation. Using the TSORT output data, SURVEY-THERM (Part 1 of the SURVEY code) establishes the fuel, graphite, and coolant temperatures as a function of time at the bottom layer of each axial fuel zone. Temperatures are established at every local point in each column along the axial layer. These temperature distributions are subsequently used in calculating kernel migration and SiC - fission product attack penetration distances in the fuel particle coatings.

SURVEY-PERFOR (Part 2 of the SURVEY code) computes the kernel migration distances, fuel particle failure fractions, fissions in failed particles, and gaseous fission product releases.

Four fuel particle failure mechanisms are considered in this study:

1. Kernel migration.
2. SiC - fission product attack at both high and low temperatures.
3. Pressure vessel failure.
4. Manufacturing defect failure.

The first two failure mechanisms are strongly dependent on fuel temperature history. A discussion on the failure mechanisms and probabilistic evaluation of failure fractions is given in Ref. 5-3. Revised failure models

TABLE 5-5
DESIGN DATA FOR FUEL PARTICLES SELECTED FOR THE
DEMONSTRATION PLANT HTGR-GT CORE

Property	Average Dimension (μm)	Standard Deviation (%)
FSV TRISO MEU UC ₂ ^(a)		
Kernel diameter	350	5.2
Coating thickness		
Buffer	115	17.7
IPyC	35	16.3
SiC	35	14.3
OPyC	40	13.8
FSV TRISO ThO ₂		
Kernel diameter	450	2.8
Coating thickness		
Buffer	70	18.0
IPyC	35	16.3
SiC	35	11.3
OPyC	40	14.0

(a) Design proposed for fuel experiencing a maximum kernel burnup of 26.5% FIMA.

were generated for the all-TRISO MEU fissile and fertile fuel particles. They are based on HEU models under equivalent conditions. Notable changes in the failure models are as follows. A fuel particle is postulated to have failed by kernel migration if the kernel has penetrated through the buffer and the inner PyC coating. In the earlier model, failure was postulated when the kernel had completely penetrated through the buffer layer. In the SiC - fission product attack model, fuel particles are postulated to have failed when the broad front is halfway advanced through the SiC coating; however, only the metallic fission products are expected to be released and the gaseous fission products are considered to be retained by the outer PyC coating. The pressure vessel failure in the fuel particles is assumed instantaneous because of expected high coating stresses.

The study also included graphite burnoff fractions and the fuel hydrolysis rates for use in estimating the impact on the fission product release rates, the stresses in the fuel element blocks, and the circulating activity. The gaseous fission product releases are characterized by the releases of the isotopes Kr-85m and Xe-138. SURVEY-PERFOR computes the release-to-birth (R/B) ratio of the gaseous fission products as a function of time. All results produced by the SURVEY code are used as input to the TRAFIC code, which calculates the metallic fission product release. Estimates of metallic fission product release from the demonstration plant HTGR-GT core are under way and will be documented in FY-79.

REFERENCES

- 5-1. "Gas Turbine HTGR Program, Quarterly Progress Report for the Period Ending March 31, 1978," DOE Report GA-A15002, General Atomic Report, June 1979.
- 5-2. Shenoy, A. S., and D. W. McEachern, "HTGR Core Thermal Design Methods and Analysis," General Atomic Report GA-A12985 (GA-LTR-17), December 31, 1974.
- 5-3. Smith, C. L., "Fuel Particle Behavior Under Normal and Transient Conditions," USAEC Report GA-A12971 (GA-LTR-15), General Atomic Company, October 1, 1974.

6. DEMONSTRATION PLANT HEAT EXCHANGER STUDIES

6.1. INTRODUCTION

Heat exchanger activities during this report period focused upon the conceptual design of the recuperator, precooler, and intercooler for the GA version of the single-loop demonstration plant. This work is being performed by Combustion Engineering under subcontract to GA and covers the heat exchanger design work downstream of thermal sizing, which was performed by GA. Items to be delivered by Combustion Engineering include the following:

1. General arrangement drawings, including major design features and mechanical interface definition.
2. Materials lists.
3. Manufacturing sequence drawings.
4. Estimated costs.
5. Design description documents.
6. Final report covering all of the above.

To date Combustion Engineering has delivered items 1 and 5 on schedule; the remaining items are due at the end of the first quarter of FY-79. The main features of the Combustion Engineering designs are discussed in the following sections.

6.2. RECUPERATOR STUDIES

As a result of GA parametric studies (Ref. 6-1), a straight-tube counterflow arrangement of heat transfer surface was selected for the recuperator. Table 6-1 presents the relevant statistics for the selected design.

The large number of tubes and corresponding large diameter requirements led to a modular design approach in which the recuperator is built up a number of close-packed hexagonal tube bundles, as shown in Fig. 6-1. High-pressure helium from the high-pressure compressor enters the recuperator near the top, flows downward inside the tubes, turns 180°, flows upward through a central pipe (integral return tube) provided in each module, and exits at the top. Low-pressure turbine-discharge helium enters the recuperator cavity at the bottom, flows upward outside the tubes transferring its heat countercurrently with the tube-side helium, and exits at the top.

The conceptual arrangement of the recuperator shown in Fig. 6-1 calls for 161 modules, containing 588 tubes each, to be supported from a hemispherical head. The head is bolted to a support flange which in turn is bolted to a support skirt on the PCRV liner. The hexagonally shaped modules are inserted into a honeycomb-like structure, which acts as a flow baffle at the periphery of the modules and provides one link in the load path for transmission of lateral loads from modules to shroud to cavity wall. The module baffle/support structure is built up from three-legged elements which are pinned together to form the honeycomb-like cross section. Also, seismic support frames are pinned at the periphery of the structure and are subsequently keyed to the shroud. The outer shroud is then keyed to the PCRV liner at the lower end.

All of the recuperator interfaces with the PCRV are mechanical (no welded attachments). Two structural interfaces are provided: (1) a bolted, keyed, and mechanically sealed shroud support flange at the upper end, and (2) a key-type lateral seismic support at the lower end. In addition, attachment points for PCRV ducting are provided.

TABLE 6-1
RECUPERATOR THERMAL SIZE DATA

Configuration	Straight-tube counterflow
Tube outside diameter by wall, mm (in.)	11.1 x 0.81 (0.4375 x 0.032)
Tube pitch/diameter	1.455 triangular
Tube material	2-1/4Cr - 1Mo (ASME SA 213-T22)
Number of tubes	94,677
Effective tube length, m (ft)	12.8 (42)

Fig. 6-1. Recuperator for 600-MW(e) one-loop intercooled plant

The conceptual design of the recuperator is amenable to a number of fabrication approaches, ranging from complete shop assembly to final assembly in the PCRV cavity. Shipping and/or site limitations will undoubtedly dictate the appropriate combination of shop versus field assembly. The recuperator is designed such that modules can be inserted in the support structure with each module hung via a mechanical attachment to the hemispherical head.

The impact of the warm liner approach on this design is considered to be minor because it entails mechanical design accommodations for the duct interfaces at the low-pressure helium inlet to and exit from the recuperator. In a more conventional cold liner recuperator installation, there would be no need for mechanical connections in these areas.

6.3. PRECOOLER STUDIES

The precooler design, as sized by GA, is a helically coiled finned tube gas-to-water counterflow heat exchanger with pressurized water within the tube circuit (Figure 6-2). The precooler module is approximately 4.88 m (16 ft) in diameter and 23.8 m (78 ft) in length. The helical bundle is approximately 13.7 m (45 ft) of this length. The helical bundle consists of 1200 finned tubes. The water inlet is at the bottom of the module and the exit is at the top. The gas inlet is from the top of the module to the bottom, where it turns 180° and flows up around the outside of the module and out around the gas inlet pipe. This exit gas is used to cool the PCRV liner in several of the cavities. The water flows in two separate circuits within the helical bundle, each of which can be independently shut down. This provides the ability to continue operation of a multiloop reactor plant with one loop down in which half of the precooler is dumped, as would be the case for a precooler segment dump. This is required to limit temperatures in the downed loop. The precooler water boxes and their respective feed lines are arranged to permit lateral piping of the heat exchanger to a pipe chase cavity in the PCRV, where access for maintenance and inservice inspection operations is more readily available.

Fig. 6-2. Intercooler for 600-MW(e) one-loop intercooled plant

The precooler is divided into four major subassemblies: (1) helically coiled tube bundle assembly, (2) upper module support assembly, (3) lower module seismic support assembly, and (4) inlet/outlet tubesheet assembly. Key features of these subassemblies are discussed in the following sections. Even though the precooler design conceptually divides the unit into four separate subassemblies, some of these subassemblies cannot be completed independently of the others. The inlet/outlet tubesheet subassemblies can be built only partially before they are ready to be attached to the tube bundle. Due to the close spacing of the tube array, special installation procedures are required to attach a tubesheet assembly to the tube bundle assembly. Basically, this means attaching the tubes from the inner rows out so that welding and inspection can be performed with a minimum of adjacent tube interference. The inlet/outlet water pipes are not welded to the nozzle forging until the precooler has been installed in its PCRV cavity, at which time the liner transition/pipe is installed through the PCRV pipe chase and attached to the liner and nozzle forging. Thus, it can be seen that the inlet/outlet tubesheet assemblies, as discussed below, are not completed until final precooler installation in the PCRV cavity. The other three subassemblies can be completed independently.

6.3.1. Helically Coiled Tube Bundle Assembly

The tube bundle consists of 1200 helically coiled finned tubes (see Table 6-2) supported by six full-size radial support plates and six partial support plates. The full-size support plates are attached to an inner shroud, and the tubes are then installed in their appropriate positions and locked into place. An outer shroud is then assembled onto the bundle and welded to the full-size support plates. The outside diameter of the inner shroud is insulated prior to tube installation. The outer shroud outside diameter is insulated after assembly to the support plates. (A closure plate is attached to the top of the inner shroud to prevent active gas flow within the inner shroud.) The tube bundle assembly is approximately 4.88 m (16 ft) in diameter and 13.7 m (45 ft) in length.

TABLE 6-2
PRECOOLER HELICALLY COILED FINNED TUBE BUNDLE

Tube outside diameter, ^(a) mm (in.)	31.8 (1.250)
Tube average wall thickness, mm (in.)	2.87 (0.113)
Fin height, mm (in.)	1.60 (0.063)
Fin thickness, mm (in.)	0.51 (0.020)
Number of fins/mm (fins/in.)	0.63 (16)
Average tube length, m (ft)	82.9 (272)
Number of tubes	1200
Bundle inside diameter, m (ft)	1.28 (4.2)
Bundle outside diameter, m (ft)	4.8 (15.75)
Number of rows	34

(a) As measured across the fins.

The tubes are supported by six full-size support plates attached to and extending radially from the inner shroud. In order to minimize the unsupported tube span, six partial radial support plates are provided for the outer tube rows. The full-size support plates are attached to both the inner and outer shrouds, while the partial support plates are not attached to either shroud. The support plates are drilled in a helical pattern such that when attached to the inner shroud each tube can be threaded into position. Each tube support location has the hole drilled at the same angle. However, due to the different number of tube turns in each row, the hole pattern is irregular. (In order to have approximately the same length in each tube in the bundle, the number of tube turns decreases as coil diameter increases.) Each tube hole is slightly larger than the tube fin outside diameter in order to facilitate tube installation. Therefore, in order to hold the tube into position after installation a tube locking device is provided at each support location. The locking device is basically an internally threaded tube, flared on the outside of one end and split longitudinally. The device is installed around the outside of the tube and threaded into the hole until the flared end seats on one side of the support plate and the other side is crimped into the tube fins. This device secures the tube from both movement within and through the support location. After all tubes are installed and locked into place, the outer shroud is installed.

6.3.2. Upper Module Support Assembly

The precoolers main support is through a flange bolted to a ledge ring in the PCR liner. The flange and bundle outer shroud are attached by an extension ring (upper shroud) of the same diameter and thickness as the bundle outer shroud. A closure plate is bolted to the top of the flange to provide a gas bypass seal and route the gas flow downward. The inlet gas enters the module through a cross duct, which is bolted to a flange welded into the side of the upper shroud. The exit gas flows around the outside of the cross duct. Insulation extends from the bundle shroud to the upper flange to prevent unacceptable regenerative heat leak to the liner

cooling gas. The upper flange has 12 integral keys, which insert into corresponding keyways in the PCRV liner. These keys locate the precooler module position and also provide upper module seismic support.

6.3.3. Lower Module Seismic Support Assembly

Seismic support is provided at the bottom of the module. The support is basically a large-diameter flange with 12 keys that are inserted into keyways machined into the PCRV liner at the bottom of the cavity. A lower shroud extends from the bundle outer shroud and is attached to the lower flange. The lower shroud is the same diameter and thickness as the bundle shroud. However, the lower shroud is perforated to allow the cooled gas to turn 180° and exit upwards between the outer shroud and the PCRV cavity liners; therefore, the lower portion of the shroud does not require insulation.

6.3.4. Inlet/Outlet Tubesheet Assemblies

The tubesheet assemblies provide cooling water ingress and egress through the helical tube bundle. There are four of these assemblies, two inlet and two outlet, with one inlet being the mirror image of the other inlet and each being identical to one outlet assembly. Each assembly consists of 600 tubes routed from one-half of the helical bundle into a tubesheet. These lead-in and lead-out tubes are bare (not finned), but are of the same inside diameter and average wall thickness as the finned tubing. These tubes are supported by segmented support plates which are built up as the tubes are installed. The tubes are welded to the tubesheet as they are installed. The tubesheet is attached to an inlet (outlet) pipe by a nozzle-type forging. The pipe outside diameter is smaller than that of the tubesheet. The inlet/outlet piping is routed through the side of the precooler cavity into a separate pipe chase, and a liner transition forging is used to provide a gas-tight seal between the pipe chase and precooler cavity at the pipe inlets and outlets. A gas seal for the outlet pipes is also provided between the upper shroud and pipe to minimize gas bypass leakage into the cavity. Similar seals are not required for the inlet pipes.

6.4. INTERCOOLER STUDIES

Since the intercooler bears a strong functional and physical similarity to the precooler in the single-loop demonstration plant, the overall technical approach taken in its conceptual design is nearly identical to that described for the precooler in the preceding section. Therefore, only the major differences between these designs are discussed here.

The intercooler assembly (Fig. 6-3) is approximately 4.72 m (15.5 ft) in diameter and 18.3 m (60 ft) in length. The active portion of the helical bundle constitutes approximately 10.97 m (36 ft) of this length and contains 1118 finned tubes. To simplify logistics and design verification and support (DV&S) considerations, the intercooler finned tube geometry is identical to that of the precooler. Table 6-3 summarizes the important data for the intercooler heat transfer matrix.

The intercooler fluid routing calls for downward flow of the water through the helical bundle; gas entry is provided at the bottom of the unit. After its upward transit through the helical bundle, the discharge helium is returned to the bottom of the intercooler cavity via the annulus created between the intercooler shroud and the cavity liner, thus accomplishing the required liner cooling in the process. The water feed piping and headering approach is the same as that taken for the precooler; the intercooler is headered into two independent circuits and the water piping is routed laterally through the PCRV to a pipe chase cavity.

REFERENCE

- 6-1. "Gas Turbine HTGR Program, Quarterly Progress Report for the Period Ending March 31, 1978," DOE Report GA-A15002, General Atomic Company, June 1979.

Fig. 6-3. Precooler for 600-MW(e) one-loop intercooled plant

TABLE 6-3
INTERCOOLER HELICALLY COILED FINNED TUBE BUNDLE

Tube outside diameter, ^(a) mm (in.)	31.8 (1.250)
Tube average wall thickness, mm (in.)	2.87 (0.113)
Fin height, mm (in.)	1.60 (0.063)
Fin thickness, mm (in.)	0.51 (0.020)
Number of fins/mm (fins/in.)	0.63 (16)
Average tube length, m (ft)	76.2 (250)
Number of tubes	1118
Bundle inside diameter, m (ft)	0.914 (3)
Bundle outside diameter, m (ft)	4.72 (15.5)
Number of rows	41

(a) As measured across the fins.

7. TURBOMACHINE STUDIES

The major elements of the work done by United Technologies in 1978 consisted of: (1) updating the mechanical design of the 620-MW(e) inter-cooled turbomachine which had originally embodied three journal bearings (Ref. 7-1), and (2) incorporating backup oil seal concepts to minimize oil ingress in the event of a labyrinth seal failure.

The aerodynamic design of the turbomachine remains unchanged from that reported in Ref. 7-1. The mechanical redesign, however, was extensive to establish a viable two-bearing configuration.

Design ground rules for the turbomachine were established by GA; the salient ones are as follows:

1. Turbomachine to be designed for the two-loop commercial plant but to be readily adaptable to the one-loop demonstration plant.
2. Plant layout to be based on a turbomachine arrangement with single turbine inlet and exit ducts.
3. Turbomachine diameter to be constrained to 3.5 m (11.5 ft).
4. Generator drive to be from the compressor end of the turbomachine, facilitating an external thrust bearing.
5. Main access capability to be provided for inspection and maintenance of the two journal bearings.
6. Design to be based on uncooled turbine blades, utilizing an existing nickel-base alloy.

As in the past, United Technologies established a credible turbomachine design. As discussed below, the GA-imposed ground rule of limiting the turbomachine diameter to 3.5 m (11.5 ft) reduced the machine's performance.

An elevation view through the turbomachine for the commercial plant, which has conventional water-cooled and insulated liners throughout, is shown in Fig. 7-1. The turbomachine is installed directly into the cavity; the interface with the PCRV is shown at the 3.5 m (11.5 ft) diameter where the casings are matched with the liner-mounted circumferential seals.

Incorporation of the warm liner feature in the demonstration plant resulted in substantial gas flow path complexity and, as can be seen from Fig. 7-2, the turbomachine cavity provides an annulus for transporting the cold [26.7°C (80°F)] gas to the low-pressure compressor inlet. In this case the turbomachine is installed in a large steel cylinder, which is integrally formed with the PCRV liner. With minor changes to the casings to accommodate the warm liner feature, the basic turbomachine is the same as that for the commercial plant.

Details of the demonstration plant turbomachine are shown in Fig. 7-3. The salient features are summarized in Table 7-1, together with a comparison of the earlier-designed 400-MW(e) non-intercooled machine. Rotor dynamic analysis done by United Technologies confirmed that the critical speed margins were adequate for the two journal bearing systems. In support of maintenance studies, the sequence of machine disassembly was established. The major elements of the casings are shown in Fig. 7-4. To alleviate the concern regarding oil ingress to the primary system resulting from seal failure, backup systems were incorporated in the mechanical design. An independently mounted secondary labyrinth seal was included. This was further backed up by a shaft eccentricity seal and an interspace cavity scavenge system.

The aforementioned design constraint of 3.5 m (11.5 ft) imposed by GA on United Technologies for the turbomachine diameter severely restricted and penalized the design. The 3.5 m (11.5 ft) diameter value is a

Fig. 7-1. HTGR turbomachine, conventional liner

Fig. 7-2. HTGR turbomachine, warm liner

Fig. 7-3. HTGR 620-MW(e) turbomachine

TABLE 7-1
COMPARISON OF HTGR-GT TURBOMACHINES

	Three-Loop Non-Intercooled	Two-Loop (a) Intercooled
Nominal machine rating, MW(e)	400	620
Machine type	Single shaft	Single shaft
Number of turbine inlet ducts	One	One
Rotor construction	Welded	Welded
Blading life, h	280,000	280,000
Turbine blading	Uncooled, IN-100	Uncooled, IN-100
Compressor pressure ratio	2.5	3.0 (1.73/1.73)
Compressor stages	18	8 + 8
Maximum compressor tip diameter, cm (in.)	182.6 (71.9)	218.8 (85.9)
Blade height (min/max), mm (in.)	80.01/125.73 (3.15/4.95)	76.2/165.1 (3.0/6.5)
Compressor efficiency (across blading), %	89.8	90.8 (LP) 90.2 (HP)
Turbine stages	8	9
Maximum turbine tip diameter, cm (in.)	218.4 (86)	240.0 (94.5)
Blade height (min/max), mm (in.)	125.7/297.2 (4.95/11.7)	165.1/393.7 (6.5/15.5)
Turbine efficiency (across blading), %	91.8	92.2
Machine diameter, m (ft)	3.51 (11.5) ^(b)	3.51 (11.5) ^(b)
Machine length, m (ft)	11.27 (37)	15.85 (52)
Machine weight, mton (tons)	276 (305)	317 (350)
Generator drive end	Compressor	Compressor
Rotor burst shield	Integral part of machine structure	Integral part of machine structure
Number of journal bearings	2	2
Thrust bearing position	External to PCRV	External to PCRV
Journal bearing man access	Yes	Yes
Inlet and exit losses ($\Delta P/P$), %	1.80	3.4

(a) Basic machine "core" adaptable to one-loop demonstration plant with changes to the casings to accommodate the warm liner features.

(b) Design constraint.

A. FULLY ASSEMBLED

B. SPLIT CASES SEPARATED

C. SHAFT SEPARATED

Fig. 7-4. Turbomachine disassembly

carryover from earlier studies (of smaller loop ratings) and was adopted to be consistent with the maintenance philosophy of transporting a failed turbomachine to a centralized facility for maintenance and overhaul. The 3.5 m (11.5 ft) diameter facilitates rail transportation when a contaminated turbomachine is installed in a shielded cask. Two factors have changed since this earlier accepted position was established: (1) there is a growing interest on the part of the utilities for on-site maintenance (the impact of this on the design of the HTGR-GT plant is being studied in FY-79), and (2) the loop ratings have increased from 275 to 620 MW(e) since the 3.5 m (11.5 ft) diameter limit was established.

For the 400-MW(e) non-intercooled turbomachine, an acceptable design was established by United Technologies in 1976 with the 3.5 m (11.5 ft) diameter casing limitation. Increasing the mass flow rate through the turbomachine by about 50% [for the 620-MW(e) intercooled variant], and limiting the diameter to the 3.5 m (11.5 ft) value resulted in fluid dynamic penalties. The penalties were twofold: (1) the limited space available for inlet and outlet scrolls and volutes gave unacceptably high pressure losses (as shown in Table 7-2), and (2) the flow distribution at the turbine and compressor inlets was considered very poor and would result in further degradation in performance. Subsequent work performed by United Technologies has indicated that for an intercooled machine in the 620-MW(e) rating class, an acceptable design can be realized with a casing diameter of 3.96 m (13 ft).

REFERENCE

- 7-1. "Gas Turbine Program, Quarterly Progress Report for the Period Ending March 31, 1977," ERDA Report GA-A14381, General Atomic Company, April 1977.

TABLE 7-2
COMPARISON OF HTGR-GT PLANT PRIMARY SYSTEM PRESSURE LOSSES

	One-Loop Demonstration Plant	Commercial Plant	
		Two Loop	Three Loop
Core thermal power, MW(t)	1530	3000	3000
Loop rating, MW(t)	1530	1500	1000
Thermodynamic cycle	Intercooled	Intercooled	Non-intercooled
Liner type	Warm liner	Insulated and water cooled	Insulated and water-cooled
Plant layout drawing	024628	024629	024760
PCRV diameter, m (ft)	36.3 (119)	42.7 (140)	39.3 (129)
Primary system duct losses ($\Delta P/P$), %			
Core to turbine	0.25	0.21	0.108
Turbine to recuperator	0.41	0.53	0.387
Recuperator to precooler	0.31	0.53	0.245
Precooler to compressor	1.76	0.03	0.008
Compressor to intercooler	0.55	0.51	---
Intercooler to high-pressure compressor	0.97	0.36	---
Compressor to core cavity	0.36		
Core cavity loss	0.01	0.70	0.141
Core cavity to recuperator	0.66		
Recuperator to core inlet	1.25	0.46	0.522
Total duct losses, %	6.53 ^(a)	3.33	1.411
Heat exchanger losses ($\Delta P/P$), %			
Recuperator high-pressure side	0.82	0.82	0.62
Recuperator low-pressure side	1.43	1.43	1.22
Precooler loss	1.08	1.08	0.99
Intercooler loss	0.50	0.50	---
Total heat exchanger losses	3.83	3.83	2.83
Turbomachine diameter, m (ft)	3.5 (11.5)	3.5 (11.5)	3.5 (11.5)
Turbomachine inlet and exit losses ($\Delta P/P$), %			
Turbine inlet	0.48	0.48	0.364
Turbine exit	0.39	0.39	0.46
Low-pressure compressor inlet	0.33	0.20	---
Low-pressure compressor exit	0.90	0.90	---
High-pressure compressor inlet	0.24	0.24	0.25
High-pressure compressor exit	1.15	1.15	0.73
Total turbomachine losses	3.49 ^(b)	3.36 ^(b)	1.804
Core pressure loss ($\Delta P/P$), %	0.69	0.69	1.11
Total primary system loss ^(c) ($\Delta P/P$), %	14.54	11.21	7.16
Plant efficiency (ISO, 15°C), %	39.65	41.2	39.7
Efficiency potential with second design iteration improvements, %	41.0	42.5	40.0

(a) High losses associated with complex warm liner gas flow paths.

(b) High inlet and outlet pressure losses for 620-MW(e) turbomachine because of 3.5 m (11.5 ft) design constraint on diameter.

(c) Losses not strictly additive, but indicative of layout merit.

8. ALTERNATE COMMERCIAL PLANT CONCEPTUAL DESIGN STUDIES

8.1. GENERAL

A conceptual design study was performed for the two-loop 1200-MW(t) commercial plant which included a series of different plant arrangements for the purpose of selecting the best plant arrangement with regard to PCR/V design, position of components, maintenance capability, and plant performance and cost.

All configurations are integrated designs with vertical heat exchangers and two horizontal turbomachine units. The PCR/Vs are conventionally prestressed by longitudinal tendons and circumferential wire winding for the portion containing the vertical cavities and by a horizontal tendon system in the area of the horizontal turbomachine cavities. An effort was made to arrange the cavities and connecting ducts of all the configurations so as to achieve the greatest efficiency of the prestressing system. The cavity dimensions tentatively assumed for this study are given in Table 8-1. The configurations studied, including a technical summary, are listed in Table 8-2 and are illustrated and briefly described in Section 8.3.

The configuration study was divided into the following categories as a function of liner cooling configuration as shown in Figs. 8-1 through 8-4:

Configuration A. Concepts with a conventional water-cooled and insulated liner in the core cavity and warm liners in the heat exchanger and turbomachine cavities (Figs. 8-5 through 8-10, Sections 8.3.1 through 8.3.5).

TABLE 8-1
CAVITY DIMENSIONS AND DESIGN DATA FOR ALTERNATE COMMERCIAL PLANT

	Warm Liner	Conventional Liner
Core cavity diameter, m (ft-in.)	14.02 (46-0)	12.65 (41-6)
Core cavity height, m (ft-in.)	17.20 (56-5)	14.96 (49-1)
Recuperator cavity top section, m (ft-in.)	7.32 (24-0)	7.32 (24-0)
Recuperator cavity bottom section, m (ft-in.)	7.01 (23-0)	7.01 (23-0)
Precooler cavity, m (ft-in.)	5.94 (19-6)	5.94 (19-6)
Intercooler cavity, m (ft-in.)	5.11 (16-9)	5.11 (16-9)
Turbomachine cavity, m (ft-in.)	4.72 (25-6)	4.11 (13-6)
Maximum system pressure, bar (psi)	78.76 (1142)	78.76 (1142)
Reactor outlet temperature, °C (°F)	850 (1562)	850 (1562)

TABLE 8-2
TECHNICAL SUMMARY OF CONFIGURATIONS STUDIED

Configu- ration	Component Size			Turbomachine Location	PCRV					
	Recup- erator	Pre- cooler	Inter- cooler		Shape	Diameter [m (ft)]	Height [m (ft)]	Cross- section Area [m ² (ft ²)]	Volume (b) [x 10 ³ m ³ (x 10 ³ ft ³)]	Offset [m (ft)]
A-1	F ^(a)	F	F	Chordal parallel	Circ.	40.54 (133)	35.97 (118)	1,290 (13,893)	46.40 (1639)	0
A-2	1/2	1/2	F	Chordal nonparallel	Circ.	43.59 (143)	35.37 (116)	1,492 (16,060)	52.74 (1863)	2.13 (7)
A-3	F	F	F	Diagonal in-line	Oval	45.42/35.97 (149/118)	35.97 (118)	1,283 (13,809)	46.12 (1629)	0
A-4	F	F	F	Chordal nonparallel	Circ.	44.20 (145)	35.97 (118)	1,534 (16,513)	55.18 (1949)	1.07 (3.5)
A-4a	F	F	F	Chordal nonparallel	Oval	45.11/38.40 (148/126)	35.97 (118)	1,361 (14,646)	48.92 (1728)	0
A-5	F	F	F	Chordal parallel	Circ.	42.06 (138)	32.00 (105)	1,390 (14,957)	44.45 (1570) + 2.26 (80) 46.71 (1650)	1.22 (4)
B-1	F	F	F	Chordal parallel	Circ.	40.54 (133)	35.97 (118)	1,291 (13,893)	46.40 (1639)	0
B-2	F	F	F	Chordal parallel	Circ.	42.06 (138)	35.97 (118)	1,390 (14,957)	49.97 (1765)	0.30 (1)
B-2A	F	F	F	Chordal nonparallel	Circ.	42.06 (138)	35.97 (118)	1,390 (14,957)	49.97 (1765)	0.30 (1)
B-3	F	F	F	Chordal nonparallel	Circ.	40.54 (133)	35.97 (118)	1,291 (13,893)	46.40 (1639)	0
B-4	F	F	F	Chordal nonparallel	Oval	45.11/38.40 (148/126)	35.97 (118)	1,361 (14,646)	48.92 (1728)	0.61 (2)
B-5	F	F	F	Diagonal in-line	Oval	45.41/38.40 (149/126)	35.37 (116)	1,370 (14,745)	48.41 (1710)	0
B-6	F	F	F	Diagonal in-line	Oval	49.38/38.40 (162/126)	32.00 (105)	1,490 (16,032)	47.65 (1683) + 2.26 (80) 49.91 (1763)	0
C-1	F	F	F	Chordal parallel	Circ.	40.84 (134)	36.88 (121)	1,310 (14,102)	48.30 (1706)	0
D-1	F	F	F	Chordal nonparallel	Circ.	41.15 (135)	36.88 (121)	1,430 (14,313)	49.03 (1732)	0
D-2	F	F	F	Chordal nonparallel	Circ.	44.20 (145)	36.88 (121)	1,534 (16,513)	56.56 (1998)	0.91 (3)

(a) F = full size.

(b) Volume including all cavities.

Fig. 8-1. Primary gas flow, configuration A, core cavity - conventional liners, heat exchanger and turbomachine cavities - warm liners

Fig. 8-2. Primary gas flow, configuration B, all cavities - conventional liners

Fig. 8-3. Primary gas flow, configuration C, all cavities - warm liners

Fig. 8-4. Primary gas flow, configuration D, core cavity - warm liners, turbomachine and heat exchanger cavities - conventional liners

Configuration B. Concepts with conventional water-cooled and insulated liners in all cavities where required (Figs. 8-11 through 8-17, Sections 8.3.6 through 8.6.12).

Configuration C. Concept with warm liners in all cavities (Fig. 8-18, Section 8.3.13).

Configuration D. Concepts with a warm liner in the core cavity and conventional water-cooled and, where required, insulated liners in the heat exchanger and turbomachine cavities (Figs. 8-19 and 8-20, Sections 8.3.14 and 8.3.15).

In order to select the best commercial plant concept, all configurations were evaluated on the basis of a simplified rating system. While it is recognized that evaluating and rating a family of plant variants is difficult, some form of comparative selection criteria was necessary to accomplish a comparative evaluation of the plant concepts. Therefore, the preliminary rating of the plant concepts was performed according to the parameters listed in Table 8-3. It could be argued that the rating procedure would have been more meaningful if "weighting factors" had been applied to emphasize areas of particular importance. This is recognized, but such a method tends to become complex, particularly the selection of factors. Therefore, the simplified selection of parameters was used instead.

For the two-loop configuration the multiplicity of main vertical cavities in the PCRV, the arrangement of auxiliary cavities as the CACS cavities, the cavities required for the cavity liner cooling or warm liner cooling system, and the circulating water headers of the water heat exchangers when fed from the side became increasingly more important and thus played an important role in the evaluation of the concepts.

For the two-loop configuration an arrangement of all required cavities in an oval PCRV appeared to have economical advantages. One reason for not

TABLE 8-3
COMPARATIVE EVALUATION OF HTGR-GT TWO-LOOP COMMERCIAL PLANT CONCEPTS

Features and Design Considerations	Evaluation Parameter (a)																	
	A						B								C		D	
	1	2	3	4	4A	5	1	2	2A	3	4	5	6	1	1	2		
PCRV																		
Diameter minimization	G	P	G	P	A	G	G	A	A	G	A	A	P	G	G	P		
Thermal barrier minimization	A	A	A	A	A	A	P	P	P	P	P	P	P	G	A	A		
Gas flow path complexity	P	P	P	P	P	P	G	G	G	G	G	G	G	P	A	A		
Structures (CPS, tendon layout, etc.)	A	A	P	A	A	G	A	A	A	A	P	P	A	A	A	A		
Quantity minimization	G	A	G	P	A	G	A	A	A	A	A	A	A	G	A	P		
Auxiliary cavities positioning	P	P	P	G	G	A	NA	A	G	NA	G	G	G	NA	P	A		
Liner																		
Core cavity	G	G	G	G	G	G	G	G	G	G	G	G	G	NT	NT	NT		
Component cavities	NT	NT	NT	NT	NT	NT	G	G	G	G	G	G	G	NT	G	G		
Turbomachine																		
Location	G	G	A	G	G	G	G	G	G	G	G	A	A	G	G	G		
Duct interfaces	G	G	G	G	G	G	G	G	G	G	G	G	G	G	G	G		
Inlet and outlet loss speculation	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A		
Coolant gas flow paths	P	P	P	P	P	P	G	G	G	G	G	G	G	P	A	A		
Component sizes	A	G	A	A	A	A	A	A	A	A	A	A	A	A	A	A		
Performance (ΔP speculation)	P	P	P	P	P	P	G	G	G	G	G	G	G	P	A	A		
Maintenance capability	A	A	A	A	A	A	A	A	A	A	A	A	A	G	A	A		
Inservice inspection capability	A	A	A	A	A	A	P	P	P	P	P	P	P	G	A	A		
Safety and licensing	A	A	A	A	A	A	G	G	G	G	A	A	A	A	P	P		
Plant cost speculation	A	P	A	P	A	G	A	P	P	A	P	P	P	G	A	P		
Overall evaluation points	53	47	49	47	53	59	65 1 NA	64	66	65 1 NA	62	60	60	55 1 NA	55	49		

(a) G = good (5 points); A = acceptable (3 points); P = poor (1 point); NA = not acceptable (0 points); and NT = substantial new technology.

pursuing this configuration further was difficulties expected with circumferential prestressing by wire winding. Comparative values of the cross-sectional areas and volumes are included in Table 8-2.

Through the selection process shown in Table 8-3, variant B-2A (Fig. 8-13, Section 8.3.8), a configuration with all conventional liners, was selected as the best commercial configuration. A conceptual layout of the selected configuration was subsequently made (Fig. 8-21, Section 8.4).

Because of the concern over installation and shipping problems for the large components, especially the recuperator, configurations with half-size heat exchangers were also investigated.

8.2. PRIMARY COOLANT GAS FLOW

The different coolant gas flows through the primary loop systems used in this study are diagrammatically shown in Figs. 8-1 through 8-4; the diagrams are self-explanatory.

8.3. DESCRIPTION OF CONFIGURATIONS

A brief description of the configurations investigated is given in the following sections.

8.3.1. Configuration A-1 (Fig. 8-5)

This configuration has a conventional liner* in the core cavity and bare liners in all other cavities. The turbomachines are chordally oriented and are parallel to each other. The recuperators are directly above the turbine exhaust end of the turbomachine cavities, and the precoolers and intercooler are arranged so that they are not over the turbomachine cavity. The precooler location is dependent on the gas flow path with a warm liner

*Conventional liner means the liner has a thermal barrier on the inside and water-cooling coils on the outside surface.

Fig. 8-5. Configuration A-1 - 1200-MW(e) HTGR-GT two-loop intercooled plant; conventional liner in core cavity, warm liner in other cavities

in the component cavities and is most advantageously located as close to the recuperators as possible, as shown for this configuration. The intercoolers are located between the three CACS cavities on the opposite side of the precoolers. The concentric intercooler ducts are connected to the turbomachine cavities with 90° bends.

The circular PCRV has a diameter of 40.54 m (133 ft) and a height of 36 m (118 ft). In the top view the core cavity is centrally located.

8.3.2. Configuration A-2 (Fig. 8-6)

For this configuration half-size recuperators and a precooler are installed. The arrangement of all cavities requires a slightly angled arrangement of the turbomachine cavities and, in the top view, an offset of 2.13 m (7 ft) of the core cavity to the centerline of the PCRV. The circular PCRV has a diameter of 43.59 m (143 ft) and a height of 35.36 m (116 ft).

8.3.3. Configuration A-3 (Fig. 8-7)

This is a configuration with an in-line arrangement of the turbomachine. The diameter required for a circular PCRV would have been too large [43.6 m (143 ft)]; therefore, an elliptical PCRV was developed which has the same cross-sectional area as the circular PCRV of configuration A-1 and provides the required structural ligaments for the cavities. The core cavity is centrally located. The major and minor axes of the ellipse are 45.42 m (149 ft) and 35.97 m (118 ft), respectively.

8.3.4. Configuration A-4 (Fig. 8-8)

For this configuration the intercoolers are moved to the outside of the turbomachine and the turbomachine cavities are angled. This arrangement results in an increased PCRV diameter and excess unused volume of the PCRV at one side of the PCRV. It appeared that an oval PCRV would provide a more

Fig. 8-6. Configuration A-2 - 1200-MW(e) HTGR-GT two-loop intercooled plant; conventional liner in core cavity, warm liner in other cavities

Fig. 8-7. Configuration A-3 - 1200-MW(e) HTGR-GT two-loop intercooled plant; conventional liner in core cavity, warm liner in other cavities

Fig. 8-8. Configuration A-4 - 1200-MW(e) HTGR-GT two-loop intercooled plant; conventional liner in core cavity, warm liner in other cavities

economical design from the standpoint of PCRV materials. This design is shown in Fig. 8-9. The oval PCRV offers a reduction in cross-sectional area of more than 167 m^2 (1800 ft^2).

8.3.5. Configuration A-5 (Fig. 8-10)

The prestressing system of the HTGR-GT PCRV with vertical heat exchanger cavities and horizontal turbomachine cavities in a vertically oriented cylindrical body has always been a difficult problem. In the system adopted to date, the PCRV is conventionally prestressed by longitudinal tendons and circumferential wire winding for the vertical cavities and by a diagonal tendon system in the area of the horizontal turbomachine cavity where wire winding can not be applied. This is an inefficient PCRV design with excess concrete and reinforcing and tendon steel for that portion of the PCRV where the horizontal turbomachine cavities are located.

The design shown in Fig. 8-10 is an attempt to ease this problem by combining two separate prestressed pressure vessels, one for the vertical heat exchanger and core cavities and one for the horizontal turbomachine cavities. The main vertical PCRV has minimum height with sufficient height for the recuperator, precooler, and intercoolers. This PCRV is conventionally prestressed by tendons for longitudinal prestress and a wire-winding system for circumferential prestress. The horizontal PCRVs for the turbomachine cavities are separate entities and are prestressed by longitudinal tendons in the horizontal direction and by a hoop tendon system for circumferential prestress. The turbomachine PCRVs are located between the vertical PCRV and the base mat and structurally connected to both by an appropriate system of reinforcing bars.

All duct connections to the turbomachine are from the top; a closure provides access for installation and removal of the turbomachine.

Fig. 8-9. Configuration A-4A - 1200-MW(e) HTGR-GT two-loop intercooled plant; conventional liner in core cavity, warm liner in other cavities

Fig. 8-10. Configuration A-5 - 1200-MW(e) HTGR-GT two-loop intercooled plant; conventional liner in core cavity, warm liner in other cavities (Sheet 1 of 2)

Fig. 8-10. Configuration A-5 - 1200-MW(e) HTGR-GT two-loop intercooled plant; conventional liner in core cavity, warm liner in other cavities (Sheet 2 of 2)

As for all other PCRVs, the PCRv support consists of a central pedestal and a concrete ring at the outer edge of the vertical PCRv with circumferential openings for the generator, access to the tendon heads, and pipe connections of the heat exchangers.

This PCRv arrangement provides a most efficient circumferential prestress, a minimum of longitudinal tendons (only for core and heat exchanger cavities in the vertical PCRv and horizontal longitudinal tendons in the turbomachine cavity), and no diagonal and vertical tendons for the turbomachine cavity. As a replacement for the latter two, a hoop tendon system needs to be developed.

In the top view the core cavity is offset 1.22 m (4 ft) from the center of the PCRv.

8.3.6. Configuration B-1 (Fig. 8-11)

This configuration, which has conventional liners (as defined in Section 8.3.1) in all cavities (primary gas flow, see Fig. 8-2), requires a different arrangement of the precooler. As can be seen in Fig. 8-11, the recuperator is directly over the turbine exhaust end of the turbomachine and is connected to the turbomachine cavity by a short vertical duct. From the recuperator the primary gas flow is through a horizontal duct into the precooler, which is located directly over the low-pressure compressor inlet and connected to it by a short vertical duct. The intercooler is not positioned over the turbomachine cavity and is connected to the cavity by a concentric duct. All three CACS cavities are positioned between the two recuperators. This cavity arrangements fits into the minimum (stress-wise) PCRv diameter without consideration of cooling-water pipe chases for the liner cooling system and side-fed water-gas heat exchangers.

In the top view the core is in the center of the PCRv.

Fig. 8-11. Configuration B-1 - 1200-MW(e) HTGR-GT two-loop intercooled plant;
all conventional liners

8.3.7. Configuration B-2 (Fig. 8-12)

This configuration has a slightly larger PCRV diameter to accommodate the required water pipe chases, but still retains the parallel arrangement of the turbomachine cavities. The larger PCRV also permits a redistribution of the CACS units.

This arrangement requires a slight offset of approximately 0.305 m (1 ft) of the core cavity from the center of the PCRV.

8.3.8. Configuration B-2A (Fig. 8-13)

This configuration incorporates all pipe chases for the precoolers and intercoolers, as well as the PCRV cooling water system. This necessitates a slight deviation from the parallel arrangement of the turbomachine cavities, without a change in PCRV diameter but with a slight change of the offset of the core.

8.3.9. Configuration B-3 (Fig. 8-14)

This configuration was another attempt to accommodate all cavities in a circular PCRV.

8.3.10. Configuration B-4 (Fig. 8-15)

This configuration has an oval PCRV, but does not result in a smaller PCRV.

8.3.11. Configuration B-5 (Fig. 8-16)

Configuration B-5 has an in-line arrangement of the turbomachine, similar to that of configuration A-3 (Fig. 8-7), in an oval PCRV. This configuration shows no advantages.

Fig. 8-12. Configuration B-2 - 1200-MW(e) HTGR-GT two-loop intercooled plant; all conventional liners (Sheet 1 of 2)

Fig. 8-12. Configuration B-2 - 1200-MW(e) HTGR-GT two-loop intercooled plant;
all conventional liners (Sheet 2 of 2)

Fig. 8-13. Configuration B-2A - 1200-MW(e) HTGR-GT two-loop intercooled plant;
all conventional liners (Sheet 1 of 2)

Fig. 8-13. Configuration B-2A - 1200-MW(e) HTGR-GT two loop intercooled plant;
all conventional liners (Sheet 2 of 2)

Fig. 8-14. Configuration B-3 - 1200-MW(e) HTGR-GT two-loop intercooled plant;
all conventional liners

Fig. 8-15. Configuration B-4 - 1200-MW(e) HTGR-GT two-loop intercooled plant;
all conventional liners

Fig. 8-16. Configuration B-5 - 1200-MW(e) HTGR-GT two-loop intercooled plant;
all conventional liners (Sheet 1 of 2)

Fig. 8-16. Configuration B-5 - 1200-MW(e) HTGR-GT two-loop intercooled plant; all conventional liners (Sheet 2 of 2)

8.3.12. Configuration B-6 (Fig. 8-17)

For configuration B-6, the PCR design of configuration A-5 (separate turbomachine PCRVs) was applied to configuration B-5. This configuration results in a larger PCR.

8.3.13. Configuration C (Fig. 8-18)

This configuration has warm liners in all cavities with a primary cooling gas flow as shown in Fig. 8-3.

8.2.14. Configuration D-1 (Fig. 8-19)

This configuration has a warm liner in the core cavity and conventional liners in all other cavities. The primary coolant flow is shown in Fig. 8-4. This coolant flow permits location of the precooler directly above the turbomachine and arrangement of the intercoolers and CACS units similar to configuration B-2.

In the top view the center of the core is in the center of the PCR.

8.3.15. Configuration D-2 (Fig. 8-20)

For this configuration the precoolers are not located over the top of the turbomachine cavity in order to provide bottom-fed feedwater supply. This arrangement increases the PCR diameter. The center of the core is offset from the center of the PCR approximately 0.61 m (2 ft).

8.4. CONCEPT LAYOUT FOR TWO-LOOP 3000-MW(t) ALTERNATE HTGR PLANT

To assist subcontractor vendors, the sketches of configuration B-2A (Fig. 8-13) were upgraded. The drawings shown in Fig. 8-21 (sheets 1 through 4) do not represent engineered layouts but rather are arrangement drawings to illustrate the main features of the selected two-loop plant and to help

Fig. 8-17. Configuration B-6 - 1200-MW(e) HTGR-GT two-loop intercooled plant

Fig. 8-18. Configuration C - 1200-MW(e) HTGR-GT two-loop intercooled plant with warm liner

Fig. 8-19. Configuration D-1 - 1200-MW(e) HTGR-GT two-loop intercooled plant;
warm liner in core cavity only

Fig. 8-20. Configuration D-2 - 1200-MW(e) HTGR-GT two-loop intercooled plant; warm liner in core cavity only

component vendors with respect to gas flow path geometries and interface areas. These drawings serve as a starting point for structural engineering work for a primary system layout drawing.

8.5. ADDITIONAL TWO-LOOP PLANT VARIANTS

Alternate approaches embodying a conventional liner in the core cavity and elimination of the thermal barrier elsewhere to the highest degree possible were further studied and evaluated. For the following reasons, it was concluded that only configuration B-2A be pursued:

1. Warm liners will be studied for the one-loop demonstration plant.
2. A direct comparison is possible between the two-loop (intercooled) and three-loop (non-intercooled) plants.
3. Component designs are well under way by United Technologies and Combustion Engineering for configuration B-2A.

Fig. 8-21. PCR configuration B-2A
for two-loop HTGR-GT al-
ternate commercial plant
(sheet 2 of 4)

SECTION ID-13

Fig. 8-21. PCRV configuration B-2A
 for two-loop HTGR-GT al-
 ternate commercial plant
 (sheet 3 of 4)

Fig. 8-21. PCRV configuration B-2A
 for two-loop HTGR-GT al-
 ternate commercial plant
 (sheet 4 of 4)

9. ALTERNATE COMMERCIAL PLANT STRUCTURAL DESIGN STUDIES

Two PCRV configurations were studied for the conceptual design of the two-loop intercooled commercial plant. Configuration B-2A, which embodies conventionally insulated and water-cooled liners throughout all cavities, was selected. In this concept, the core inlet and outlet temperatures during CAHE operation, by virtue of insulating the liners, can be held at the same values that exist during main loop operation. This enables a reduction in the CAHE size and simplification of the auxiliary loop helium flow path, which is reflected in the smaller and simpler configuration of the auxiliary loop liners, equipment supports, and closures. Insulating the liners also eliminates the need to sweep the turbomachine cavity liner with cold gas. The pressure-resistant shroud within the turbomachine cavity, which was required for the demonstration plant, is therefore eliminated. The resulting turbomachine cavity liner configuration more closely resembles that of a non-intercooled loop design than the intercooled demonstration plant design. The conceptual design for configuration B-2A as shown in Fig. 9-1 resulted in a PCRV with an outside diameter of 42.7 m (140 ft) and a height of 48.8 m (116 ft).

A study of an alternate concept, A-1, was conducted to evaluate the effects of employing a cold liner in the core cavity and warm liners elsewhere. The major ramification of the limited warm liner approach was in the auxiliary cooling loop and turbomachine cavity liner and closures, as indicated in the demonstration plant. Although lack of detailed interfacing equipment outlines precluded more than a cursory study of the auxiliary cooling loop arrangement, the turbomachine cavity was studied at some length. To accommodate a warm liner in this cavity, it would be necessary to introduce either a main bypass duct below the cavity or a pressure-restraint shroud within the cavity. The bypass duct would lead to increased design and construction complexities, a longer construction

schedule, and increased PCRV dimensions. The pressure-resisting shroud, which was used for the demonstration plant, would not affect construction time or PCRV size. The shroud was 76 mm (3 in.) thick and required insulation on portions of the inside diameter. To minimize thermal stresses, the shroud was suspended from the cross ducts. Thus, all turbomachine loadings were transmitted to the PCRV through shear anchors attached to the cross-duct liners. The PCRV configuration for alternate concept A-1 is shown in Fig. 9-2. The PCRV diameter and height are 43.9 m (144 ft) and 48.8 m (116 ft), respectively.

During the fourth quarter, preliminary layout drawings for thermal barrier and reactor internals were released. The reactor internal drawings (Fig. 9-3) developed are common to the two-loop and reference three-loop commercial plants. Thermal barrier drawings included the areas of the hot duct between the core outlet and the turbomachine and the lower core plenum (Figs. 9-4 and 9-5). Major design issues for the thermal barrier have been identified and are summarized as follows:

1. A carbon/carbon (C/C) composite material has been identified for use as the cover plate material in the high-temperature regions such as the lower cross duct and the lower plenum. No data are available on the use of C/C materials in HTGRs. In particular, expected high sound power levels (up to about 170 db) and high duct velocities (up to about 100 m/s) could cause problems with this material.
2. Removability requirements for parts of the hot duct will require development of flange coupling and removal guide systems.
3. The rapid depressurization rates expected in the HTGR-GT will require design changes to the standard thermal barrier metallic cover plate and fibrous insulation design.

GENERAL NOTES:

1. MINIMUM LINEAR PRESTRESSING TENDON FORCE IMMEDIATELY AFTER ANCHORING SHALL BE 1755 KIPS.
2. PRECAST CONCRETE PANELS SHALL BE USED AS FORMS FOR PLACEMENT OF PCRV CONCRETE.
3. ALL TENDON BEARING PLATES SHALL BE GROUTED IN-PLACE AFTER CONCRETING OF THE PCRV.
4. ALL SPECIFICATIONS FOR THE 8850 MW (E) LEAD PLANT ARE APPLICABLE TO THIS DRAWING.
5. PCRV SIZING IS BASED ON THE CAVITY ENVELOPE INFORMATION DOCUMENTED IN MEMO NO. 58D. PB-184-76 EXCEPT AS NOTED BELOW.
THE SIDE CAVITY DIMENSIONS ARE BASED ON THE FOLLOWING COMBUSTION ENGINEERING DRAWINGS:
E 63678255 - RECUPERATOR CAVITY
E 63678256 - PRECOOLER CAVITY
E 63678257 - INTERCOOLER CAVITY
6. DIMENSIONS AND OPENINGS SHOWN IN THE PCRV SUPPORT STRUCTURE ARE SUGGESTED ARRANGEMENTS ONLY. FINAL CONFIGURATION AND LOCATION ARE PART OF THE SDP DESIGN.
7. THE DUCTS BETWEEN CAVITIES SHOWN ON THE DRAWING IS SCHEMATIC AND REQUIRES VERIFICATION BY THE INTERFACING ORGANIZATIONS.
8. DIMENSIONS SHOWN ON THIS DRAWING ARE VERY PRELIMINARY.
9. THE FOLLOWING MAJOR REVISIONS WERE INCORPORATED ON THIS DRAWING:
A. ADDED SHEET 2- VERTICAL SECTION THRU THE PCRV.
B. TENDON ARRANGEMENT IN THE TOP HEAD.

Fig. 9-1. PCRV configuration B-2A for HTGR-GT two-loop commercial plant, Sheet 1

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Fig. 9-1. PCRV configuration B-2A for HTGR-GT two-loop commercial plant, Sheet 2

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GENERAL NOTES:

1. MINIMUM LINEAR PRESTRESSING TENDON FORCE IMMEDIATELY AFTER ANCHORING SHALL BE 1735 KIPS.
2. PRECAST CONCRETE PANELS SHALL BE USED AS FORMS FOR PLACEMENT OF PCRV CONCRETE.
3. ALL TENDON BEARING PLATES SHALL BE GROUTED IN PLACE AFTER CONCRETING OF THE PCRV.
4. ALL SPECIFICATIONS FOR THE 3360 MW(L) LEAD PLANT ARE APPLICABLE TO THIS DRAWING.
5. PCRV SIZING IS BASED ON THE CAVITY ENVELOPE INFORMATION DOCUMENTED IN MEMO NO. SED.PS. 289-75.
6. DIMENSIONS AND OPENINGS SHOWN IN THE PCRV SUPPORT STRUCTURE ARE SUGGESTED ARRANGEMENTS ONLY. FINAL CONFIGURATION AND LOCATION ARE PART OF THE BOP DESIGN.
7. THE DUCTING BETWEEN CAVITIES SHOWN ON THIS DRAWING IS SCHEMATIC AND REQUIRES VERIFICATION BY THE INTERFACING ORGANIZATIONS.
8. ALL DIMENSIONS SHOWN ON THIS DRAWING ARE VERY PRELIMINARY.

Fig. 9-2. PCRV configuration A-1 for HTGR-GT two-loop commercial plant

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Fig. 9-3. Reactor internals layout for HTGR-GT commercial plant, configuration B-2A (sheet 2 of 3)

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Fig. 9-3. Reactor internals layout for HTGR-GT commercial plant, configuration B-2A (sheet 3 of 3)

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Fig. 9-4. Thermal barrier lower plenum

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SCALE : 1/24 OR [1/2 IN = 1 FT] + NOTED

Fig. 9-5. Hot duct thermal barrier

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4. Due to the elevated temperatures in the lower plenum, it is expected that higher temperature hard ceramic materials will be required for the thermal barrier in that region.

10. ALTERNATE COMMERCIAL PLANT SYSTEM ENGINEERING

10.1. PERFORMANCE CODE DEVELOPMENT

The cost optimization and design evaluation routine (CODER) is a computerized method developed to perform parametric and plant optimization studies for a HTGR-GT operating on a non-intercooled cycle. The optimization is performed on the basis of thermodynamic considerations and component design economic tradeoffs, which are derived from factors affecting the overall plant. In order to calculate plant performance for the intercooled design, CODER was reprogrammed to include a two-stage compressor with an intercooler heat exchanger. The addition of the intercooler for the two-loop alternate commercial plant was made on the premise of increased plant thermodynamic efficiency. This is the result of adding a second stage of compression at a higher density, which reduces the work of compression and results in an increase in network output. The additional heat exchange equipment and ducting, increased pressure drop, and resulting added complexity of the PCRV design are all factors which tend to negate the advantage which is gained in plant output with the addition of the intercooler and therefore need to be assessed.

In addition to being programmed to calculate intercooled cycle performance, two different program options have been included. One program option can be used to perform thermodynamic calculations for a plant with conventional liner cooling, while the other is used to perform calculations for a plant embodying the warm liner concept. Six additional independent variables have been incorporated. Specifically, these are the intercooler tube diameter, cavity diameter, outlet temperature, outlet water temperature, tube pitch-to-diameter ratio, and the high-pressure compressor pressure ratio.

In order to perform plant optimization computations for the intercooled cycle with CODER, extensive modifications are required in the PCRIV and DUCT subroutines since the added pieces of equipment and new layout for the intercooled cycle completely change the PCRIV design and the ground rules used for PCRIV sizing. Furthermore, the manner in which the program is presently structured to generate the optimized design requires that the component geometry and general arrangement be defined in advance; i.e., an initial base case is programmed in CODER and subsequent cost optimizations are made by making perturbations in the base case and iterating until an optimum cost is attained. In addition, base costs and cost algorithm updates are necessary before plant optimization studies can be initiated.

During this report period, an HTGR-GT PCRIV sizing subroutine model was developed for eventual use in CODER. This model calculates, for various independent design variables, the basic PCRIV dimensions and cavity locations. From this geometry data base, a multitude of cost and performance parameters can be determined, including concrete and prestressing related quantities, thermal barrier areas, duct lengths, pressure drops, and cavity and duct heat losses. Additional programming changes are required, however, in order to implement the PCRIV sizing routine and complete the PCRIV optimization model.

PCRIV computer modeling work was performed on configurations B-3 [40.5 m (133 ft) diameter reference base case] and B-2 [42.1 m (138 ft) diameter reference base case] but was not completed since the PCRIV design evolved such that configuration B-2A became the reference base case, and most of the reprogramming effort was therefore centered around configuration B-2A. A PCRIV sizing model for this configuration was completed; the model calculates PCRIV diameter and placement of the CACS, reactor plant cooling system, intercooler, precoolers, recuperator, and turbomachine cavities. In addition, circumferential and radial ligaments are calculated for these cavities based upon pressure and stress considerations. The required heights through the core cavity, CAHE cavity, recuperator cavity, and precoolers cavity are calculated in order to determine the overall PCRIV height. In summary, the PCRIV sizing subroutine for CODER calculates the basic PCRIV layout for any given input values of cavity pressures and diameters established by the CODER optimization process.

10.2. CYCLE PARAMETERS AND PERFORMANCE

After completion of the programming changes to CODER for the intercooled cycle, plant performance calculations were made for the two-loop intercooled 3000-MW(t) alternate commercial plant with and without the warm liner. Table 10-1 presents the resulting data and includes data for the three-loop non-intercooled 3000-MW(t) reference plant for comparison and ease of reference purposes. The cycle data shown in Table 10-1 are not optimized, but rather are an initial set of parameters to be used in preliminary design-related activities. Tables 10-2 and 10-3 give the heat balances for the two-loop alternate commercial intercooled plants with and without the warm liner, respectively.

The process flow diagram for the primary coolant system of the two-loop alternate commercial intercooled plant with conventional liner is shown in Fig. 10-1. This figure shows the primary coolant system comprised of those subsystems and components required for transfer of energy from the reactor core to the gas turbine and secondary coolant systems. The major system components involved in this function are the gas turbine, precooler, intercooler, and recuperator, which comprise a single power conversion loop. The process flow diagram identifies the primary coolant temperatures, pressures, and flow conditions existing at various identified areas around the loop which affect each component and subsystem. The parameter conditions listed in Fig. 10-1 are considered as preliminary data for 100% nominal operating condition. These data will be updated as the plant design is developed. A tentative decision has been made to utilize the conventional water-cooled and insulated liner throughout the two-loop alternate plant primary system.

It can be seen by comparing the cycle data given in Table 10-1 with that given in Table 2-1 of Ref. 10-1 that there are only very small differences in the cycle parameters. Although the parameters included in the present study are not optimized with respect to cost, from the thermodynamic standpoint it is believed that the set of parameters previously chosen was not too far from optimum. This is because, in combination with the thermodynamic

TABLE 10-1
MAJOR CYCLE PARAMETERS AND PERFORMANCE RESULTS

Item	Alternate Two-Loop Commercial (Warm Liner)	Alternate Two-Loop Commercial (Conventional Liner)	Reference Three-Loop Commercial (Conventional Liner)
Cycle type	Intercooled	Intercooled	Intercooled
Reactor inlet temperature, °C (°F)	443.9 (831.1)	441.9 (827.4)	498.1 (928.5)
Reactor outlet temperature, °C (°F)	850.1 (1562.2)	850.1 (1562.2)	850.1 (1562.2)
Compressor pressure ratio (each)	1.75	1.75	---
Compressor pressure ratio (overall)	3.06	3.06	2.5
Minimum cycle helium temperature, °C (°F)	26.1 (79.0)	26.1 (79.0)	26.1 (79.0)
Low-pressure compressor inlet temperature, °C (°F)	28.2 (82.7)	26.8 (80.2)	---
High-pressure compressor inlet temperature, °C (°F)	26.8 (80.2)	26.7 (80.0)	26.7 (80.1)
High-pressure compressor exit pressure, MPa (psia)	7.874 (1142)	7.874 (1142)	7.929 (1150)
Low-pressure compressor pressure rise, MPa (psi)	1.969 (285.6)	19.64 (284.9)	---
High-pressure compressor pressure rise, MPa (psi)	3.343 (484.9)	3.374 (489.4)	4.757 (690.0)
Primary system ΔP , MPa (psi)	0.459 (66.6)	0.424 (61.5)	0.391 (56.7)
Recuperator effectiveness	0.898	0.898	0.898
Turbine isentropic efficiency, ^(a) %	92.2	92.2	91.8
High-pressure compressor isentropic efficiency, ^(a) %	90.2	90.2	89.8
Low-pressure compressor isentropic efficiency, ^(a) %	90.8	90.8	---
Generator efficiency, %	98.8	98.8	98.8
Turbine disk cooling flow, %	3.8	3.8	3.6
Compressor flow, kg/s (lbm/h)	747 (5,924,055)	743.1 (5,893,096)	572.2 (4,537,450)
Reactor rating, MW(t)	3000	3000	3000
Net electrical output, MW(e)	1249	1263	1187
Plant efficiency, %	41.63	42.09	39.55

^(a) Efficiency across bladed sections.

TABLE 10-2
ALTERNATE TWO-LOOP COMMERCIAL PLANT HEAT BALANCE (WARM LINER CASE)

	Plant Output (MW)	Primary System (MW)
Reactor power	--	3000
Primary system heat loss	--	(3.6)
Turbine gross power	2601.7	(2601.7)
Low-pressure compressor gross power	(659.3)	659.3
High-pressure compressor gross power	(659.3)	659.3
Heat rejected from precooler	--	(1037)
Heat rejected from intercooler	--	(676)
Bearing losses	(8.0)	--
Generator losses	(15.3)	--
Plant auxiliary power	(11.0)	--
Net plant output	1248.9	--
Primary system heat balance (error)	--	0.2

TABLE 10-3
ALTERNATE TWO-LOOP COMMERCIAL PLANT HEAT BALANCE
(CONVENTIONAL LINER CASE)

	Plant Output (MW)	Primary System (MW)
Reactor power	--	3000
Primary system heat loss	--	(15.6)
Turbine gross power	2605.7	(2605.7)
Low-pressure compressor gross power	(654.2)	654.2
High-pressure compressor gross power	(654.2)	654.2
Heat rejected from precooler	--	(1028.7)
Heat rejected from intercooler	--	(658.3)
Bearing losses	(8.1)	--
Generator losses	(15.5)	--
Plant auxiliary power	(11.0)	--
Net plant output	1262.6	--
Primary system heat balance (error)	--	0.2

- LEGEND**
1. MAIN LOOP COOLANT FLOW PER LOOP.
 2. VARIABLE ORIFICE REGION COOLANT CHANNELS.
 3. FIXED ORIFICE COOLANT CHANNELS.
 4. CONTROL ROD CHANNELS.
 5. CORE GAPS.
 6. SIDE REFLECTOR GAPS.
 7. VARIABLE ORIFICE REGION EXIT.
 8. FIXED ORIFICE COOLANT EXIT.
 9. CORE GAP COOLANT EXIT.
 10. SIDE REFLECTOR AND THERMAL BARRIER EXIT.
 - 11.12. AUXILIARY LOOP COOLANT FLOW PER LOOP.
 13. AUXILIARY LOOP BYPASS RETURN.
 14. AUXILIARY LOOP SHUTOFF VALVE LEAKAGE PER LOOP.
 15. FLOW TO HELIUM PURIFICATION SYSTEM.
 16. PURIFIED HELIUM RETURN FROM SERVICE SYSTEM.
 17. PURIFIED HELIUM RETURN THROUGH PRESSURE RELIEF TRAINS.
 18. PRIMARY BYPASS.
 19. CORE OUTLET.
 20. TURBINE INLET.
 21. TURBINE LEAKAGE.
 22. TURBINE COOLING.
 23. HELIUM PURIFICATION RETURN TO TURBINE.
 24. TURBINE OUTLET.
 25. ATTENUATION.
 26. MIXED PRIMARY BYPASS AND ATTENUATION.
 27. TRIM VALVE BYPASS.
 28. SAFETY VALVE BYPASS.
 29. CONTROL VALVE BYPASS.
 30. RECUPERATOR HOT LINE.
 31. RECUPERATOR INTERNAL BYPASS LEAKAGE.
 32. RECUPERATOR INTERNAL MAIN FLOW.
 33. RECUPERATOR COLD OUTLET (LOW PRESSURE).
 34. PRECOOLER INLET.
 35. PRECOOLER INTERNAL BYPASS LEAKAGE.
 36. PRECOOLER INTERNAL MAIN FLOW.
 37. PRECOOLER OUTLET.
 38. LOW PRESSURE COMPRESSOR INLET.
 39. LOW PRESSURE BYPASS LEAKAGE.
 40. LOW PRESSURE COMPRESSOR OUTLET.
 41. INTERCOOLER INLET.
 42. INTERCOOLER INTERNAL BYPASS LEAKAGE.
 43. INTERCOOLER INTERNAL MAIN FLOW.
 44. INTERCOOLER OUTLET.
 45. HIGH PRESSURE COMPRESSOR INLET.
 46. HIGH PRESSURE COMPRESSOR BYPASS LEAKAGE.
 47. HIGH PRESSURE COMPRESSOR OUTLET.
 48. RECUPERATOR INLET.
 49. RECUPERATOR OUTLET.
 50. PRIMARY FLOW BYPASS.

- REFERENCES**
1. Q23492 STANDARD INSTRUMENT & PIPING SYMBOLS.
 2. (LATER) P & I DIAGRAM, R.T.S. PRIMARY COOLING SYSTEM.
 3. (LATER) P & I DIAGRAM CORE AUXILIARY COOLING SYSTEM.
 4. (LATER) R.T.S. THERMAL PERFORMANCE REQUIREMENTS.

- NOTES**
1. VALUES GIVEN IN TABLE FOR THESE FLOW PATHS DO NOT INCLUDE THE FLOWS THAT RECYCLE TO THE SERVICE SYSTEMS WITHOUT ENTERING THE PRIMARY COOLANT.
 2. DUE TO ROUND OFF OF DATA, MASS FLOW IN A GIVEN LOOP MAY NOT EXACTLY BALANCE.
 3. PCRV FREE GAS VOLUME (LATER) FT³; NORMAL HELIUM INVENTORY (LATER) LBS.
 4. PRIMARY COOLANT FLOW DURING CACS OPERATION GIVEN PER REF. 3.
 5. FOR EXTREME ATTENUATION AND EXTREME HOT CONDITIONS SEE REF. 4.
 6. FLOW STREAM NOS 1 THRU 10 ARE GIVEN AS TOTAL PLANT FLOWS. REMAINING STREAM NOS 11 THRU 50 IDENTIFY FLOW PER LOOP.
 7. PCRV DESIGN INCORPORATES COLD LINER CONCEPT.

CASE	STREAM No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25
NORMAL PLANT OPERATING MODE	FLOW (LB/HR)	11243640	10186440		393330	325070	332310	3086740	303330	394220	106660									11243640	563000	23660	230380		585400	2890
ACFM LATER	PRESSURE (PSIA)	1124	1124		1124	1124	1124	1114	1114	1114	1114									1114	1108	1108	1142		395	660
TEMP (F°)		830	830		830	830	830	1568	1202	1463	1124									1562	1560	1560	232		902	232
STREAM No.		26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50
FLOW (LB/HR)		7890			5628	5062620	29460	583960	5862520	5872300	14730	5862520	5872350	5915740	23660	5915740	5872350	5862520	5872350	5915740	22340	5633340	5627150	5627150	5628	
ACFM LATER	PRESSURE (PSIA)					395	393	393	393	387	386	386	386	382	380	665	665	659	659	659	655	653	1142	1142	1133	395
TEMP (F°)						830	898	898	898	324	324	324	324	80	80	231	231	231	231	231	79	80	232	232	232	830

Fig. 10-1. Process flow diagram for primary coolant system, HTGR-GT alternate commercial plant

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efficiency studies, considerable information has been derived from the optimization work which was previously done for the non-intercooled plant. From this work it is felt that the major factors affecting the thermodynamics of the cycle, such as pressure ratio, equipment efficiency, and compressor inlet temperature, will not change significantly. Other possible perturbations such as liner heat losses and increases in the helium loop pressure drop are factors affecting the overall plant efficiency and not necessarily the plant optimization. However, these are refinements which are made as the design evolves.

REFERENCE

- 10-1. "Gas Turbine HTGR Program, Quarterly Progress Report for the Period Ending March 31, 1977," ERDA Report GA-A14381, General Atomic Company, April 1977.

11. ALTERNATE COMMERCIAL PLANT AUXILIARY SYSTEM DESIGN

During the reporting period efforts were devoted to developing conceptual-stage design documents for three of the ten auxiliary systems within the GA scope of responsibility, as listed below.

1. Auxiliary circulator service system
 - a. System Description
 - b. Process Flow diagram
 - c. Piping and Instrument diagram
2. Helium purification system
 - a. System Description
 - b. Process Flow diagram
 - c. Piping and Instrumentation diagram
3. Gas waste system
 - a. System Description
 - b. Process Flow diagram
 - c. Piping and Instrumentation diagram

These documents are conceptual in nature because design criteria have not been established for these three systems, nor for any of the seven other auxiliary systems. While development of these design criteria is a priority undertaking based upon lead-time requirements, it would have necessitated an effort well in excess of the available funding levels.

These conceptual designs are purposeful in that they enable development of a design approach for a system, which is an iterative process, and identification and correction of potential problem areas. For example, a conceptual regeneration section design and procedure has been developed for the helium purification system to satisfy the special requirements for low-temperature adsorber regeneration.

12. ALTERNATE COMMERCIAL PLANT REACTOR CORE DESIGN

A core arrangement study was undertaken to determine the best layout for the alternate commercial HTGR-GT core. The basic parameters were a thermal power level of 3000 MW(t), a power density of $<7.0 \text{ W/cm}^3$, and an eight-block-high core. In searching for an optimum layout within these criteria, an attempt was made to find a layout with the best possible core-reflector boundary. This means a core with a relatively smooth boundary with as few fixed orifice boundary regions as possible (i.e., one or two column fuel regions) to reduce power peaking at the core boundary. In addition, to the extent possible, every fuel region should have a control rod column to ensure adequate core shutdown margins with one or more struck rods.

The core layout selected, shown in Fig. 12-1, consists of 625 fuel columns arranged in 91 regions. At a power output of 3000 MW(t), the core power density is 6.5 W/cm^3 . The core boundary is relatively smooth and there are no fixed orifice regions.

Table 12-1 lists the major parameters for the commercial HTGR-GT core and compares them with the demonstration plant HTGR-GT design. A preliminary pressure drop estimate has also been made for this core layout, based on a maximum expected region peaking factor of 1.6 and the basic plant parameters. The estimated core pressure drop is 63.4 kPa (9.195 psi) in comparison with a corresponding value of 58.9 kPa (8.543 psi) for the demonstration plant HTGR-GT core. The breakdown of the overall core pressure drop into its various components, such as orifice pressure drop, active core pressure drop, reflector pressure drop, etc., is given in Table 12-2 together with equivalent demonstration plant HTGR-GT values for comparison.

Fig. 12-1. HTGR-GT commercial plant core layout

TABLE 12-1
CORE DESIGNS FOR HTGR-GT

	1530-MW(t) Demonstration Plant	3000-MW(t) Commercial Plant
Number of full regions	37	85
Number of five-column regions	12	6
Number of fixed orifice columns	12	0
Number of fuel columns	331	625
Power density, MW/m ³	6.5	6.8
Number of refueling penetrations	55	91
Number of control rod drives	49	91
Effective core diameter, m (ft)	6.92 (22.69)	9.51 (31.2)
Helium inlet temperature at core, °C (°F)	452.5 (846.6)	452.5 (846.6)
Helium temperature at reactor outlet, °C (°F)	850 (1562)	850 (1562)
Helium flow rate, kg/s (Mlbm/h)	741 (5.883)	1452 (11.535)
Power density, W/cm ³	6.5	6.8
Fixed orifices	Yes	No

TABLE 12-2
CORE PRESSURE DROP DATA FOR HTGR-GT^(a)

	Pressure Drop [kPa (psi)]	
	1530-MW(t) Demonstration Plant	3000-MW(t) Commercial Plant
Orifice	3.41 (0.495)	3.69 (0.535)
Top reflector	6.31 (0.916)	6.80 (0.987)
Active core	41.36 (6.000)	44.47 (6.451)
Bottom reflector	2.31 (0.335)*	2.49 (0.361)*
Core support	5.49 (0.797)*	5.49 (0.861)*
Total core	58.90 (8.543)	63.39 (9.195)

(a) Maximum pressure drop values expected based on a region power factor of 1.6. The pressure drop data should be used as a best estimate. In particular, data marked with an asterisk are subject to change.

13. REFERENCE AND ALTERNATE COMMERCIAL PLANT HEAT EXCHANGER DESIGN

13.1. INTRODUCTION

Commercial plant heat exchanger activities initiated during this report period addressed conceptual heat exchanger design for the two commercial plant variants currently under study, i.e., the three-loop non-intercooled 3000-MW(t) reference HTGR-GT commercial plant and an alternate two-loop intercooled 3000-MW(t) plant extrapolation of the single-loop demonstration plant. Five heat exchangers are involved in this effort: the recuperator and precooler for the reference plant and the recuperator, precooler, and intercooler for the alternate plant. These design efforts are complementary to the extent that common design approaches are being pursued for the recuperators (straight-tube counterflow modular) and for the helium-water units (helical bundle with outside diameter finned tubes). From a heat exchanger standpoint, the major differences between the two commercial plant variants being considered are (1) the influence of the difference in loop thermal ratings on the physical size of the recuperator, and (2) the influence of intercooling, which offsets the effect of the higher loop rating on the reject heat exchanger size by allowing the heat rejection to be shared between two helium-water heat exchangers that are comparable in size to the precooler in the reference plant design.

This conceptual design work is being performed by Combustion Engineering under subcontract to GA and covers the heat exchanger design work downstream of thermal sizing, which was performed by GA. Items to be delivered by Combustion Engineering include the following:

1. General arrangement drawings, including major design features and mechanical interface definition.
2. Materials lists.

3. Manufacturing sequence drawings.
4. Estimated costs.
5. Design description documents.
6. Final report covering all of the above.

These items are scheduled for completion in the next report period. Therefore, the following sections address the thermal sizing of these heat exchangers by GA to provide the basis for the Combustion Engineering effort currently in progress.

13.2. HTGR-GT THREE-LOOP REFERENCE PLANT HEAT EXCHANGER SIZING

New thermal size data were developed for the recuperator and precooler in the three-loop non-intercooled reference plant as part of a general effort to update this plant design. While the general heat exchanger requirements are very similar to those of the previous reference design study, resizing these units at this time is appropriate so that the updated designs can account for the following changes and modifications:

1. Incorporation of the integral return tube module approach in the recuperator to eliminate the licensing questions and complexity associated with the previous high-pressure center return duct approach.
2. Helical fin-tube precooler (in accordance with the findings of the HTGR-GT heat exchanger concept selection study).
3. Frontal area utilization (packaging) efficiencies consistent with Combustion Engineering preliminary mechanical design approaches developed for the one-loop intercooled plant studies.

4. Lower precooler water outlet temperature (from 168° to 132°C) (334° to 270°F), reflecting dry-cooled plant design.

The thermal design statistics resulting from this study are summarized in Table 13-1. A comparison of these results with the heat exchanger designs for the previous three-loop reference plant study shows that (1) the precooler diameter has remained essentially the same, (2) neither heat exchanger reflects a PCRV height problem at this time, and (3) more conservative assumptions regarding tube support parasitic loss and Combustion Engineering packaging procedures project a recuperator overall pressure loss of 2.8%, which is roughly 0.6% higher than the previous reference design when estimated on the basis of common boundaries.

The recuperator thermal size selection was taken from the parametric survey shown in Fig. 13-1 and represents the minimum pressure loss (2.8%) for the combination of 12.2 m (40 ft) maximum effective length and 5.33 m (17.5 ft) maximum heat exchanger outside diameter limits set by PCRV layout considerations. This will produce an updated recuperator design that should be roughly 0.3 m (1 ft) larger in diameter and 0.6% higher in pressure loss than its predecessor. However, this information is subject to the scaling assumptions used, and the ultimate recuperator design solution will be the product of an iterative process of interfacing between GA and Combustion Engineering. The mechanical design of this unit will draw heavily upon the concept and approaches being established for the one- and two-loop inter-cooled plant recuperators.

The parametric survey to establish the precooler thermal size is shown in Fig. 13-2. These results are based on design data that are comparable to those of previous precooler studies; however, the water outlet temperature was lowered from 168° to 132°C (334° to 270°F) and the water flow rate was adjusted correspondingly upward to reflect the precooler operating conditions associated with a HTGR-GT plant that has been optimized for dry cooling. The parametric results were generated with the GA computer code GT*SIZE, a variant of NUSIZE that has been modified to accommodate helical fin-tube geometries.

TABLE 13-1
THERMAL DESIGN SUMMARY FOR HTGR-GT THREE-LOOP REFERENCE PLANT HEAT EXCHANGERS (HX)

	Recuperator		Precooler	
Number per plant	3		3	
Loop rating, MW(t)	1000		1000	
Fluid circuit	HP He	LP He	Helium	Water
Fluid routing	Tube side	Shell side	Shell side	Tube side
Flow, kg/s (lb/h)	545.5 (4.33×10^6)	566.4 (4.49×10^6)	567.9 (4.508×10^6)	933.2 (7.408×10^6)
Inlet temperature, °C (°F)	174.5 (346.1)	534.8 (994.7)	223 (433.5)	20.6 (69)
Inlet pressure, MPa (psia)	7.85 (1138.5)	3.25 (472.2)	3.21 (465.3)	2.07 (300)
Outlet temperature, °C (°F)	498.1 (928.6)	223.3 (434)	26.1 (79)	132.2 (270)
Pressure loss	1.42%	1.40%	~0.75%	0.69 MPa (1000 psi)
Overall $\epsilon(\Delta P/P_1)$, %	2.82		~0.75	
Effectiveness	0.898		0.972	
LMTD, °C (°F)	42.5 (76.5)		30.5 (54.9)	
UA/HX required, mW/°C (Btu/hr-°F)	21.6 (40.97×10^6)		19.1 (36.14×10^6)	
UA margin, %	10		10	
Q, MW/HX	918		581	
HX configuration	Counterflow		Multipass cross counterflow	
Construction	Straight-tube modular		Helical fin-tube	
Tubes per HX	66,483		827	
Tube outside diameter x wall, mm (in.)	11.1 x 1.1 (0.4375 x 0.045)		28.6 x 2.87 (1.125 x 0.113)	
Effective tube length, m (ft)	12.2 (40)		105.1 (345)	
Fin height x thickness, mm (in.)	---		1.59 x 0.51 (0.0625 x 0.020)	
Fin count, fins/mm (fins/in.)	---		0.63 (16)	
Effective bundle height, m (ft)	12.2 (40)		12.5 (41)	
Net HX frontal area, m ² /HX (ft ² /HX)	13.94 (150)		13.94 (150)	
Tube pitch arrangement	Equilateral triangle		In-line rectangular	
Tube pitch specification	p/d = 1.40		Transverse and longitudinal pitch = 45.7 mm (1.80 in.)	
Approximate bundle outside diameter, m (ft)	5.33 (17.5)		4.57 (15)	
HX diameter envelope, ^(a) m (ft)	5.64 (18.5)		4.88 (16)	
HX height envelope, m (ft)	22.4 (73.5)		22.4 (73.5)	

(a) Diameter inside PCRV liner thermal barrier.

Fig. 13-1. Recuperator thermal size study for three-loop HTGR-GT reference plant

HELIUM-SIDE
PRESSURE LOSS,
psi (MPa)

	S_T in. (m)	n fins/in. (fins/mm)
X	1.80 (15.7)	A 19 (0.75)
Y	2.20 (55.9)	B 16 (0.63)
Z	2.60 (66)	C 11 (0.43)
		D 6 (0.24)

Fig. 13-2. Precooler thermal size study for three-loop HTGR-GT reference plant

The basic fin-tube geometry considered is generally similar to existing hardware and results in precooler helical bundle statistics (number of tubes and bundle height) that are amenable to fabrication and maintenance.

Figure 13-2 explores the influence that changes in precooler diameter, fin count, and transverse tube pitch have upon the height and helium pressure loss of the precooler. To be compatible with the PCRV envelope and system performance objectives, the results, which are displayed as "islands" of constant precooler diameter, must fall inside the parametric window defined by the allowable helium pressure loss and bundle height. The thermal size selection indicated in Fig. 13-2 is based on the same fin-tube surface geometry as that of the single-loop intercooled demonstration plant helium-water heat exchangers (logistics and DV&S advantage); it has the maximum permissible bundle height (uses up the available envelope) and has a helium pressure loss that falls well within the allowable limit (considerable allocation available for analytical uncertainty adjustments and parasitic losses). The water-side pressure loss of 0.69 MPa (100 psid), which prevailed for all cases investigated, is the current ceiling established by plant efficiency considerations. Relevant design statistics are summarized in Table 13-1.

The follow-on Combustion Engineering mechanical design effort to this precooler study is expected to draw upon the work already in process for the intercooled plant heat exchangers. With the precooler situated on top of the turbomachine, side penetrations above and below the unit for the water boxes are envisaged, and the unusable frontal area in the center of the helical bundle will be set primarily by fabrication considerations. One of the main challenges to Combustion Engineering will be to ensure that the precooler does not govern PCRV height.

13.3. ALTERNATE TWO-LOOP COMMERCIAL PLANT HEAT EXCHANGER STUDIES

The conceptual design of the recuperator, precooler, and intercooler for the alternate two-loop commercial plant variant draws from the work being performed by Combustion Engineering on the single-loop demonstration

plant HTGR-GT heat exchangers. The only significant difference between the two-loop plant heat exchangers and their counterparts in the single-loop demonstration plant is the requirement to convert the designs from the warm liner approach (demonstration plant) to the more conventional cold liner approach (two-loop plant). From the standpoint of the three heat exchangers in question, this essentially means that there is no need to create a cooling flow annulus between each heat exchanger and its PCRV cavity in the two-loop plant, thus eliminating most mechanical cross-duct connections to the heat exchanger and simplifying outer shroud requirements. These changes are minor enough to permit using the demonstration plant heat exchanger design to illustrate the general features of the two-loop heat exchangers.

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