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ABSTRACT

This report describes conceptual design and analysis performed by General Atomic Company for the U.S. Department of Energy on the direct cycle gas turbine high-temperature gas-cooled reactor (GT-HTGR). Three GT-HTGR plant concepts were studied:

1200-MW(e) three-loop non-intercooled cycle,
1200-MW(e) two-loop intercooled cycle,
600-MW(e) one-loop intercooled cycle (demonstration plant).

General Atomic co-operated with the German/Swiss HHT project on the development of the GT-HTGR concept in the areas of systems analysis, safety and accident analysis, and PCRV-liner-internals design.

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1. INTRODUCTION

This report describes technical work performed by General Atomic Company (GA) under the direct cycle gas turbine high-temperature gas-cooled reactor (GT-HTGR) program, Department of Energy (DOE) Contract EY-76-C-03-0167,* Project Agreement No. 46, for the period October 1, 1978, through March 31, 1979. Per agreement with DOE, this is the first of a series of semiannual reports which replaces the quarterly reporting system. Following the report summary in Section 2, technical progress is presented in Sections 3 through 21 in the numerical sequence of the Contract Work Breakdown Structure (CWBS) (see Contents) for the GT-HTGR specific work. This CWBS is a portion of that used for identification of work elements in the total National HTGR Program. Project Agreement No. 46 scope includes major program elements involving the GT-HTGR systems and component design and program management.

The major activities during the first half of FY-79 involved further concept definition of the commercial version of the GT-HTGR power plant, a re-definition of the GT-HTGR Program as a result of the steam cycle HTGR (SC-HTGR) phase-out decision, and cooperation with the German-Swiss HHT project in the definition of the HHT demonstration plant.

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*Effective May 1, 1979, this contract number was changed to DE-AT03-76SF70046.

2. SUMMARY

Concept studies for the reference three-loop 1200-MW(e) GT-HTGR plant were completed, and on this basis an updated plant economic evaluation is under way. An alternate 1200-MW(e) plant concept having two loops and an intercooled compressor was completed for the purpose of determining the differences due to intercooling as compared with the non-intercooled reference concept. This alternate two-loop plant embodies power conversion loops (PCLs) similar in size and cycle to that selected by the German/Swiss HHT project for the HHT demonstration plant. The feature of compressor intercooling improves plant performance at the expense of additional plant complexity due to adding the intercooler.

An associated activity was the completion of the conceptual design of a one-loop 1500-MW(t) intercooled plant intended as a backup design for the HHT demonstration plant project. This design differs from the HHT concept primarily in that it features a prismatic core, a single heat exchanger train, and a significantly smaller prestressed concrete reactor vessel (PCRV). The design has been forwarded to HHT to supplement its investigations.

Co-operation with the HHT project in the areas of systems analysis, safety and accident analysis, and PCRV-liner-internals design has produced a substantial exchange of relevant technical information which has been beneficial to both projects. The majority of the work by GA was directed toward investigation of critical design issues. One issue in particular is the "warm liner concept," which allows PCRV liner inspection/repair through elimination of the thermal barrier on the liner. This inspection/repair feature is desired by German utilities. GA analysis to date has indicated a number of safety and design problems which neither GA nor the HHT project has solved to date. The GA project has recommended to HHT

that an alternate means of meeting the German utility requirement be developed: as a first goal a leak detection-collection system and secondly a removable thermal barrier.

Progress in the systems area included transient analysis of the reference and alternate 3000-MW(t) plants. These transients are being used to establish requirements for the system and component designs. Other progress involved the upgrading of the systems optimization code CODER, the modification of the core auxiliary cooling system (CACS) code RECA to accommodate the GT-HTGR, and the re-determination of the performance of the various plant configurations under investigation. The performance of the reference 3000-MW(t) commercial plant was confirmed at 39.54%.

In the safety/accident analysis and licensing areas, the design basis depressurization accident (DBDA) of turbine deblading was analyzed, HHT safety criteria were critiqued, and GT-HTGR licensing requirements were addressed. Pressure transients from turbine deblading in the area of the core outlet plenum were calculated in the range of 5.5 to 24.1 MPa/s (800 to 3500 psi/sec) with 17.2 MPa/s (2500 psi/sec) recommended as a design basis for the thermal barrier in the core outlet plenum.

In the structures area, the PCRV design requirement for the reference plant was established taking into account the multi-pressure conditions in the cavities, the effect of differential pressure on inner ligaments, and the proof-test pressure specification. The effect of pressure relief settings on the PCRV cavity pressure was also addressed. Analysis of the HHT demonstration plant PCRV bottom head showed that a sufficient number of tendons to produce the required prestressing could not be provided. An alternate scheme was proposed which incorporates sufficient horizontal straight and circumferential tendons in the bottom head to resist the turbine and heat exchanger cavity pressures. Also, for the HHT demonstration plant investigations, seismic analysis was initiated. In support of the reference and alternate commercial plant efforts, PCRV, liner, and thermal barrier conceptual designs were supplied.

Conceptual designs for two-bearing 400-, 500-, and 620-MW(e) turbomachines were prepared by United Technologies Power Systems Division (UTC). Double labyrinth buffer valves have been included to preclude ingress of lubricating oil into the primary helium. The performance rewards of increasing the turbomachine outer case diameter were estimated. Sound power levels at the compressor and turbine inlet and exits were estimated, and potential methods of attenuation were identified. Critical speed analysis was performed for both the 400- and 620-MW(e) configurations to ensure resonance-free normal operation. An approach to remote turbomachine maintenance was identified.

Heat exchanger design work by Combustion Engineering (CE) to establish manufacturability of the components led to identification by CE and GA of areas where further design work is warranted to establish design feasibility. The designs studied covered the heat exchangers for the demonstration plant and the reference and alternate commercial plants.

3. SYSTEM DESIGN METHODS (261002)

3.1. SCOPE

The purpose of this task is to:

1. Modify CACS design, performance, and safety analysis computer programs for application to the GT-HTGR.

2. Update the cost and design optimization code CODER for use with the three-loop reference plant design and the two-loop non-intercooled plant design.

3.2. SUMMARY

3.2.1. Computer Program RECA-GT*

The essential changes to RECA are provisions to receive data from REALLY and to account for backflow through GT-HTGR main loops. These changes have been incorporated into the program, but checkout and informal documentation are not complete.

3.2.2. Computer Program ECSEL-GT**

Work has been done on suboptimization within subroutine ECSEL-GT, which calculates the core auxiliary cooling water system (CACWS) air blast heat exchanger. However, no revisions have yet been made.

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*The primary core cooling evaluation code which is used as the basis for primary system limit evaluations and licensing.

**The sizing code which balances the conflicting system requirements to obtain the component sizes using costs as the balancing function.

3.2.3. Computer Program ECSTRA-GT

The capability to model core auxiliary heat exchanger (CAHE) leaks is being added to ECSTRA. Work is beginning under this subtask to produce a utility program to translate ECSEL-GT output to ECSTRA input.

3.2.4. Computer Program CODER

In support of evaluations required for the Primary System Parameter Review Study, CODER has been upgraded to more adequately represent current design of the 3000-MW(t), three-loop non-intercooled GT-HTGR with a delta prestressed concrete reactor vessel (PCRV) layout and conventional liner. This program version is named CODER6. The CODER6 program is now operational and incorporates the modeling changes and additions briefly reviewed below. More detailed documentation of the changes to CODER will be included in a design report specifically established for recording and documenting changes to CODER.

It should be emphasized that only the fuel, thermal barrier, and precooler/recuperator costs reflect present-day cost estimating, while all other costs are 1975 costs escalated to 1979 dollars to account for inflation. Revision of all other equipment cost estimates cannot be incorporated until June 1979. Nonetheless, it is felt that cost trends resulting from parameter variations will adequately represent first order effects and provide guidance on how parameter changes impact overall system costs and performance.

CODER6 can presently accommodate variations in system temperatures, pressures, and mass flow rates up to approximately ±10%, which is adequate for the near-term evaluation measurements. To expand the parametric range beyond ±10%, additional modifications to the PCRV, recuperator, and precooler models may be required.

The modifications to CODER incorporated in CODER6 are as follows:

1. A new delta PCRV model has been installed and includes reprogramming of the following aspects: PCRV elevation view, PCRV plan view, turbomachine sizing, duct lengths, thermal barrier areas, helium inventory, concrete volume, liner weight, and tendon requirements.

2. Duct pressure drop algorithms have been revised for the three-loop configuration. $\Delta P/P$ values are current estimates and include turbomachine loss updates.

3. The recuperator pressure drop model and packing efficiency have been revised to reflect the design change from a central return duct to an integral return tube configuration.

4. The thermal barrier heat load and gas differential temperature algorithms have been upgraded to incorporate 34 thermal barrier zones instead of 25 and include a more rigorous CODER calculation for heat load.

5. A medium-enriched uranium (MEU) fission product release model has been incorporated and calculates Cs-134 and Cs-137 as well as Ag-110m curie releases. An overall fissile particle fraction is also computed.

6. MEU fuel cycle cost algorithms have been incorporated.

7. The helical finned tube precooler model has been included.

8. All design data for the three-loop configuration have been revised to reflect present-day knowledge, and the base case for CODER6 is therefore now current.

9. Cost data for the reference design have been upgraded for the recuperator and precooler components to CE estimates. Fuel-related and thermal barrier costs are also 1979 estimates. All other costs are 1975 costs escalated to current dollars.

10. A plot routine and automated parameter perturbation routine have been incorporated.

4. SYSTEM DYNAMICS METHODS DEVELOPMENT (261003)

4.1. SCOPE

The purpose of this task is to develop system dynamics models in the REALY2 code for the GT-HTGR reference commercial plant.

4.2. SUMMARY

The reference commercial plant transient physical system model has been completed along with modifications to the REALY2 code to improve output and resolve a number of small operational difficulties encountered in FY-78 analyses. Modeling of the initial operations control and plant protection system (PPS) functions has been completed and documented to form the basis for analysis of the component limiting transients of System Dynamics Task 1003 (see Section 13). Effort has been initiated on development of a secondary system model.

4.3. DISCUSSION

Some of the changes to the REALY2 code in connection with development of a GT-HTGR reference commercial model are discussed briefly below:

1. The power distribution in the core was changed to the expected power distribution for the 625-column core which has been used in core layout drawings for the reference commercial 3000-MW(t) design.

2. The number of fuel rod columns has been corrected; 126,082 is the proper number of fuel rod columns for the 625-column core.

3. The design point reference gas flow rate for the CORE subroutine was changed to the steady-state core flow rate which is used in the reference commercial model.

4. The section of the CORE subroutine for calculation of approximate starting temperatures was deleted because it was a non-operational part of the subroutine and is not needed.

5. An option now exists which allows selection of either an exponential or a linear area curve for either the primary bypass or attemperation valve. The changes were made in HIFLO.

6. The warm liner options are accounted for by modifications in the DUCT subroutine.

7. A problem called "compressor turbining" which occurred in intercooled plant models and restricted the REALY2 predictions of emergency shutdown transients to a very brief period has been eliminated by extending the compressor map data.

8. An improved model of the circulating water system (CWS) to support the PPS and control system design has been initiated to simulate precooler leak detection and isolation and also CWS transients such as pump or cooling tower shutdown.

9. The REALY2 model (catalogued file REALY2*1) has been updated to permit the integration of the GT-HTGR configuration variants (intercooled, non-intercooled, and split-shaft) and the three current preliminary plant designs [3000-MW(t) reference commercial, 3000-MW(t) alternate commercial, and 1530-MW(t) warm liner demonstration] into a single computer program.

5. ALTERNATE DESIGN (630101)

5.1. SCOPE

The purpose of this task is to document one-, two-, and three-loop plant studies conducted during FY-78 and the first quarter of FY-79.

5.2. SUMMARY

During FY-78, work in the GT-HTGR program was directed at two major areas:

1. Development of a 1530-MW(e), one-loop demonstration plant design which would be the U.S. version of the HHT demonstration plant. The features required for follow-on U.S. commercial plants were incorporated in this design. It was intended that this design effort would influence the HHT configuration so that the HHT plant could provide a maximum of information for the U.S. and European commercial plants. Based on meetings with HHT later in FY-78, it was agreed that the design should be documented and act as a backup to the HHT design, providing critical data related to the evaluation of the "warm liner" concept.

2. Development of a two-loop intercooled and a three-loop non-intercooled commercial plant conceptual design. The two-loop design was to maximize the utilization of the demonstration plant data since it utilized two 620-MW(e) intercooled power conversion loops (PCLs). The three-loop non-intercooled design effort represented an update of the existing reference design. A selection was to be made between the two-loop and three-loop designs related to risk in proceeding from the demonstration

plant to the commercial plant, utility requirements, cost, maintainability, safety, etc. The two-loop and three-loop efforts were completed and documented, but the selection of the commercial plant configuration was never made because of the redirection of the National HTGR Program to adopt the GT-HTGR as the prime concept. This selection has now been rescheduled to the end of FY-80 and will cover more than the two plants which were previously studied.

5.3. DISCUSSION

5.3.1. U.S. Version of HHT Demonstration Plant Configuration

In FY-78, a study was performed in which 20 plant configurations were evaluated consistent with the project ground rule for all of the configurations to embody the "warm liner" feature in the core cavity to facilitate liner inspection.

Essentially two families of plant variants evolved from the study: (1) configurations which involved eliminating the thermal barrier to the highest degree possible, and (2) configurations in which the core cavity warm liner feature was retained, but conventional water-cooled and insulated liners were featured in the major heat exchanger and turbomachine cavities.

While it is recognized that evaluating and rating a family of plant variants is difficult, some form of comparative selection criteria was necessary to identify the candidate plant for specialist design attention. All the configurations were evaluated on the basis of a simplified and non-weighted rating system which included the following:

1. Primary system gas flow path complexity (strongly influenced by major cavity thermal barrier/warm liner requirements).

2. Utilization of one or two heat exchangers (particularly the recuperator) per PCL.

3. Adaptability of configuration to a two-loop commercial plant.

4. Maintenance and in-service inspection (ISI) considerations.

5. Plant cost (based on intuitive feeling, since cost data were not generated).

6. Safety and licensing considerations.

7. Feasibility issues.

8. Attractiveness as a commercial plant [i.e., prestressed concrete reactor vessel (PCRV) diameter minimization].

The screening and evaluation process led to the following decisions with regard to plant concept selection:

1. It was felt that the role of the demonstration plant was to not only prove the direct cycle concept (performance and integrity) but also to verify the actual components that would go into the first commercial plant. Accordingly, a concept was selected embodying full-size [1530 MW(t) loop rating] heat exchangers.

2. While many major problems exist with the warm liner approach, it was agreed that for the first time GA would pursue a design concept in which the thermal barrier was eliminated to the highest degree possible; i.e., all the major cavities would be swept with fairly low-temperature gas.

Upon selection of the demonstration plant concept, specialist tasks were initiated in the following areas: (1) PCRV structural design (GA), (2) reactor internals design (GA), (3) turbomachine design [United Technologies (UTC)], and (4) heat exchanger sizing and design (GA and

CE). Toward the end of the 1-yr study, technical inputs from these areas were factored into the generation of a plant layout drawing, which is shown in Figs. 5-1 through 5-4. Figure 5-5 is a schematic of the warm liner flow path. Major plant design parameters are shown in Table 5-1. The main features of the demonstration plant are given in Table 5-2, and a description of the plant primary system is given below.

As shown in the plan view of Fig. 5-1, the goal of minimizing the PCRV diameter was realized by offsetting the core cavity some 2.6 m (8.5 ft). The chordal orientation of the turbomachine is compatible with the two-loop commercial plant arrangement. The single, very large recuperator cavity is positioned over the turbomachine at the turbine discharge end of the cavity. The plant layout was strongly influenced by the decision (based on inputs from UTC) to utilize a single core-to-turbine duct. Growth potential from the 620-MW(e) turbomachine was not a factor in the demonstration plant study, but it should be pointed out that for machines much above this rating the increased mass flow rate would probably necessitate double turbine inlet ducts. This in turn would essentially mandate dual turbine exits and two recuperator trains, and the result would be a layout with features similar to the European demonstration plant.

As seen in Fig. 5-1, both of the helium-to-water heat exchangers are positioned to the side of (not over) the turbomachine cavity. To comply with the German requirements, four CACS units, each of 100% capacity, are incorporated in the primary system.

Referring to Figs. 5-1 through 5-5, the primary system helium gas flow paths are as follows. After exiting from the core bottom plenum, the 850°C (1562°F) gas is transported in a coaxial duct down to the turbomachine turbine inlet plane. After expansion in the turbine, the 494°C (921°F) helium leaves the turbomachine cavity and flows vertically upward into the low-pressure side of the straight tube modular recuperator. With the low-pressure gas flowing outside the tubes, there is a regenerative heat transfer (low-pressure to high-pressure gas) of about 1250 MW(t) in this exchanger. The gas exits radially from the top of the modular assembly at 162°C (324°F)

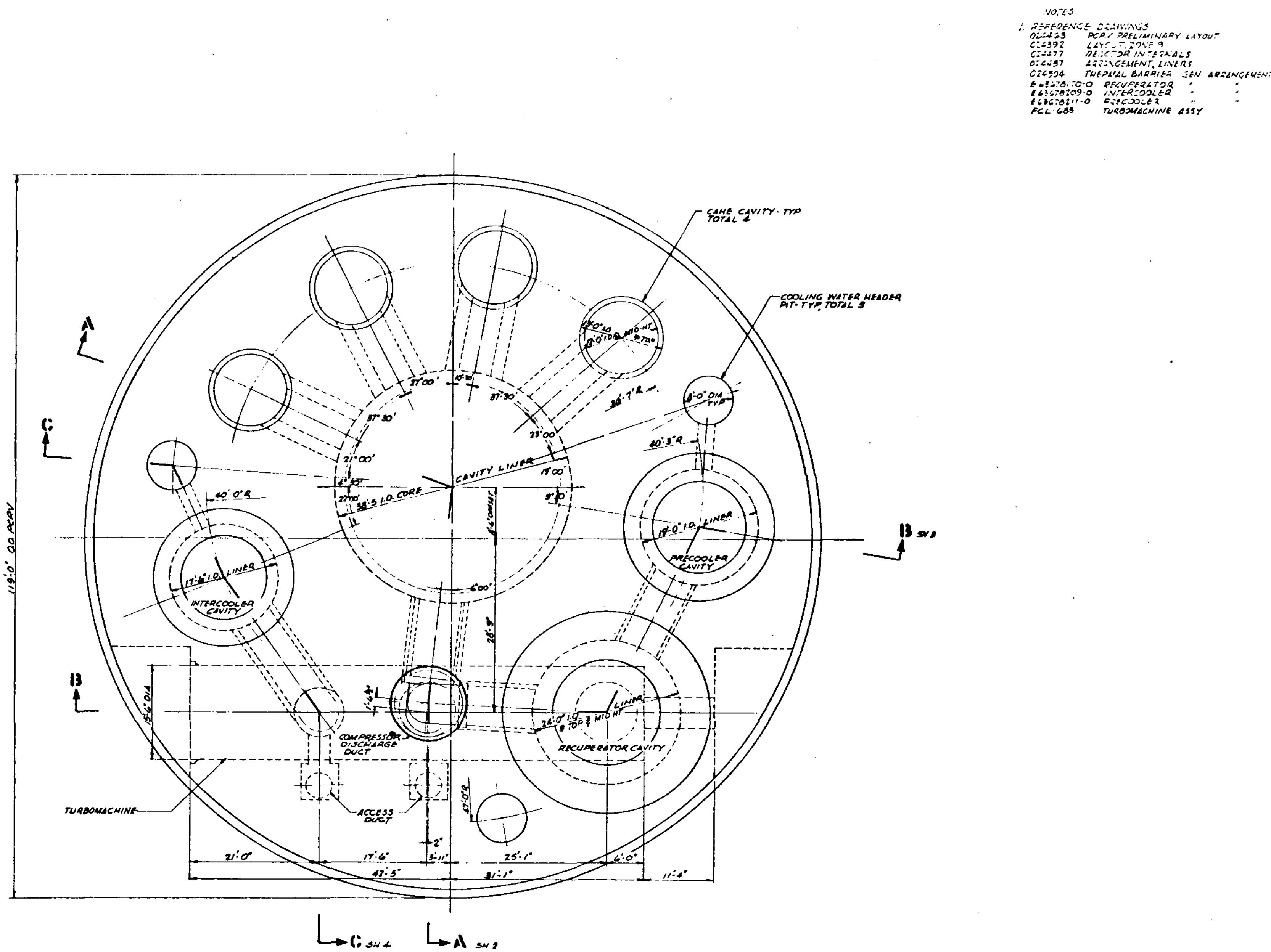

Fig. 5-1. Plan view of PCRV top head for 1530-MW(t), one-loop GT-HTGR demonstration plant

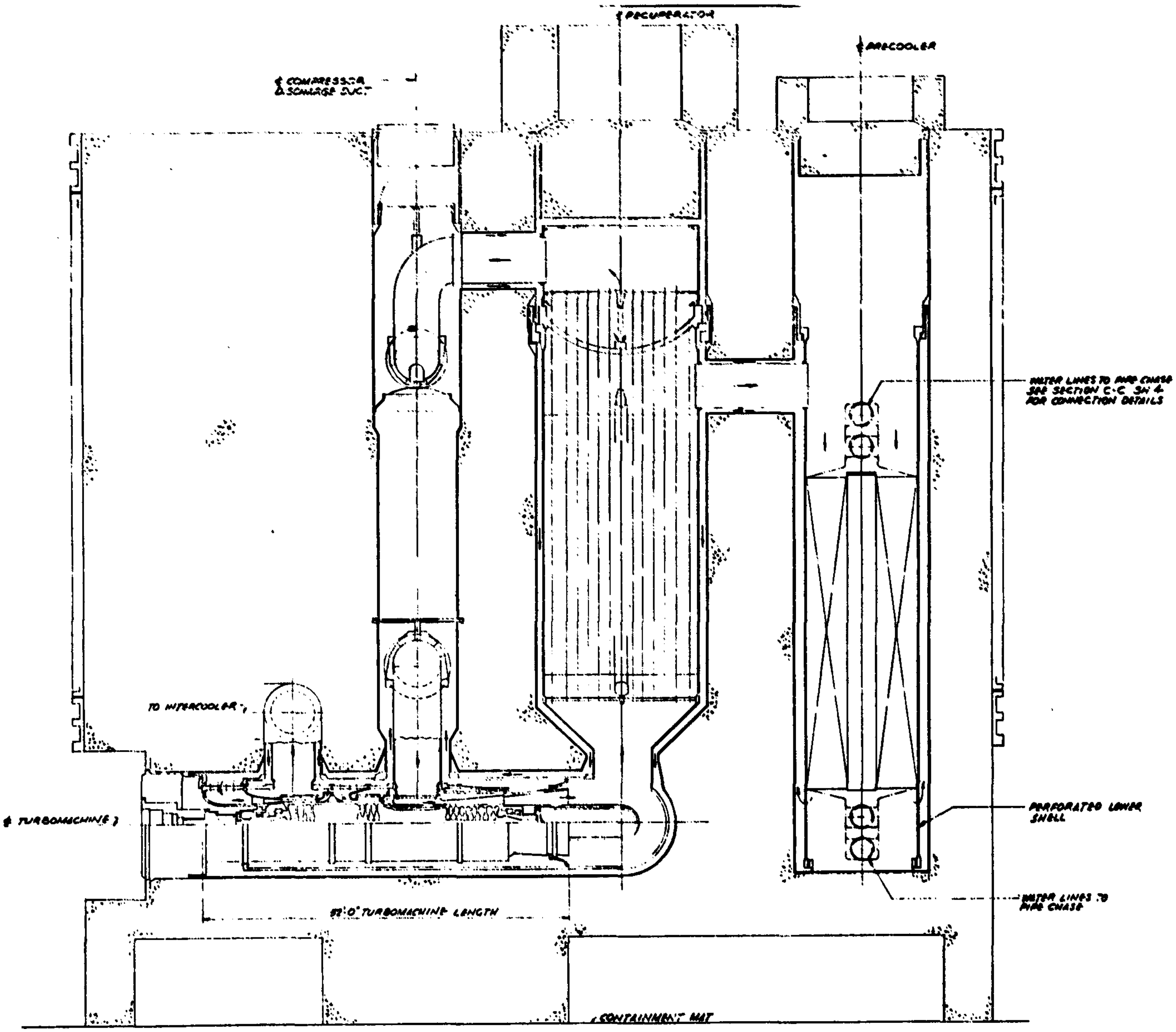

Fig. 5-2. Section B-B of Fig. 5-1

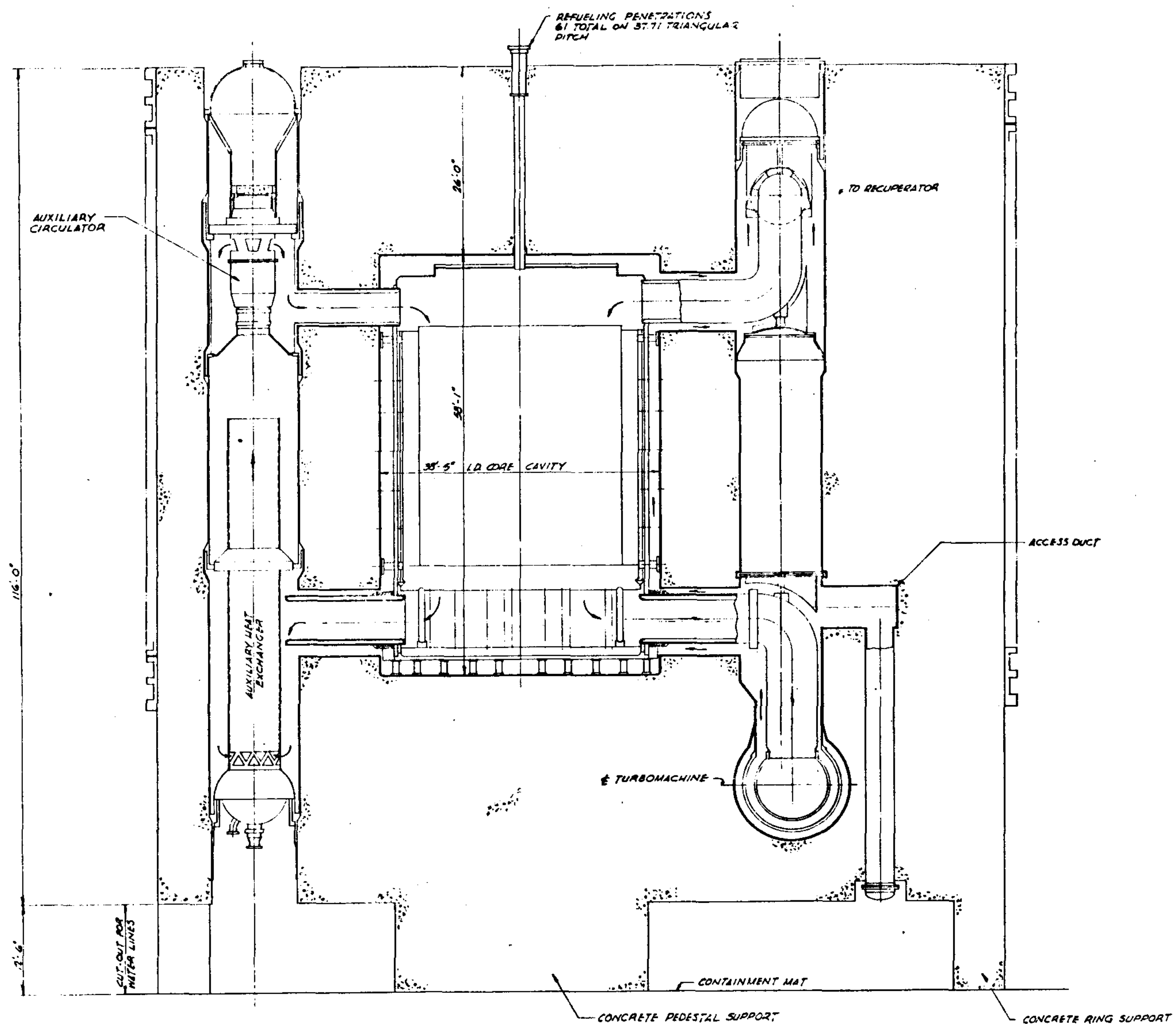

Fig. 5-3. Section A-A of Fig. 5-1

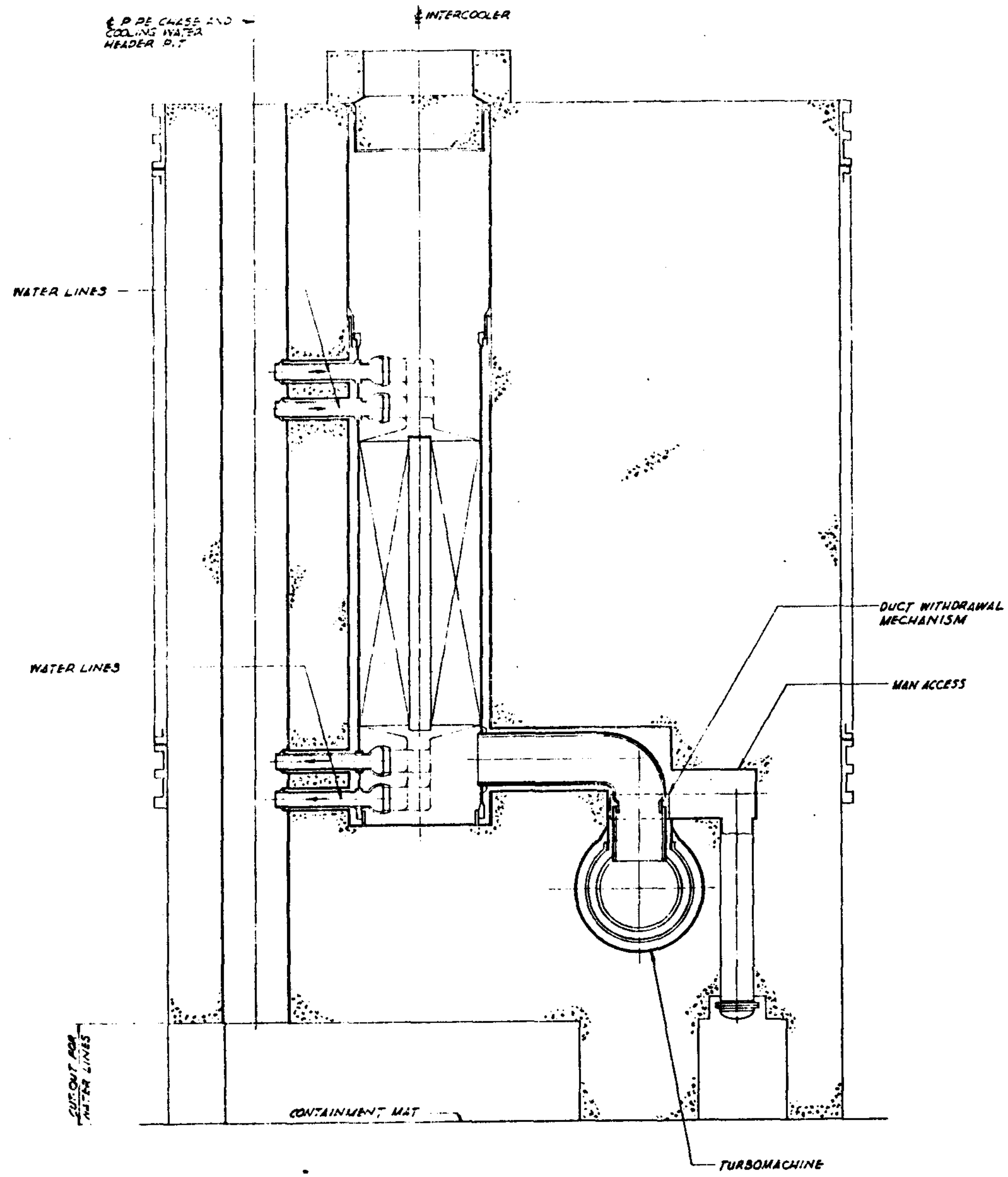

Fig. 5-4. Section C-C of Fig. 5-1

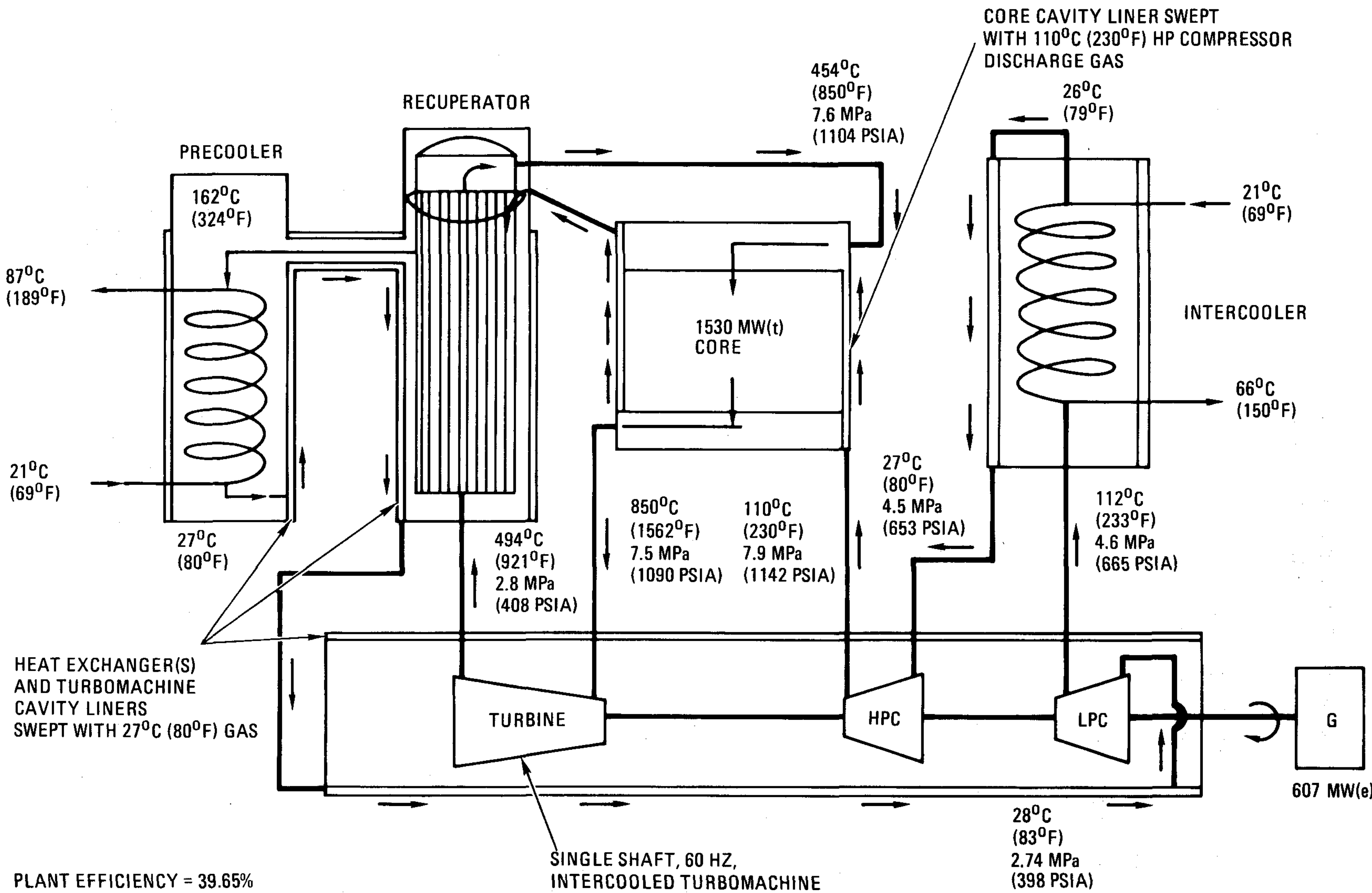

Fig. 5-5. Flow path diagram for GT-HTGR demonstration plant with warm liner

TABLE 5-1
MAJOR DESIGN PARAMETERS FOR 1530-MW(t), ONE-LOOP
GT-HTGR DEMONSTRATION PLANT

Turbine inlet temperature	850°C (1562°F)
Ambient air temperature	15°C (59°F)
Thermodynamic cycle	Intercooled
Heat rejection mode	Dry cooling
Liner type	Warm liner
Compressor pressure ratio	3.0
Compressor inlet temperature	LP - 28.1°C (82.6°F) HP - 26.7°C (80.1°F)
Maximum system pressure	7.87 MPa (1142 psia)
Overall system pressure loss, Σ ($\Delta P/P$)	14.54%
Compressor flow	778 kg/s (6.173×10^6 lb/hr)
Recuperator effectiveness	0.898
Turbine isentropic efficiency, across blading	92.2%
Compressor adiabatic efficiency, across blading	LP - 90.8% HP - 90.2%
Generator efficiency	98.8%
Turbine cooling flow, discs	3.6%
Precooler water outlet temperature	87°C (188.5°F)
Intercooler water outlet temperature	65.6°C (150°F)
CACS parasitic heat loss	1.8 MW(t)
Primary system heat loss	7.0 MW(t)
Auxiliary power	8.0 MW(t)
Station efficiency	39.65%[a]
Net electrical power output	607 MW(e)[b]

[a] Parameter selected on thermodynamic basis; plant not optimized for minimum power generating cost.

[b] Initial rating of the turbomachine was 620 MW(e). However, the higher than projected primary system pressure losses associated with the final plant layout resulted in a loss of station efficiency.

TABLE 5-2
MAIN FEATURES OF 1530-MW(t) DEMONSTRATION PLANT

Integrated, Direct Cycle Plant

Prismatic Core, Thermal Rating 1530 MW(t)

MEU Fuel (3-yr Fuel Cycle)

Reactor Core Power Density 6.5 W/cc

Intercooled Cycle with High Degree of Regeneration

$$\left.\begin{array}{lcl} P_{max} & = & 7.93 \text{ MPa } (1150 \text{ psia}) \\ T_{max} & = & 850°\text{C } (1562°\text{F}) \\ R_{comp} & = & 3.0 \\ E_{recup} & = & 0.90 \end{array}\right\} \text{Plant Efficiency} = 39.65\%^{(a)}$$

Turbomachine Rating 607 MW(e) [620 MW(e) Design]

Warm Liner in Core Cavity [Swept with 110°C (230°F) Gas]

HX and Turbomachine Cavities Swept with 26.7°C (80°F) Gas

PCRV Details

Offset Core Cavity

Diameter 36.3 m (119 ft)

Height 35.4 m (116 ft)

Chordal Turbomachine Position

CACS - 4 x 100% Units (FRG Requirement)

Two-Bearing Turbomachine (Single Turbine Inlet Duct)

Man Access Provision to Bearing Cavity Areas

Straight Tube, Modular Recuperator

Helical Bundle Precooler and Intercooler

Dry-Cooled Plant

Major PCL Components (Turbomachine and Heat Exchangers) Representative for Two-Loop Commercial Plant

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(a) Low cycle efficiency of the intercooled plant is a reflection of the gas flow path complexity associated with compliance of the warm liner feature.

and enters the precooler via a short horizontal coaxial duct. With all the useful thermal energy extracted from the gas stream, the cycle reject heat is dissipated in the precooler where the primary system helium flowing outside the tubes is reduced in temperature to 26.7°C (80°F). The circulating water in the helical, finned-tube precooler, which is pressurized to avoid phase change, is increased in temperature from 20.6°C (69°F) to 87.2°C (189°F), the reject heat transfer rate from the primary system being on the order of 533 MW(t).

Unlike previous plant arrangements, where the cold helium is transported directly from the precooler to the compressor, the 26.7°C (80°F) gas in the demonstration plant is used for liner cooling. As shown in Fig. 5-2, an annular gas flow path is formed between the precooler shroud and the cavity liner. The full primary system helium mass flow [at 26.7°C (80°F) and 2.76 MPa (400 psia)] leaves the bottom of the precooler assembly and flows vertically upward in the annular passage, effectively keeping the liner (and concrete) at an acceptable temperature level without the need for a thermal barrier on the precooler cavity liner.

Leaving the precooler cavity just above the exchanger bundle, the 26.7°C (80°F) gas is transported via a short horizontal duct to the recuperator cavity. In a manner identical to that described above, the cold gas flows downward in an annular passage in the recuperator cavity to cool the liner. As shown in Fig. 5-2, there is a transition from the recuperator to the turbomachine cavities to enable this cooling gas to enter a horizontal annulus passage in the turbomachine cavity. A substantial steel shield of cylindrical geometry is built into the turbomachine cavity. This cylinder forms the interface between the vessel and turbomachine. The cold gas flows axially through this annular passage and via holes in the turbomachine casing enters the low-pressure compressor. Because of the large "wetted" areas of shrouds and liners between the precooler outlet and low-pressure compressor inlet, some regenerative heat transfer has taken place, and it has been estimated that the helium enters the compressor at a temperature of 28.3°C (83°F).

After compression in the eight-stage low-pressure compressor [to 4.59 MPa (665 psia)] the 112°C (233°F) gas leaves the top of the turbomachine cavity and flows to the intercooler in a short coaxial duct as shown in Fig. 5-4. The intermediate pressure gas flows upward through the finned-tube, helical intercooler bundle, giving up its compression heat of around 337 MW(t). The temperature of the water (single phase) following through the intercooler is increased from 20.6°C (69°F) to 65.6°C (150°F).

The requirement for a two-journal-bearing turbomachine (for man access inspection and maintenance) necessitates a short distance between the low-pressure and high-pressure compressors to minimize the bearing span and hence give acceptable critical speed margins. To accomplish this, a concentric inlet (outlet) duct as shown in Fig. 5-3 is necessary. Thus, the flow paths in the intercooler cavity serve two functions: (1) the 26.7°C (80°F) gas flowing down the annulus between the exchanger shroud and liner eliminates the thermal barrier requirement; and (2) it provides an acceptable gas flow path inlet to the high-pressure compressor. Returning from the intercooler, the gas at 26.7°C (80°F) and 4.5 MPa (653 psia) enters the turbomachine through a multiplicity of holes in the casing and enters the high-pressure compressor.

After compression in the high-pressure compressor, the gas, now at 7.93 MPa (1150 psia) and 110°C (230°F), exits the turbomachine radially (again through holes in the casing) and flows into the vertical compressor discharge cavity in the PCRV. It is in this cavity that the hot gas core-to-turbine duct is installed, and an extremely desirable situation exists here in that the hot gas duct (which, after the turbomachine, is regarded as the most critical element in the PCL) is nearly pressure balanced.

From Fig. 5-3 it can be seen that the high-pressure compressor discharge gas enters the core cavity from a single duct and flows upward in an annulus formed between the core barrel (thermal shield and the liner). The gas exits from the top of the cavity via a short coaxial cross duct. The most difficult feature of the plant is, of course, the embodiment of the warm liner in the core cavity. To avoid local hot spots on the liner, uniform flow distribution in the annulus is essential. With a very large diameter [on the order of 11.7 m (38.5 ft)] annulus having only a single source and sink, it is mandatory that flow distribution devices (baffles, etc.) be incorporated. In this phase of the program the necessary detailed fluid dynamic analyses were not performed to resolve the uncertainties regarding flow maldistribution in the core cavity outer annulus.

The high-pressure compressor discharge gas [at 110°C (230°F)] flows from the core cavity via the vertical compressor discharge cavity and a short coaxial duct to the top of the recuperator cavity. The high-pressure gas enters the recuperator assembly in a plane just above the main support plate. The high-pressure gas flows downward inside the tubes, picking up heat from the low-pressure turbine discharge gas, The heated high-pressure gas, now at 454°C (850°F), is transferred back to the top of the recuperator assembly via integral return tubes in each of the 160 modules, thus eliminating the need for a large central return duct that was featured in earlier designs. Leaving the recuperator top plenum, the 454°C (850°F) gas is transported back to the core in a short coaxial duct. This "warm" duct is surrounded by compressor discharge gas, so it is essentially pressure balanced. With the return of the gas to the top core plenum, the loop circuit is completed.

The definition of the core auxiliary cooling system (CACS) was not addressed in the first design iteration of the demonstration plant because of a combination of (1) limited funding and resources, (2) more urgent SC-HTGR and GCFR priorities, and (3) perhaps more important, the lack of needed data from the HHT project. The major problem postulated was the ability to keep the core cavity liner cooled under CACS operation. A decision was made to size the CAHE so that gas at about 125°C (257°F) could be utilized for liner cooling. From Fig. 5-3 it can be seen that the CACS features resemble those in the SC-HTGR in that a bottom-fed bayonet tube CAHE and a top-positioned circulator are utilized. In fact, the size estimate for the CAHE was scaled from the 900-MW(e) SC-HTGR. Although the lack of CACS definition represents a problem in the demonstration plant design, it was necessary to "phantom" in a concept to complete the plant layout arrangement.

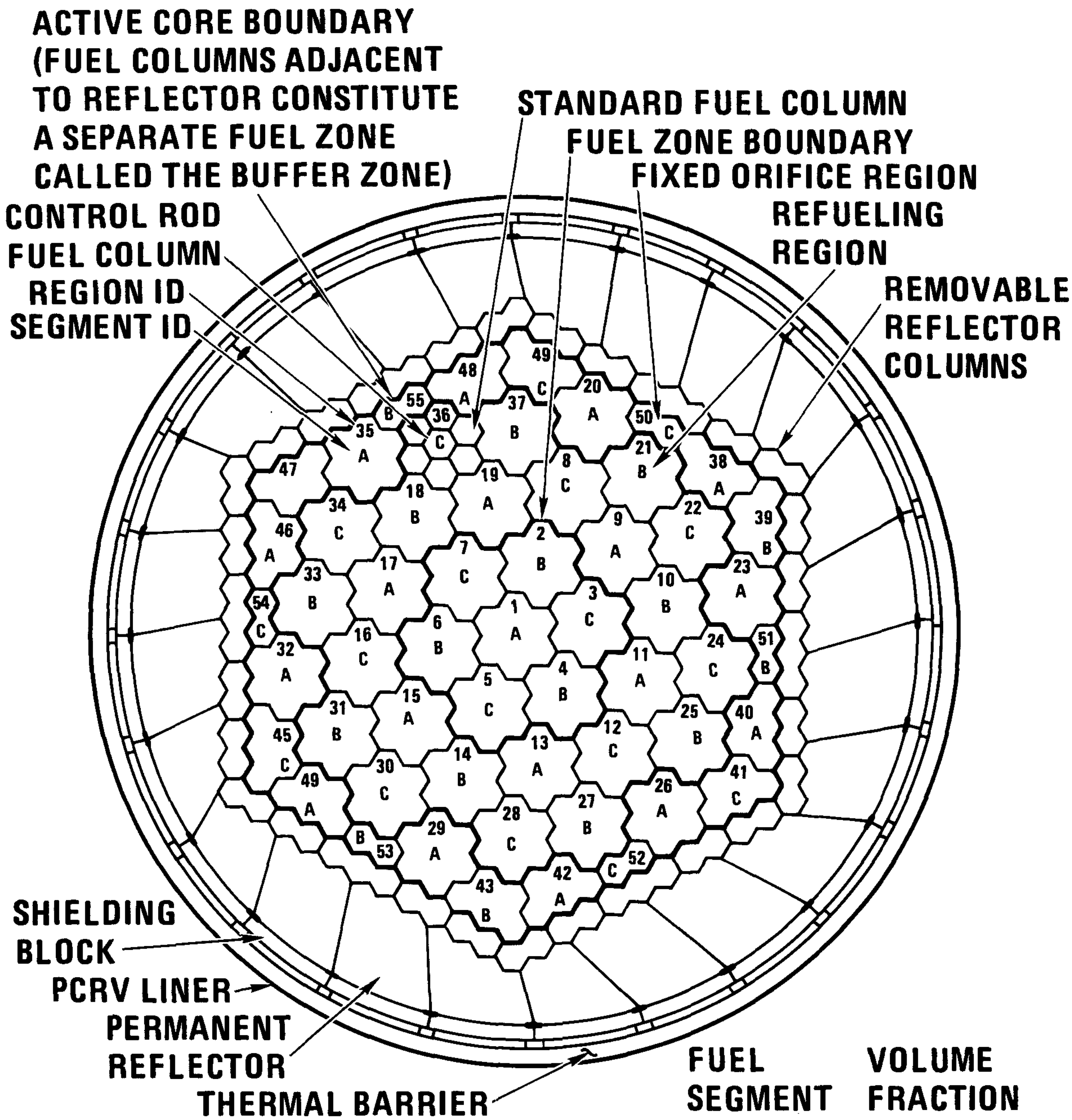
5.3.2. U.S. Version of HHT Demonstration Plant Core Study

The reactor core for the single-loop, intercooled GT-HTGR demonstration plant is designed for operation with MEU fuel. In the initial core layout studies, the basis for selection of the actual power level [1500 MW(t) nominal being the GA-HHT ground rule] was the desire to establish a physically compact arrangement with a minimum number of partial regions at the periphery. An attempt was made to avoid four-column regions because of problems with power tilts.

A 1530-MW(t) configuration was selected. The core layout, shown in Fig. 5-6, has a one-third symmetry. Region identification numbers and the refueling segments are also shown in Fig. 5-6. The major core design parameters and thermal and coolant data are shown in Tables 5-3 and 5-4, respectively. The reactor coolant enters the core at 453°C (848°F) and exits at 850°C (1562°F). Unlike a highly enriched uranium (HEU) core, an MEU core cannot support a heavy thorium load owing to the presence of the second fertile particle U-238. The GT-HTGR demonstration plant core uses a carbon-to-thorium ratio of 350 during initial core operation and approximately 580 during equilibrium conditions. The prismatic fuel element employed in the core is identical to the 10-row FSV element. The designs of a standard and a control element are shown in Fig. 5-7.

Core performance studies were completed for a conceptual 1530-MW(t) GT-HTGR demonstration plant. The core is fueled with MEU and has a power density of 6.5 W/cc and a 3-yr fuel cycle. The fuel particles consist of fissile TRISO UC_2 and fertile TRISO ThO_2 and are accommodated in 10-row FSV type fuel elements. The performance studies included core thermal behavior, fuel particle failure, fuel kernel migration, fast neutron flux time histories, coolant flow distribution, and gaseous fission product release. The significant results of these studies are shown in Table 5-5.

The performance of the GT-HTGR demonstration plant core is good. The gaseous fission product releases are within limits developed for the SC-HTGR.



FUEL SEGMENT	VOLUME FRACTION
A	0.3656
B	0.3172
C	0.3172

Fig. 5-6. Core layout for 1530-MW(t) GT-HTGR demonstration plant

TABLE 5-3
BASIC CORE PARAMETERS OF GT-HTGR DEMONSTRATION PLANT

Core thermal power	1530 MW(t)
Power density	6.5
Fuel block design	FSV - 10 row
Fuel in-core lifetime	3 yr
Reload interval	1 yr
Fraction of core reloaded annually	∿33%
Capacity factor	80%
Carbon/thorium initial core reloads	350/580
Core volume	234.6 m^3
Fuel rod diameter	1.2 cm
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}	453°C (848°F)
T_{Outlet}	850°C (1562°F)

TABLE 5-4
THERMAL AND COOLANT FLOW DATA FOR 1530-MW(t)
GT-HTGR DEMONSTRATION PLANT

Coolant (helium) inlet temperature at the core	453°C (848°F)
Coolant outlet temperature at the reactor	850°C (1562°F)
Coolant flow rate	741 kg/s (5.883×10^6 lb/hr)
Coolant pressure at core inlet	7.77 MPa (1128 psia)
Core bypass flow fraction	0.13
Core bypass power fraction	0.0935

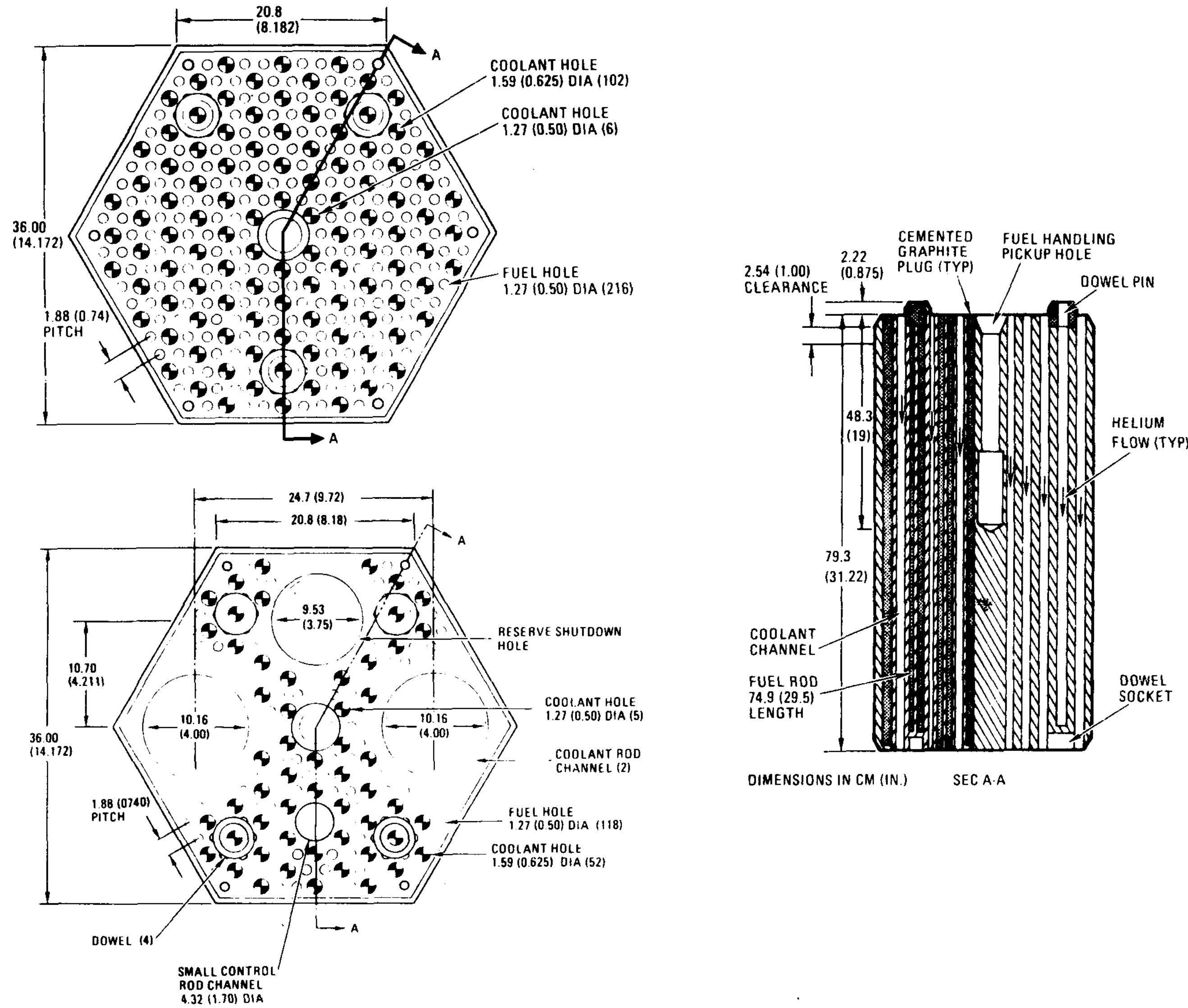


Fig. 5-7. HTGR fuel element

TABLE 5-5

1530-MW(t) GT-HTGR DEMONSTRATION PLANT CORE PERFORMANCE SUMMARY

[T_{in} = 453°C (848°F); T_{out} = 850°C (1562°F)]

Peak fuel centerline temperature	1205°C (2200°F)
Time-averaged fuel centerline temperature during 3-yr core residency (maximum)	1149°C (2100°F)
Peak moderator (H-451 graphite) temperature	1149°C (2100°F)
Time-averaged moderator temperature during 3-yr fuel core residency (maximum)	1094°C (2000°F)
Peak fast neutron fluence	5×10^{21} nvt
Fast neutron fluence design limit	8×10^{21} nvt
Peak burnup (FIMA fraction)	
TRISO UC_2	0.25
TRISO ThO_2	0.035
Total failed fuel particle fraction	
TRISO UC_2	0.14%
TRISO ThO_2	0.07%
Fraction of fissions in failed particles (total)	
Computed value	0.00124
Level A[a]	0.0025
Kr-85m (R/B) ratio	
80% confidence level	3.11×10^{-5}
Level A[a]	6.16×10^{-5}
Xe-138 (R/B) ratio	
80% confidence level	2.678×10^{-6}
Level A[a]	4.57×10^{16}

[a]Level A values refer to the design criteria values based on safety considerations.

The use of uniquely optimized axial power and flux profiles coupled with excellent fuel zoning minimized the peak fuel rod centerline temperatures and yielded fairly uniform core axial temperature profiles.

The most notable feature of the core performance is the negligible fraction of failure due to either SiC-fission product attack or kernel migration (zero on a core average basis). The estimated peak fuel rod centerline temperature is 1205°C (2200°F) at any time during core operation. It appears feasible to eliminate the negligible local fuel particle failure which occurred owing to the SiC-fission product attack by finer adjustments in the fuel zoning. The total failed fuel fraction and fissions in failed fuel particles are also within design limits.

Although the results of the studies presented herein predict an excellent performance by the GT-HTGR demonstration plant core, the metallic fission product release from the core (studied separately) is the critical parameter which validates the core design and performance. The present estimate of cesium release from the core is approximately 36,000 Ci on a 40-yr plateout basis.

5.3.3. GA-HHT Demonstration Plant Comparison

One of the basic criteria established for the demonstration plant studies was that close technical coordination between GA and HHT must be observed in order for the GA prismatic core version to be considered a backup or real alternate to the European design. However, at the initiation of design studies, a renewal of the technical exchange agreement had not been effected, and both GA and HHT started work without the benefit of knowing what direction was being taken by the other. Design considerations were adopted which had a lasting effect (in 1978) regarding plant configuration. A comparison of these design considerations is given in Table 5-6, and some of the basic issues are briefly discussed below.

TABLE 5-6
HHT-GA DEMONSTRATION PLANT DESIGN CONSIDERATIONS

Criteria	HHT	GA
Main criteria		
PCRV	Simulate commercial plant design	Diameter minimization
Diameter [m (ft)]	46 (151)	36.3 (119)
Height [m (ft)]	46 (151)	35.4 (116)
Design criteria	Max. system pressure in all cavities	System cavity pressure
Core	1640-MW(t) pebble bed	1530-MW(t) prismatic
Concentric cavity	Concentric	Offset as necessary to minimize PCRV diameter
Cavity diameter [m (ft)]	13.16 (43.2)	11.9 (39)
Turbomachine	675 MW(e), 50 Hz, dual turbine inlets and outlets Growth to 800-900 MW(e)	620 MW(e), 60 Hz, single turbine ducts, same machine as for commercial plant
Heat exchangers		
Size	Half-size units, two trains/loop	Full-size units as for two-loop commercial plant
Inspectability	Ability to inspect and repair individual tubes	Module plugging for recuperator, tube plugging for water-to-helium units
CACS	4 x 100% capacity	4 x 100% capacity
Position in PCRV	Equispaced	Positioned to minimize impact on PCRV size
Dominant criteria	Simulation of 3000-MW(t) plant PCRV arrangement, scaling of all components from demonstration plant necessary	PCRV size minimization Commercialization aspects Utilization of demonstration plant proven components for commercial plant

The following three criteria established by HHT (with no flexibility afforded at this stage) had a strong impact on the plant layout:

1. Dual turbine inlet and exits.
2. PCRV plan view to simulate one-loop commercial plant.
3. Four equispaced CACS's.

The first of these criteria understandably stems from the need for dual inlet and exits in a single machine of 1240 MW(e) and the desire to scale up from the tested demonstration plant turbomachine. GA has confirmed that the adoption of these criteria results in a very large PCRV diameter. For the HHT project, going from the one-loop demonstration plant to the one-loop commercial plant necessitates scaling of the turbomachine power by a factor of two and the diameters of the heat exchangers and ducts by root two.

The GA established criteria are centered around two basic issues, which are regarded as vitally important: (1) the commercial plant should utilize PCL components that have been tested and proven in the demonstration plant, and (2) efforts should be made to minimize the diameter of the demonstration plant PCRV, because it may well prove possible to commercialize a plant in this power class at a future date. The ramifications of the differences between GA and HHT, as far as the major features of the demonstration plants are concerned, are given in Table 5-7.

5.3.4. Two-Loop Alternate Commercial Plant Configuration

A study was performed in which 16 plant configurations were established, evaluated, and rated. The 16 plant concepts can be classified into the following four general categories:

1. Concepts with a conventional water-cooled and insulated liner in the core cavity and warm liners in the heat exchanger and turbomachine cavities.

TABLE 5-7
MAJOR FEATURES OF HHT AND GA DEMONSTRATION PLANTS

Demonstration Plant Concept	HHT Design	Alternative GA Design
Core type	Pebble bed	Prismatic
Core rating [MW(t)]	1640	1530
Output power [MW(e), Hz]	675, 50	620, 60
Core cavity	Centralized	Offset
Turbomachine	Dual turbine inlet and exit ducts	Single turbine inlet and exit duct
	Positioned under core	Chordal position
Heat exchangers	Two trains of exchangers	Single exchangers
	7-module, straight tube recuperator	161-module, straight tube recuperator
	7-module, straight tube He-H_2O units	Helical geometry He-H_2O units
CACS	4 x 100% capacity units, equispaced	4 x 100% capacity units, PCRV diameter minimization
Salient ground rules	Simulation of PCRV arrangement for 1200-MW(e) commercial plant	Commonality with two-loop commercial plant
	Layout dominated by turbomachine	Utilization of demonstration plant proven components for commercial plant
	All components must be scaled up for commercial plant.	Minimum cost for possible commercialization of plant in 600-MW(t) range
Plant status	All of the above, essentially utility-directed, ground rules are fixed.	Design work completed December 1978
		Cost estimates made by March 1979
	Plant design to be pursued and cost estimates prepared.	Further design studies of this alternative not planned.

2. Concepts with conventional water-cooled and insulated liners in all cavities where required.

3. Concepts with warm liners in all cavities.

4. 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.

While it is recognized that evaluating and rating a family of plant variants is difficult, some form of comparative selection criteria was necessary to identify the candidate plant for specialist design attention. All the configurations were evaluated on the basis of a simplified and non-weighted rating system which included the following:

1. Primary system gas flow path complexity (strongly influenced by major cavity thermal barrier/warm liner requirements).

2. Utilization of one or two heat exchangers (particularly the recuperator) per PCL.

3. Adaptability of the demonstration plant configuration to a two-loop commercial plant.

4. Maintenance and ISI considerations.

5. Plant cost (based on intuitive feeling, since cost data were not generated).

6. Safety and licensing considerations.

7. Feasibility issues.

8. Attractiveness as a commercial plant (i.e., PCRV diameter minimization).

The screening and evaluation process led to the following decisions with regard to plant concept selection:

1. Circular PCRV shape.
2. Chordal turbomachine location; turbomachines not parallel.
3. Offset core cavity.
4. Conventional liner approach.

Upon selection of the two-loop plant concept, specialist tasks were initiated in the following areas: (1) PCRV structural design (GA), (2) reactor internals design (GA), (3) turbomachine design (UTC), and (4) heat exchanger sizing and design (GA and CE). Toward the end of the 1-yr study, technical inputs from these areas were factored into the generation of a plant layout drawing, which is shown in Figs. 5-8 through 5-11. The main features of the two-loop plant are given in Table 5-8. Major plant parameters are given in Table 5-9, and a simplified flow diagram is shown in Fig. 5-12. A description of the plant primary system is given below.

As shown by the plan view in Fig. 5-8, the goal of minimizing PCRV diameter was realized by offsetting the core cavity and by establishing a chordal (as opposed to radial) turbomachine orientation. Each loop contains a single train of heat exchangers, with the recuperator and precooler cavities positioned over the turbomachine. The recuperator cavity is very large, owing to the combined effects of the 1500-MW(e) loop rating, high effectiveness requirement (0.898), and the decision (based on inputs from UTC) to utilize a single core-to-turbine duct. This had a strong influence on the two-loop plant layout, and it should be pointed out that there is little extension in power range beyond 620 MW(e) for this turbomachine if the single inlet duct approach remains a requirement. If power growth potential becomes an important aspect of future two-loop plant studies, dual turbine inlet and exit ducts will essentially be mandated, requiring two heat exchanger trains per loop with attendant complications in the PCRV design.

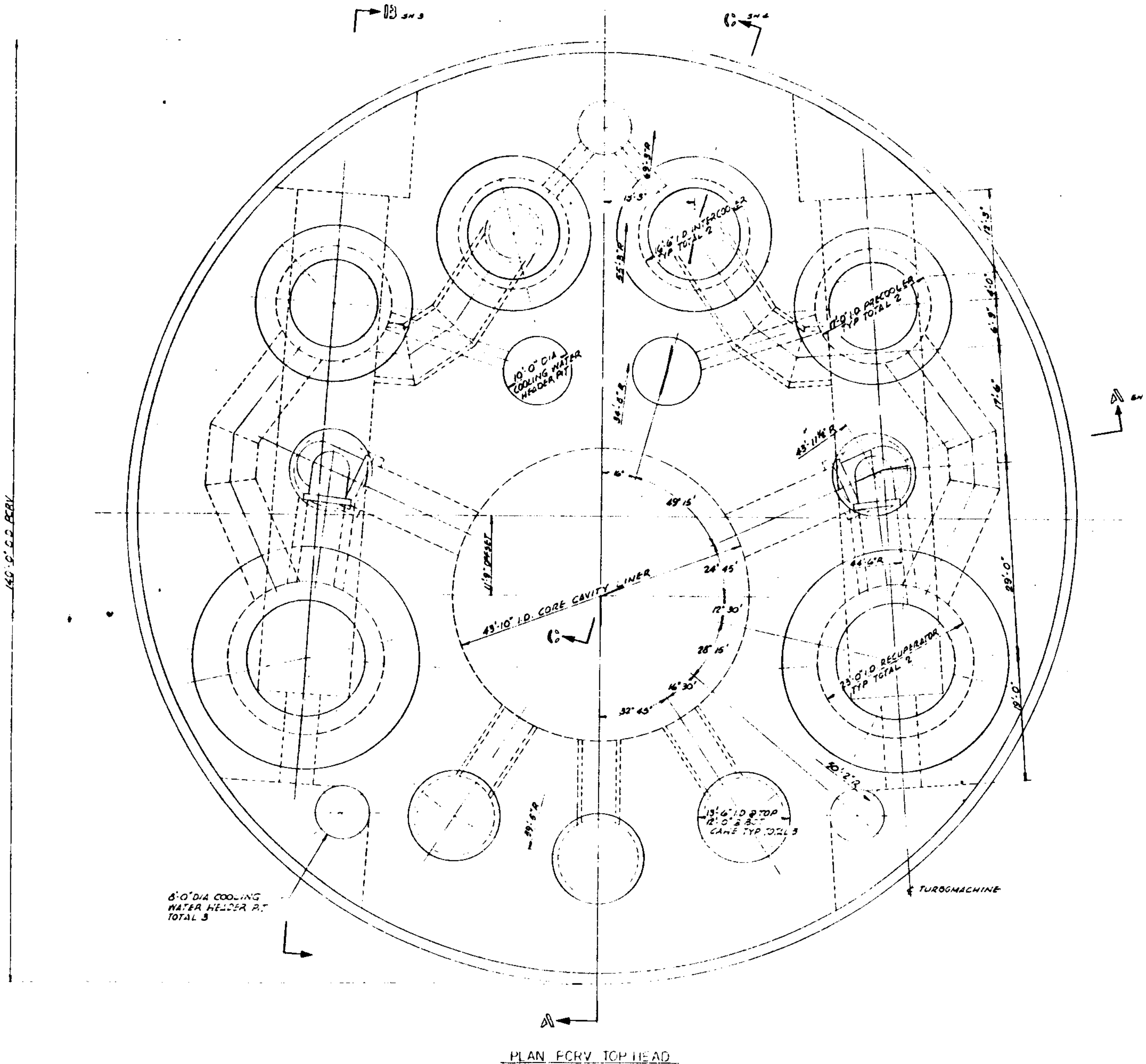


Fig. 5-8. Plan view of PCRV top head for two-loop alternate commercial plant configuration

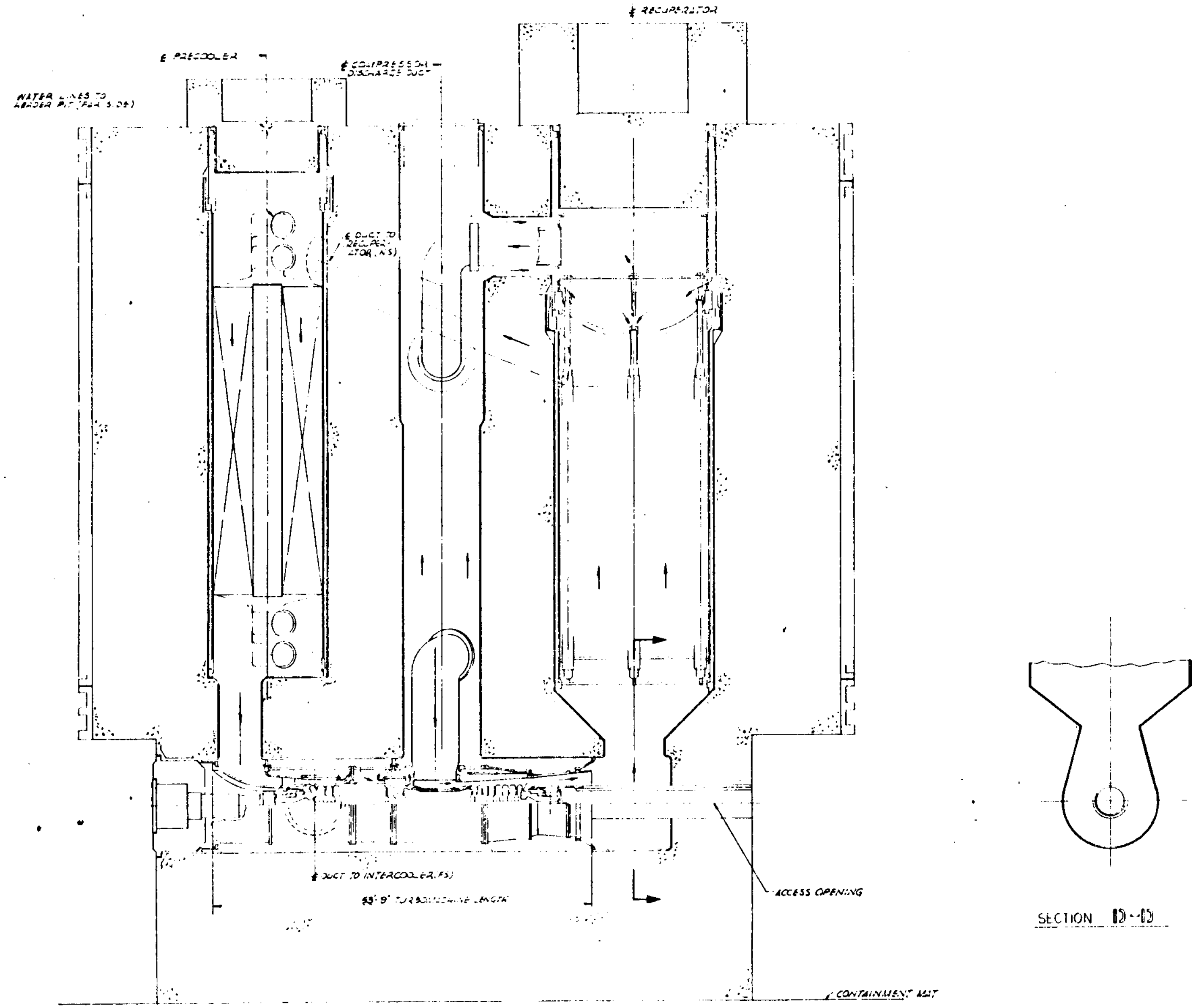


Fig. 5-9. Section B-B of Fig. 5-8

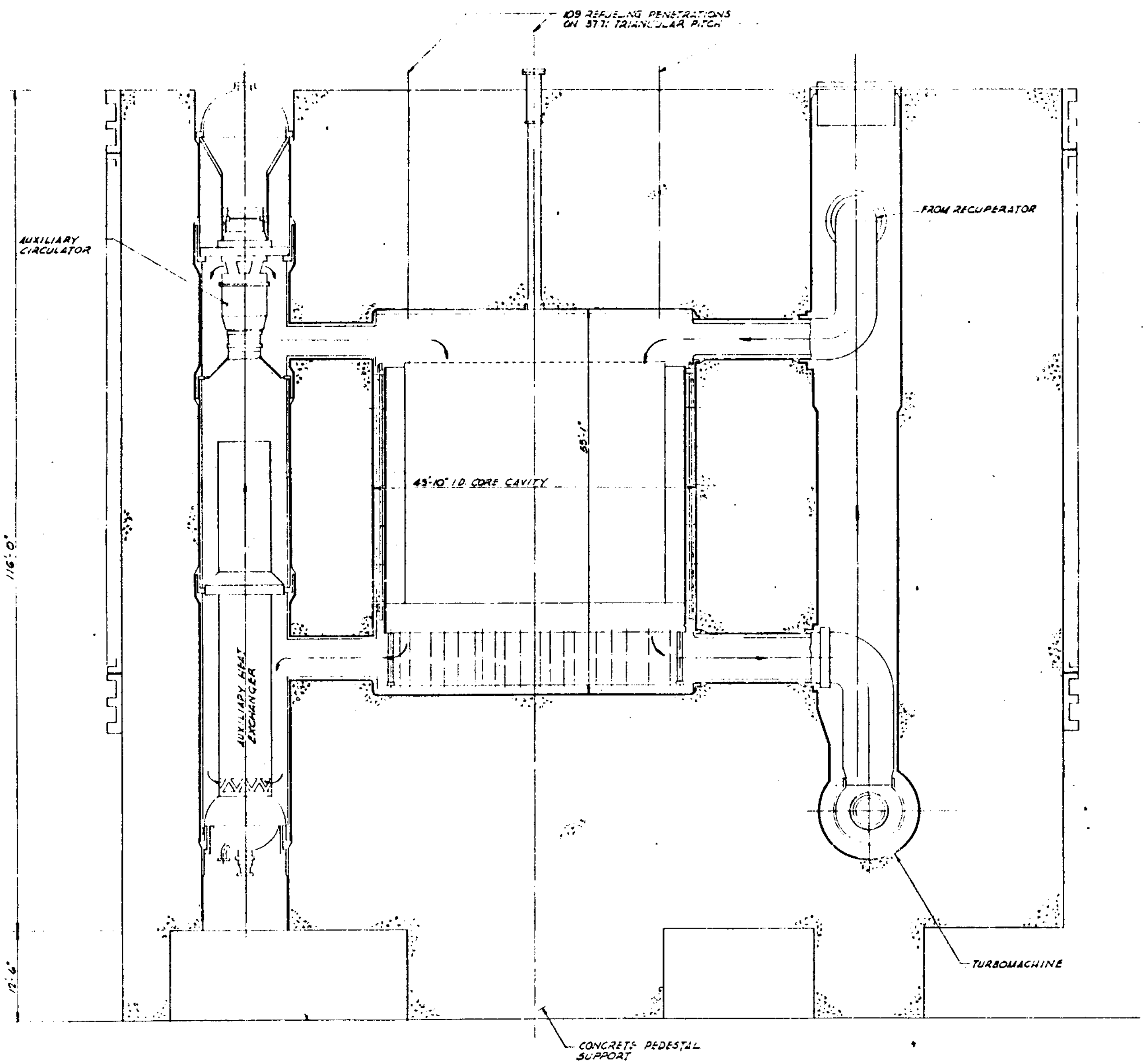


Fig. 5-10. Section A-A of Fig. 5-8

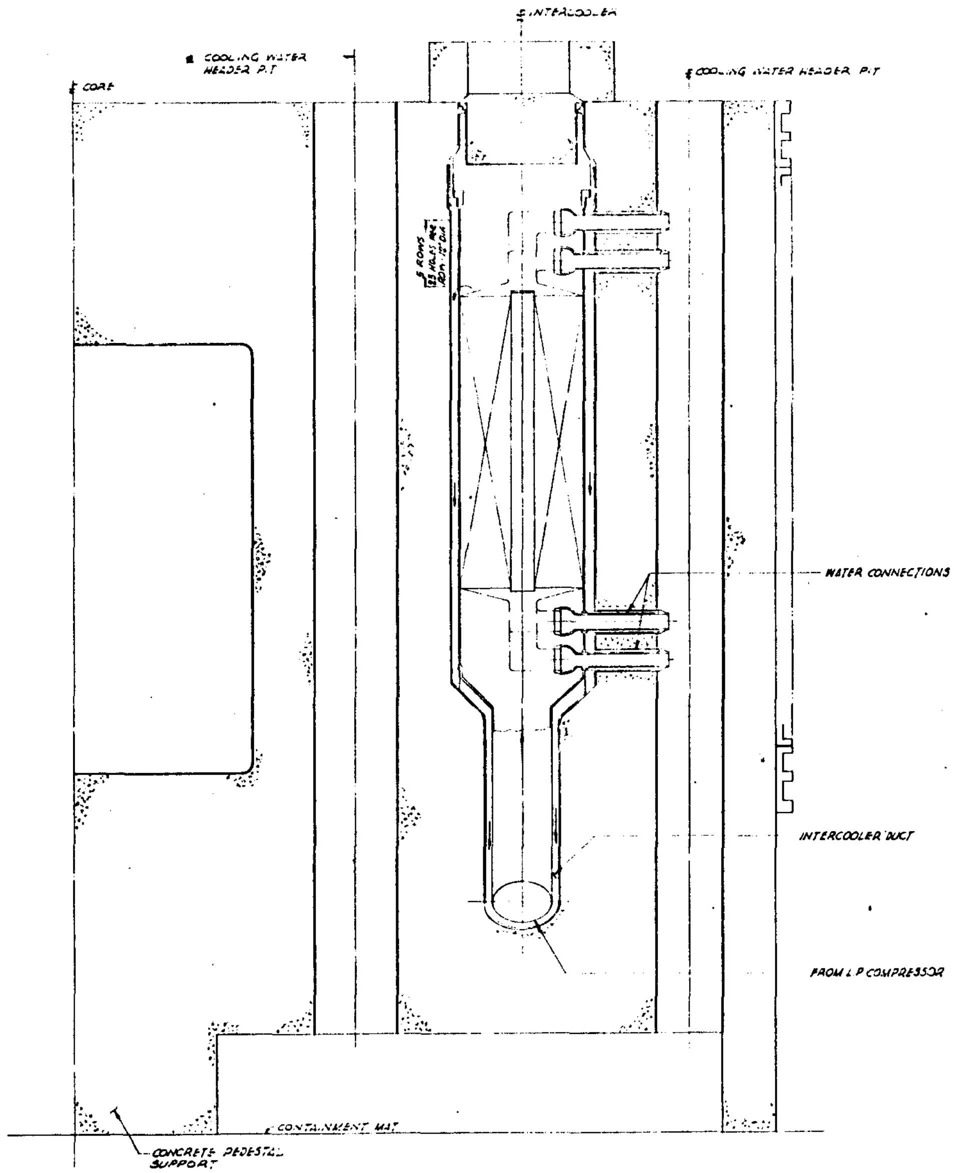


Fig. 5-11. Section C-C of Fig. 5-8

TABLE 5-8
MAIN FEATURES OF TWO-LOOP COMMERCIAL PLANT ALTERNATE

Integrated Direct Cycle Plant

Prismatic Core, Thermal Rating 3000 MW(t)

MEU Fuel (3-yr Fuel Cycle)

Reactor Core Power Density 6.8 W/cc

Intercooled Cycle with High Degree of Recuperation

$$\left.\begin{array}{lcl} P_{max} & = & 7.93 \text{ MPa (1150 psia)} \\ T_{max} & = & 850°\text{C (1562°F)} \\ R_{comp} & = & 3.0 \\ E_{recup} & = & 0.90 \end{array}\right\} \text{Plant Efficiency} = 41.2\%$$

Turbomachine Rating 620 MW(e) (same as demonstration unit with minor casing changes)

Water-Cooled and Insulated Liners Throughout

PCRV Details

Offset Core Cavity

Diameter 42.7 m (140 ft)

Height 35.4 m (116 ft)

Chordal Turbomachine Position

CACS - 3 x 100% Units

Two-Bearing Turbomachine (Single Turbine Inlet Duct)

Man Access Provision to Bearing Cavity Areas

Straight Tube, Modular Recuperator
Helical Bundle Precooler and Intercooler } Exchangers as for Demonstration Plant

Dry-Cooled Plant

Emphasis placed on gas flow simplicity and utilization of components tested and proven in demonstration plant

TABLE 5-9
MAJOR DESIGN PARAMETERS FOR 3000-MW(t), TWO-LOOP
GT-HTGR ALTERNATE COMMERCIAL PLANT

Turbine inlet temperature	850°C (1562°F)
Ambient air temperature	15°C (59°F)
Thermodynamic cycle	Intercooled
Heat rejection mode	Dry cooling
Liner type	Conventional
Compressor pressure ratio	3.0
Compressor inlet temperature	LP - 26.8°C (80.2°F) HP - 26.7°C (80.0°F)
Maximum system pressure	7.87 MPa (1142 psia)
Overall system pressure loss, Σ ($\Delta P/P$)	11.21%
Compressor flow	753 kg/s (5.978×10^6 lb/hr)
Recuperator effectiveness	0.898
Turbine isentropic efficiency, across blading	92.2%
Compressor adiabatic efficiency, across blading	LP - 90.8% HP - 90.2%
Generator efficiency	98.8%
Turbine cooling flow, discs	3.6%
Precooler water outlet temperature	87°C (188.5°F)
Intercooler water outlet temperature	65.6°C (150°F)
CACS parasitic heat loss	1.9 MW(t)
Primary system heat loss	15.6 MW(t)
Auxiliary power	11.0 MW(t)
Station efficiency	41.17%[a]
Net electrical power output	1235 MW(e)[b]

[a] Parameter selected on thermodynamic basis; plant not optimized for minimum power generating cost.

[b] Initial rating of the turbomachine was 620 MW(e). However, the higher than projected primary system pressure losses associated with the final plant layout resulted in a slight loss of station efficiency.

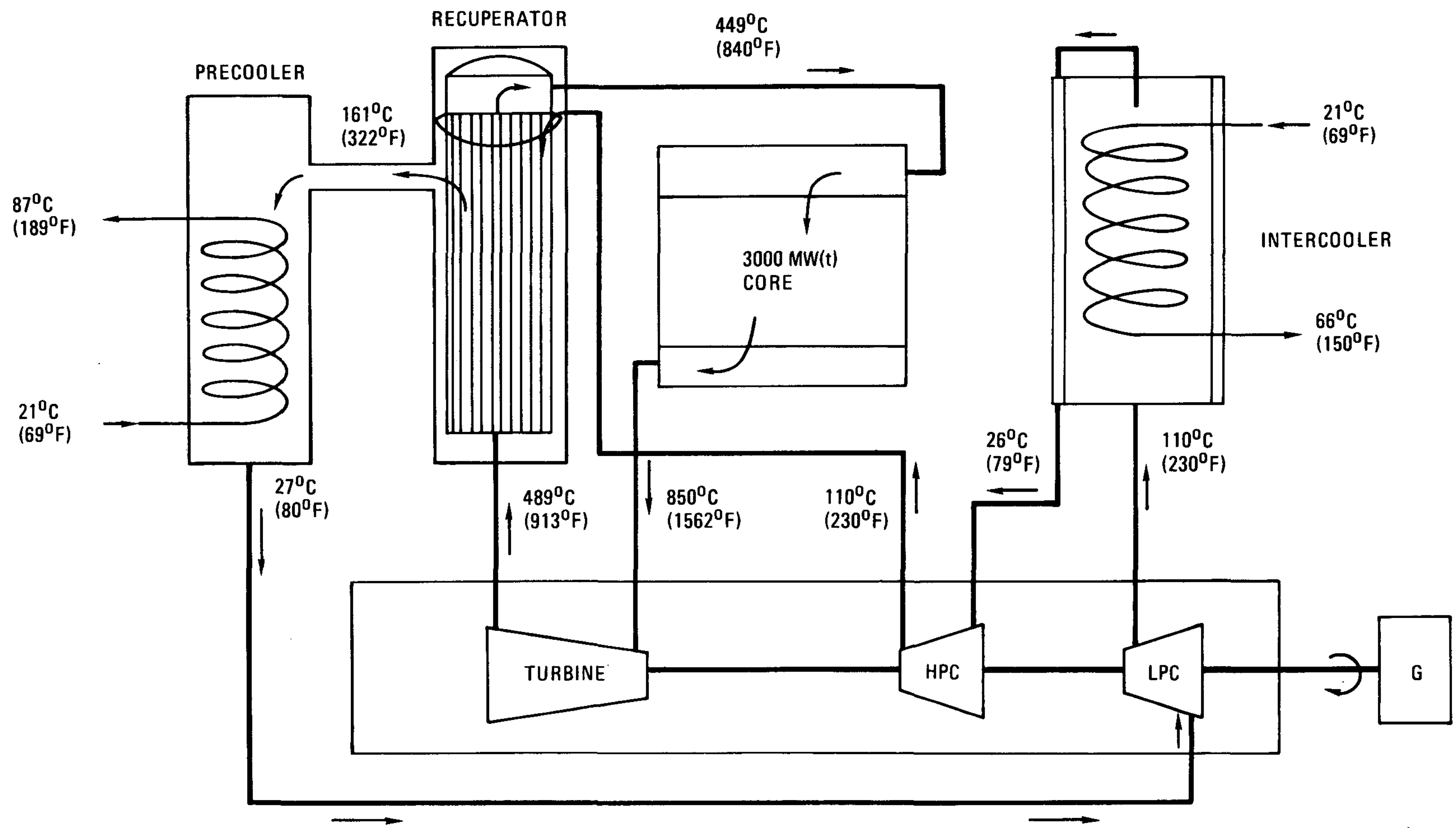


Fig. 5-12. Cycle diagram for two-loop plant with conventional liners

Figure 5-8 also shows that each intercooler is positioned to the side of its turbomachine cavity. The PCRV arrangement includes unpressurized, small-diameter PCRV cavities adjacent to the precooler and intercooler cavities to provide maintenance access to the pipe chases that route the water plumbing to the heat exchangers. The PCRV space allocated for the two intercooler cavities is counterbalanced to some extent by the three CACS cavities, each of which houses a 100% capacity CACS unit.

Referring to Figs. 5-8 through 5-12, the primary system gas flow paths can be traced as follows. Hot high-pressure helium at 850°C (1562°F) exiting from the core bottom plenum is transported in a coaxial duct down to the turbomachine turbine inlet plane. After expansion in the turbine, the 494°C (921°F) low-pressure helium at the turbine discharge leaves the turbomachine cavity and flows vertically upward to enter the recuperator cavity, where regenerative heat transfer to high-pressure helium enroute to the reactor takes place. During this waste heat recovery process, the low-pressure helium flowing upward on the shell side of the recuperator is cooled to 161°C (322°F) before it leaves the recuperator cavity via a sidewall opening located just beneath the recuperator main support plate. The low-pressure helium then proceeds via cross-ducting embedded in the PCRV to the precooler, in which the cycle reject heat is transferred to a pressurized, single-phase circulating water system. After entering the precooler cavity, the low-pressure helium flows downward across the helically coiled precooler tube bundle, giving up its heat to the water flowing upward inside the tubes. At the bottom of the precooler cavity, the low-pressure helium, which has been cooled to 26.7°C (80°F), re-enters the turbomachine cavity at the plane of the low-pressure compressor inlet. The eight-stage low-pressure compressor pumps this helium up to 6.90 MPa (670 psia) and 110°C (230°F), after which the helium is routed to the bottom of the intercooler cavity via coaxial ducting. Upon entering the intercooler cavity, the helium flows upward across the shell side of the helically coiled intercooler tube bundle, transferring the heat of low-pressure compression to pressurized, single-phase cooling water flowing downward inside the tubes. At the top of the intercooler cavity, the helium, which has been cooled back down to 26.7°C (80°F), is returned to the turbomachine cavity via the annular flow passages formed between (1) the PCRV cavity sidewall and intercooler shroud and (2) the concentric inlet and outlet cross-ducting connecting the intercooler and turbomachine cavities. Returning helium from the intercooler enters the turbomachine through a perforated section of its casing and flows to the high-pressure compressor, which pumps the gas up to 7.93 MPa (1150 psia) and 110°C (230°F). The high-pressure helium exits the turbomachine radially (again through holes in the casing) and flows into the vertical compressor discharge cavity in the PCRV. It is in this cavity that the hot gas core-to-turbine duct is installed, providing the highly desirable conditions required to maintain the hot duct (regarded as the most critical element after the turbomachine) in a nearly pressure-balanced operating mode. Flowing upward in this vertical cavity, the high-pressure compressor discharge helium reaches the high-pressure region of the recuperator cavity via the annular portion of a short coaxial duct, then flows downward inside the recuperator tubes and is returned to the top of the recuperator assembly via individual "return tube" risers provided in the 161 modules (one return tube per module). This "integral return tube" recuperator concept avoids the need for the large, high-pressure center return duct that was featured in earlier designs. Leaving the recuperator top plenum, the 449°C (840°F) gas is returned to the upper plenum of the core via the inner duct of the coaxial installation at the top of the recuperator cavity. The downward, heated passage of this helium through the core completes the cycle.

In comparison with the GA demonstration plant circuitry, which is predicated on the "warm liner" concept, the gas flow paths in the two-loop plant are simple and straightforward. The technical aspects of the decision to employ conventional water-cooled and insulated liners in the two-loop plant are two-fold:

1. Unresolved technical uncertainties about the warm liner concept that appeared during the GA demonstration plant study (core support, cooling flow distribution, etc.).

2. Questionable operability of a multiple-loop plant embodying a warm liner in the core. A multiple loop plant concept predicated entirely on the warm liner approach couples the loops via the core cavity in such a way that a single-loop rapid shutdown (which could be triggered by a variety of events) can result in flow redistributions with potentially adverse impact on primary system components. Incorporating valves to prevent this problem is not considered practical at this time. Hybrid liner concepts (e.g., conventional/warm liner combinations) conceived specifically to circumvent this operability issue were beyond the scope of this study.

In retrospect, the significance of this study lies in the application of intercooling in conjunction with higher loop thermal capacity to establish an alternate commercial plant concept that can be compared with the reference three-loop, non-intercooled plant. The outcome of this study is a plant design that meets the cycle efficiency goal and offers immediate prospects for additional performance improvement with further study. However, the PCRV diameter of 42.7 m (140 ft) is 3.35 m (11 ft) larger than that of the three-loop non-intercooled commercial plant design, a reflection of the complexity and packaging inefficiencies associated with incorporating intercooling and 50% larger primary system components into a circular PCRV.

5.3.5. Three-Loop Non-Intercooled Plant Configuration

The design effort related to the three-loop non-intercooled plant was aimed at updating the earlier delta reference plant design. It was felt prudent to also briefly evaluate alternate approaches, particularly in light of the decision to adopt the warm liner concept for the demonstration plant. Basically, three different plant concepts were studied:

1. Conventional liner throughout.

2. Conventional liner in core cavity and warm liners in the heat exchanger cavities.

3. Warm liners in all cavities with the exception of the turbomachine cavities.

Because many of the engineering, safety, and licensing problems associated with the warm liner (as identified in the demonstration plant) have not been resolved, a decision was made to pursue a plant variant with conventional water-cooled and insulated liners throughout. It should be pointed out that partial elimination of the thermal barrier is desirable for the following reasons: (1) cost reduction, (2) partial liner inspectibility/repair, and (3) minimization of problems associated with oil ingress into the primary system. Item 2 warrants further study, since in 1978 the limited funding and resources did not permit an in-depth study.

The power plant plot plan concept shown in Fig. 5-13 illustrates the general layout of buildings and dry cooling towers for a twin 3000-MW(t) plant embodying three PCLs of the non-intercooled type. The reactor service building and fuel storage facilities are shared by the two reactor units. Each unit has a separate control building and safety-related auxiliaries. A runway system is provided for turbomachinery and generator handling. Space is allocated on the plot plan for an ammonia turbine building should the binary-cycle option be selected.

Based on the utilization of an existing 3000-MW(t) core design, the GT-HTGR embodies three PCLs, each rated at 1000 MW(t). The simplified isometric diagram of the reactor and primary system in Fig. 5-14 shows the core, turbomachinery, heat exchangers, and entire helium inventory enclosed in the PCRV. The major design parameters for the non-intercooled plant are given in Table 5-10. The simplified loop diagram is shown in Fig. 5-15. As shown in this diagram, each loop includes a single-shaft gas turbine, a recuperative gas-to-gas heat exchanger, and a precooler (gas-to-water exchanger) for cycle heat rejection.

Upon selection of the three-loop plant concept, specialist tasks were initiated in the following areas: (1) PCRV structural design (GA), (2) reactor internals (GA), (3) turbomachine inputs (UTC), and (4) heat

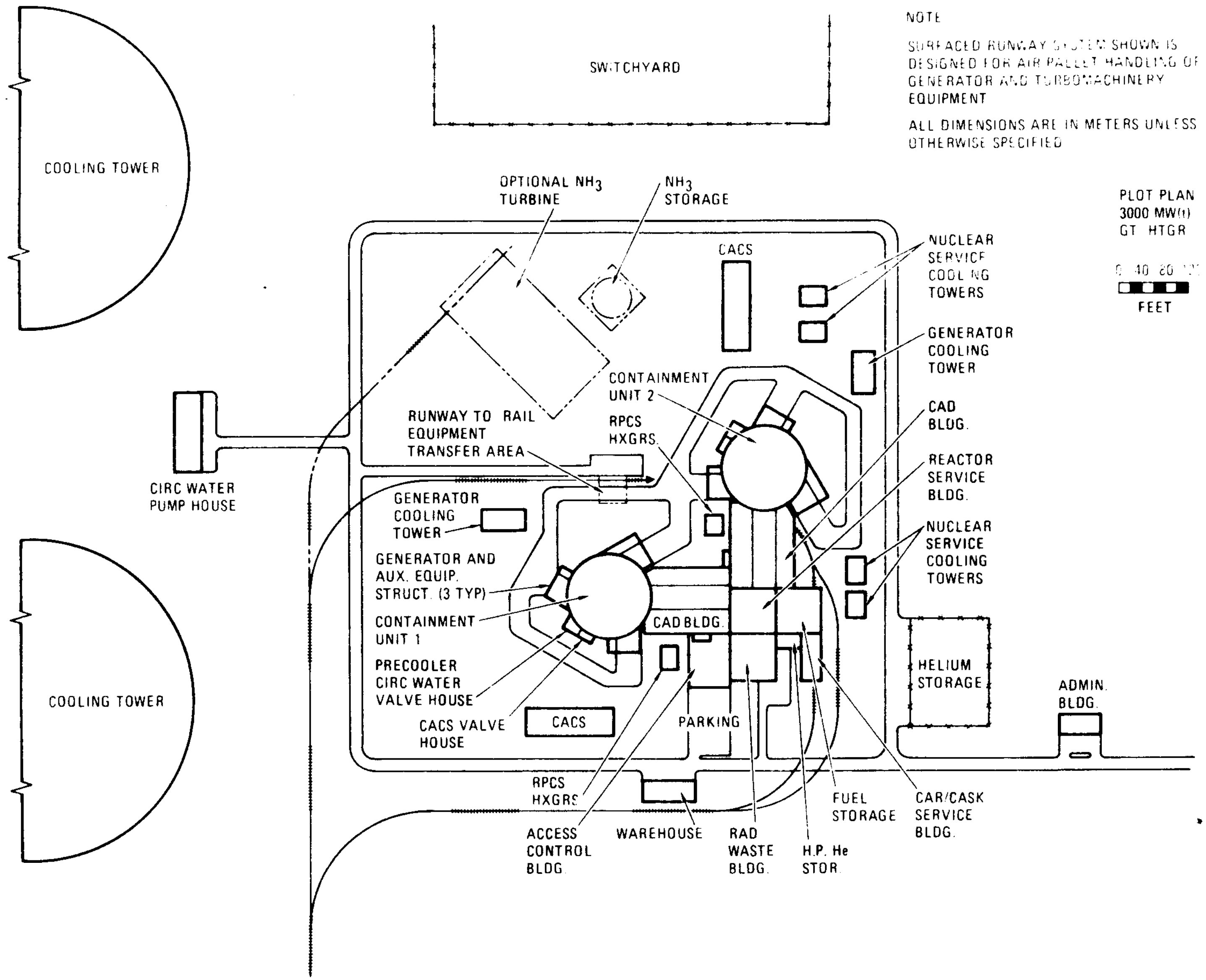


Fig. 5-13. Plot plan for GT-HTGR plant with twin 3000-MW(t) reactors

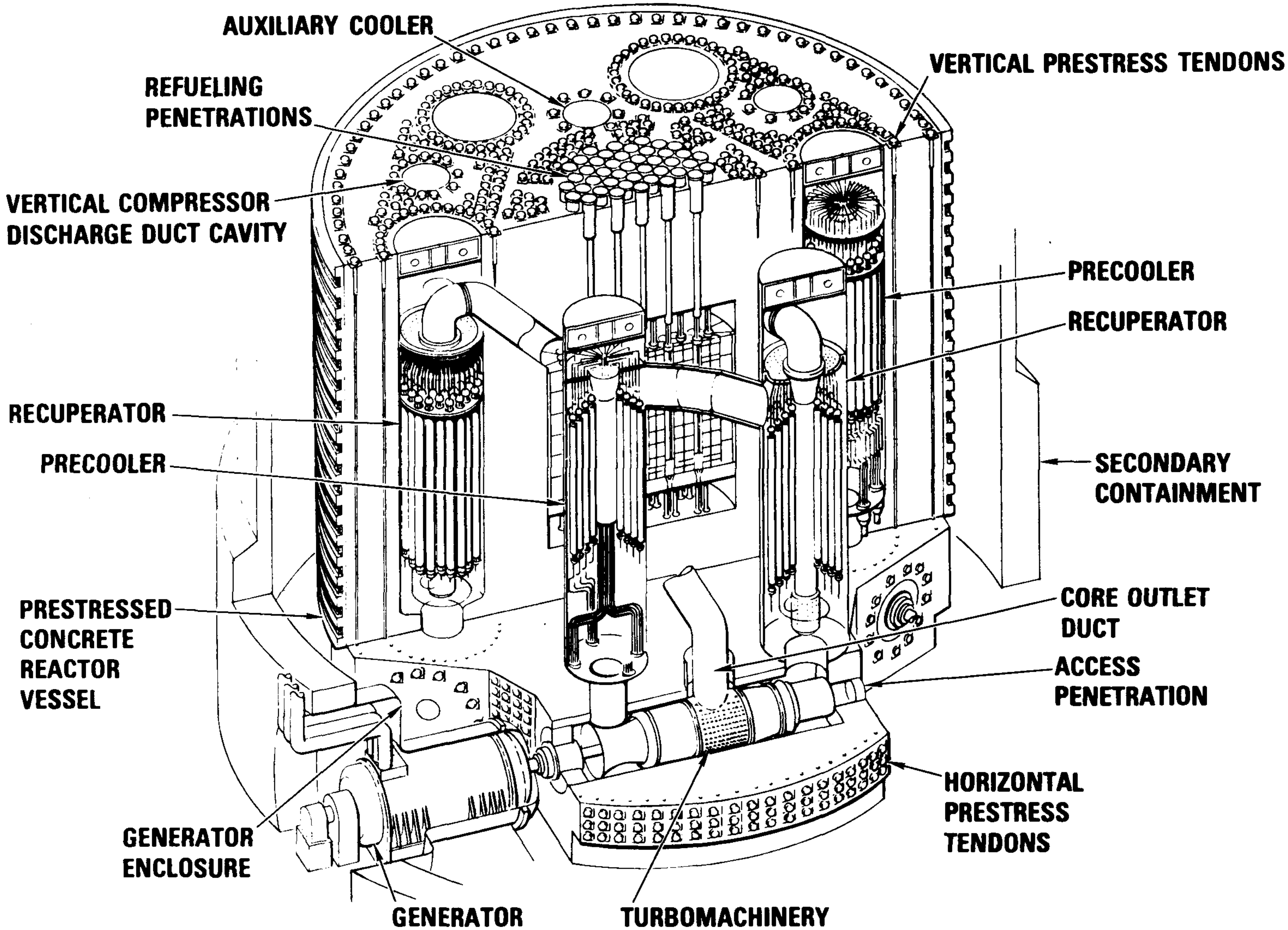


Fig. 5-14. Integrated GT-HTGR with 3000-MW(t) reactor core and three power conversion loops

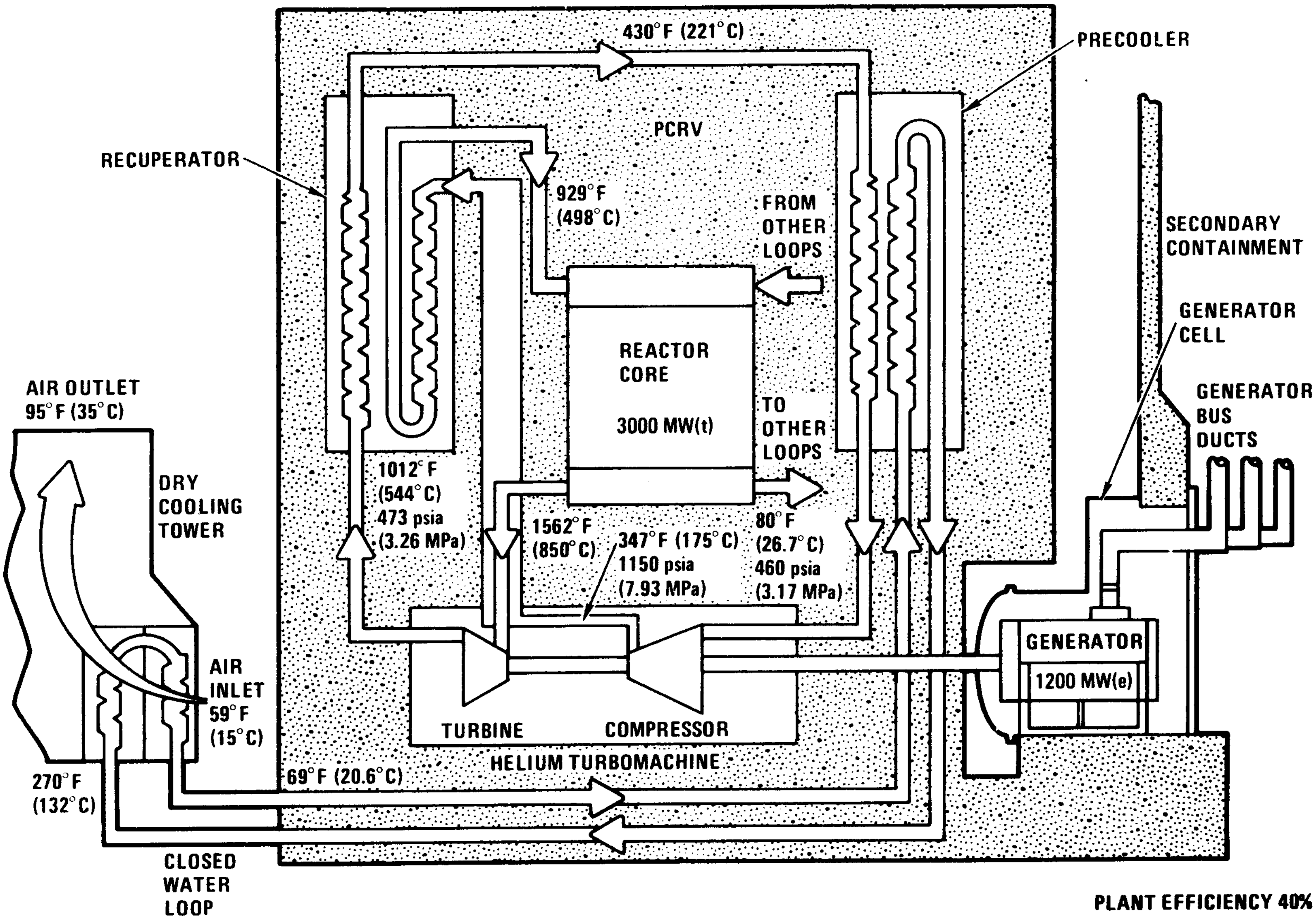


Fig. 5-15. Loop cycle diagram for dry cooled GT-HTGR power plant

TABLE 5-10
MAJOR DESIGN PARAMETERS FOR 3000-MW(t), THREE-LOOP
GT-HTGR COMMERCIAL PLANT

Power conversion loop rating	1000 MW(t)
Turbine inlet temperature	850°C (1562°F)
Ambient air temperature	15°C (59°F)
Thermodynamic cycle	Non-intercooled
Heat rejection mode	Dry cooling
Liner type	Conventional liner
Compressor pressure ratio	2.5
Compressor inlet temperature	26.7°C (80°F)
Maximum system pressure	7.93 MPa (1150 psia)
Overall system pressure loss, Σ ($\Delta P/P$)	7.5%
Compressor flow	570 kg/s (4.52×10^6 lb/hr)/loop
Recuperator effectiveness	0.898
Turbine isentropic efficiency, across blading	91.8%
Compressor adiabatic efficiency, across blading	89.8%
Generator efficiency	98.8%
Turbine cooling flow, discs	3.6%
Precooler water outlet temperature	132°C (270°F)
CACS parasitic heat loss	1.9 MW(t)
Primary system heat loss	15.6 MW(t)
Auxiliary power	11.0 MW(t)
Station efficiency	39.7%[a]
Net electrical power output	1190 MW(e)

[a]Parameter based on optimization for minimum power generating cost for HEU fuel.

exchanger sizing and mechanical design (GA and CE). Toward the end of the commercial plant study, inputs from these areas were factored into the generation of a plant layout drawing, which is shown in Figs. 5-16, 5-17, and 5-18. The main features of the three-loop plant are given in Table 5-11, and a description of the plant primary system is given below.

Based on the utilization of a 3000-MW(t) core design, the commercial plant embodies three PCLs, each rated at 400 MW(e). Each loop consists of a single-shaft gas turbine, a recuperative gas-to-gas heat exchanger, and a precooler (gas-to-water exchanger) for cycle heat rejection. As shown in the plan view of the PCRV in Fig. 5-16, the three PCLs are located symmetrically around and below the central core cavity. The three turbomachines are oriented in a delta arrangement, and the heat exchangers are installed in vertical cavities within the PCRV sidewalls, two for each loop. This orientation of the major components results in a minimum PCRV diameter, which is economically important since the PCRV is the largest cost item in the plant. The elevation views through the PCRV shown in Figs. 5-17 and 5-18 illustrate the helium gas flow path within the primary system. The components are connected by large internal ducts within the PCRV. The horizontal turbomachine cavities are located directly below their associated loop heat exchangers. The recuperator is positioned directly above the turbine exhaust, and the precooler is above the compressor inlet. Three equispaced CACS units are positioned in the PCRV as shown in Fig. 5-16.

The PCRV is longitudinally prestressed by linear tendons. The circumferential prestressing is conventional for the top part of the PCRV, with wire winding in steel-lined channels of precast panels. In the bottom head section of the PCRV, the wire winding is replaced by diagonal tendons.

5.3.6. Commercial Plant Core Design

The core design related activities were directed toward the demonstration plant. Owing to staff/funding limitations, the effort expended

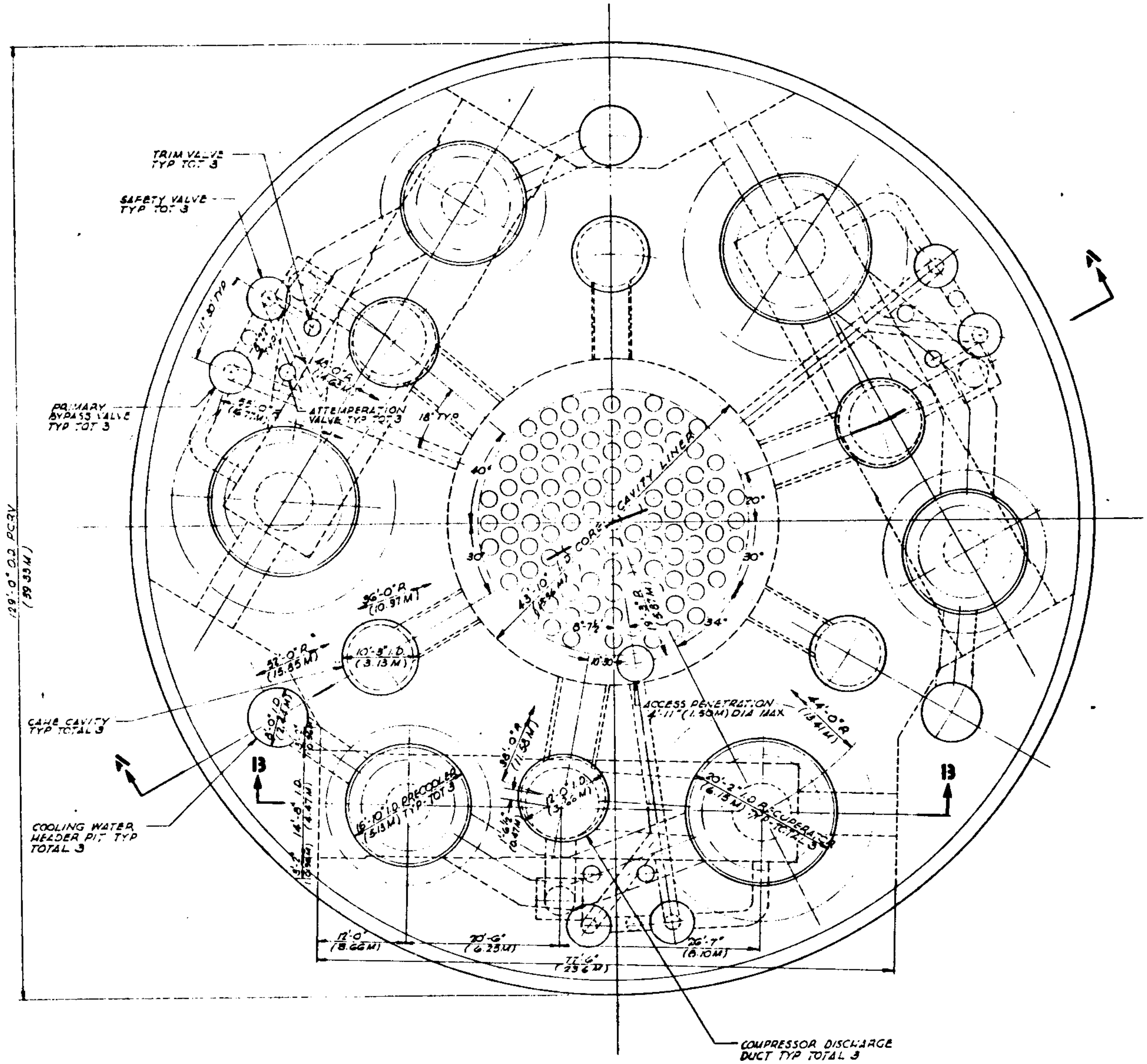


Fig. 5-16. Plan view of PCRV top head for three-loop commercial plant

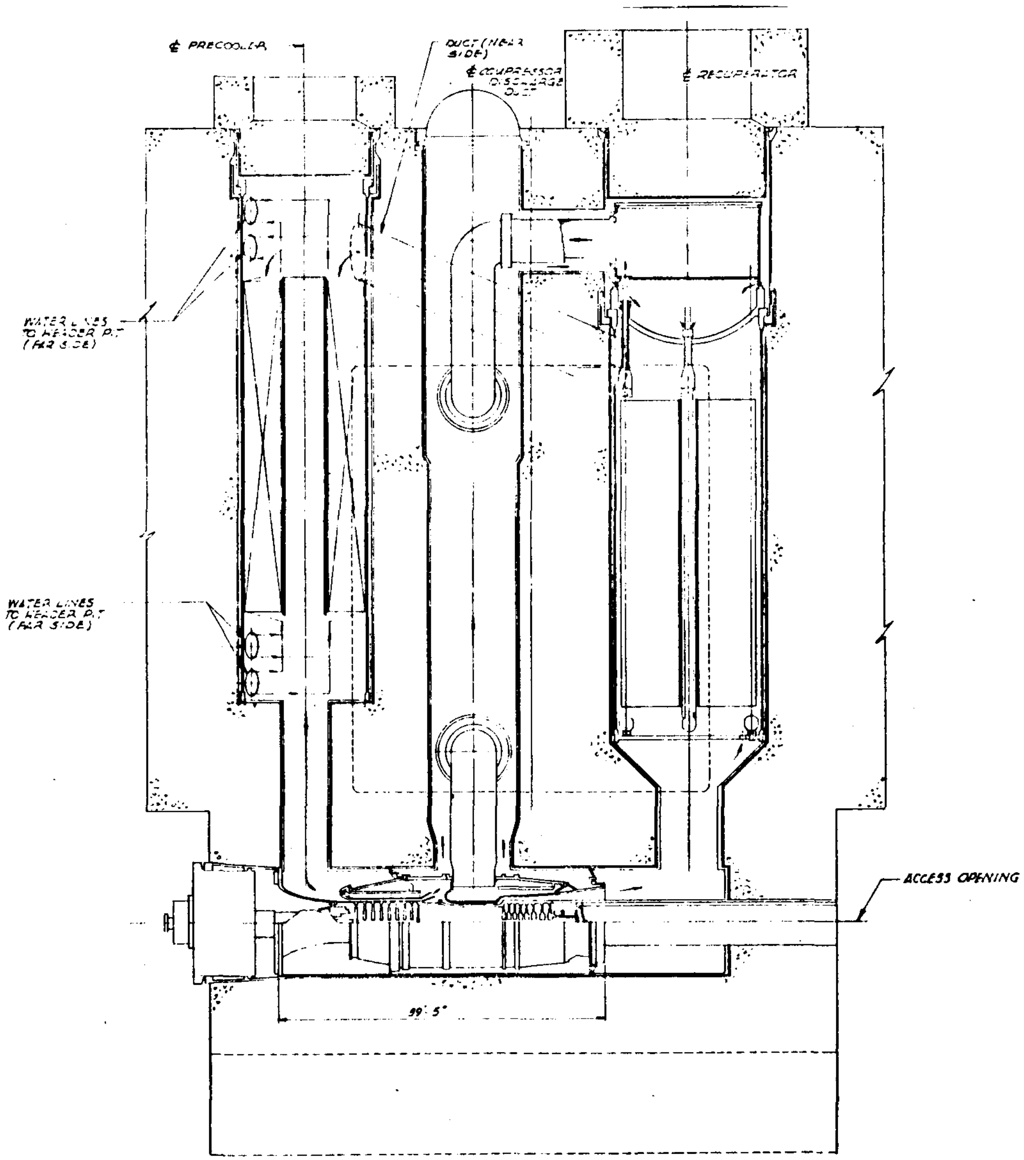


Fig. 5-17. Section B-B of Fig. 5-16

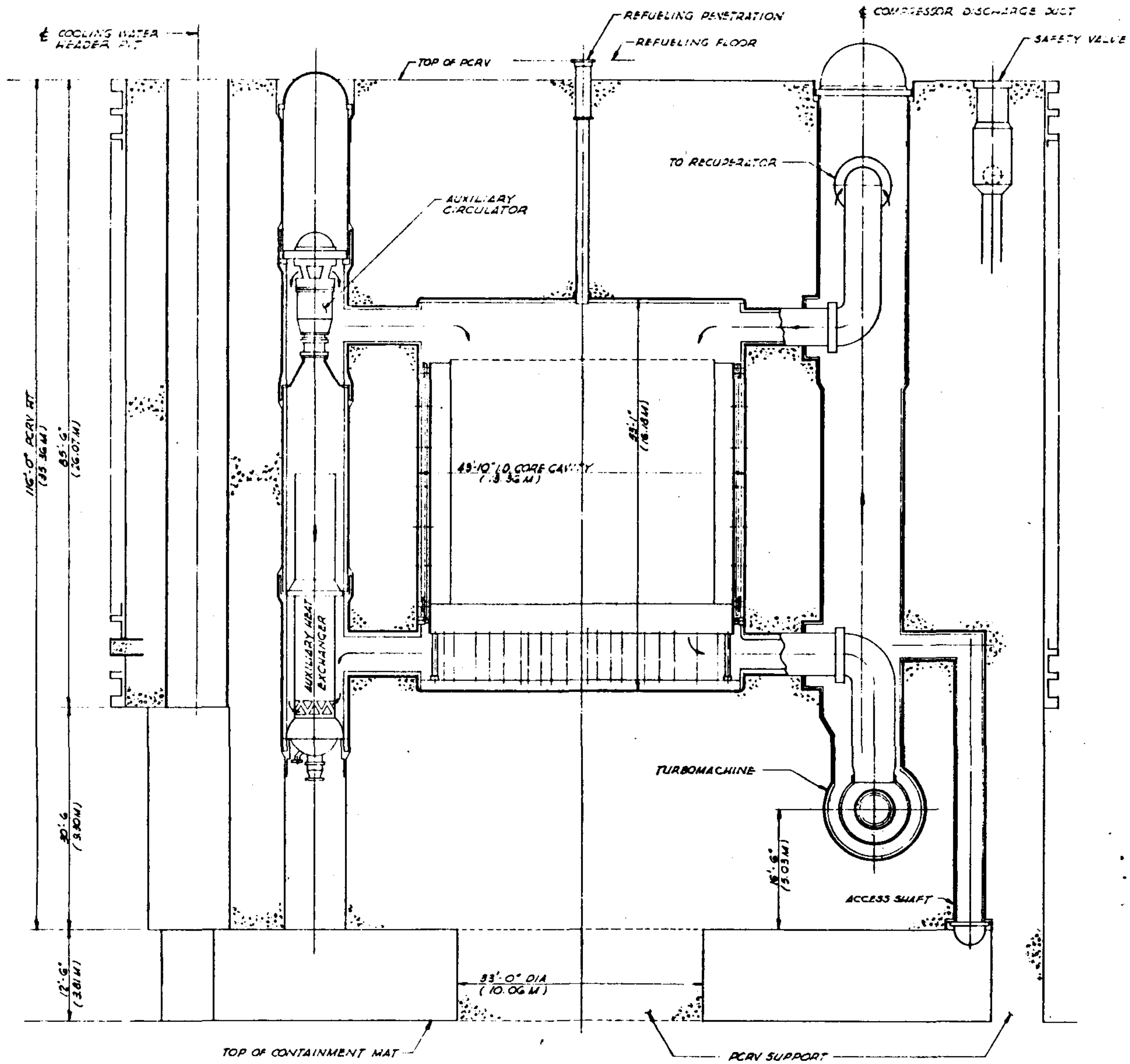


Fig. 5-18. Section A-A of Fig. 5-16

TABLE 5-11

MAIN FEATURES OF THREE-LOOP COMMERCIAL PLANT ALTERNATE

Integrated Direct Cycle Plant

Prismatic Core, Thermal Rating 3000 MW(t)

MEU Fuel (3-yr Fuel Cycle)

Reactor Core Power Density 6.8 W/cc

Non-Intercooled Cycle with High Degree of Recuperation

$$\left.\begin{aligned} P_{max} &= 7.93 \text{ MPa (1150 psia)} \\ T_{max} &= 850^\circ\text{C } (1562^\circ\text{F}) \\ R_{comp} &= 2.5 \\ E_{recup} &= 0.90 \end{aligned}\right\} \text{Plant Efficiency} = 39.7\%$$

Turbomachine Rating 400 MW(e)

Water-Cooled and Insulated Liners Throughout

PCRV Details

Central Core Cavity

Diameter 39.3 m (129 ft)

Height 35.4 m (116 ft)

Delta Turbomachine Position

CACS - 3 x 100% units

Two-Bearing Turbomachine (Single Turbine Inlet Duct)

Man Access Provision to Bearing Cavity Areas

Straight Tube, Modular Recuperator	Exchanger Features Nearly Identical to Demonstration Plant
Helical Bundle Precooler	

Dry-Cooled Plant

Cycle Adaptable to Waste Heat Rankine Bottoming Plant

Emphasis placed on gas flow path simplicity and minimization of primary system pressure loss

Parameters and plant layout based on 1976 optimization study (minimum power generating cost with HEU fuel) on the design of the commercial plant(s) reactor core was considerably less than that devoted to the demonstration plant core. In the initial core layout studies, emphasis was placed on establishing a compact arrangement with a minimum number of partial regions at the periphery. The established 3000-MW(t) core layout is regarded as a satisfactory design because there are no fixed orifice columns or four-column regions.

The design of the prismatic reactor core for the GT-HTGR commercial plant has the following features:

1. MEU fuel.
2. Power density of 6.8 W/cc.
3. Three-year fuel cycle.
4. Fissile TRISO UC_2 fuel particles.
5. Fuel particles contained within 10-row FSV type fuel elements.

Table 5-12 compares the major features of this core design with those of the 1530-MW(t) core. Except for minor differences in inlet helium conditions and flow distribution considerations associated with the number of loops, the core design requirements and considerations for the two-loop plant and the three-loop non-intercooled commercial plant designs are very similar.

TABLE 5-12
COMPARISON OF GT-HTGR CORE DESIGNS

	1530-MW(t) Demonstration Plant	3000-MW(t) Commercial Plant
Number of fuel 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)

6. MISCELLANEOUS CONTROLS AND AUXILIARY SYSTEMS (630102)

6.1. SCOPE

The purpose of this task is to establish the helium bypass valve service system requirements and interfaces.

6.2. SUMMARY

A conceptual design for the helium bypass valve operating system has been developed and is shown in Figs. 6-1 and 6-2. This concept will be the basis for further studies related to the trim, attemperation, and main bypass valves. Since the safety trip valve is a two-position valve, the decision has been made to utilize a different actuation system, and this design study is now under way.

The trim and attemperation valves utilize a hydraulic system at 10.34 MPa (1500 psia) and the main bypass valve has a system pressure of 17.24 MPa (2500 psia). This selection was made in an attempt to keep the hydraulic fluid pressure as low as possible in order to minimize potential leaks and to utilize hardware which is "off the shelf."

In the selection of the actuator design, pneumatic and electric motor operators were evaluated. Conventional pneumatic cylinders and diaphragm operators were found to be too large. Electric motor drives were evaluated but their inherent "fail as is" mode precluded them for this application. High-speed pneumatic turbine drives will be investigated as an alternative in the future.

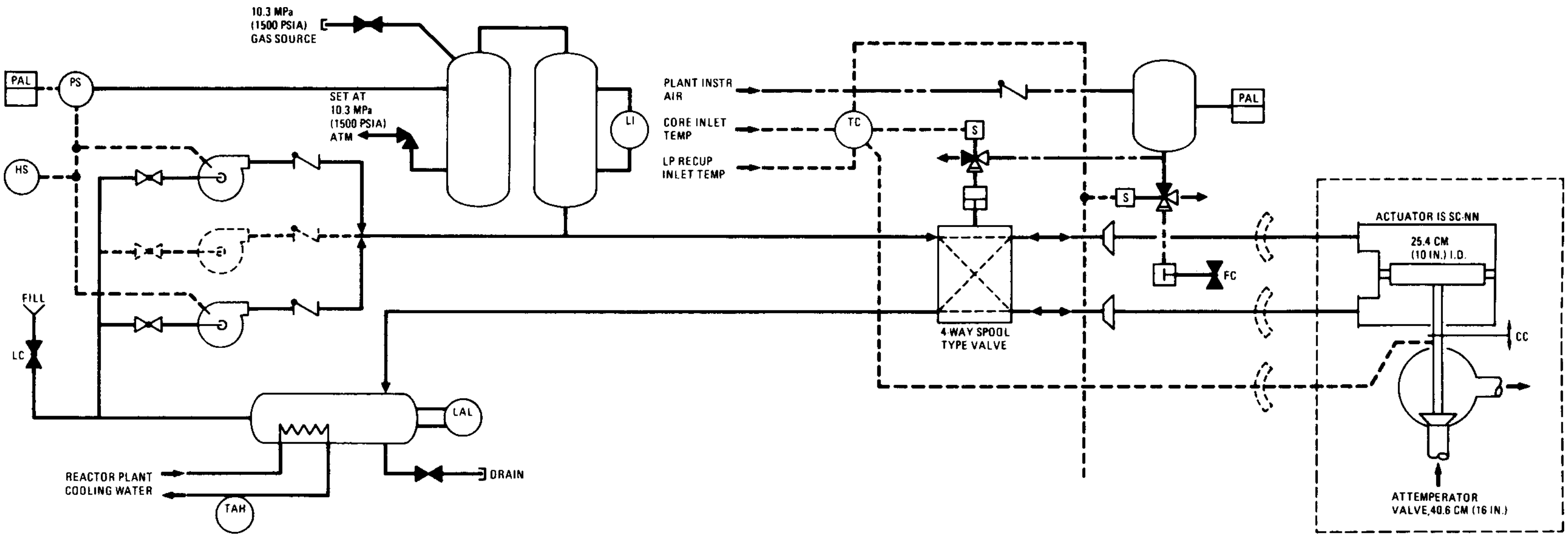


Fig. 6-1. Flow diagram for helium bypass valve actuating system

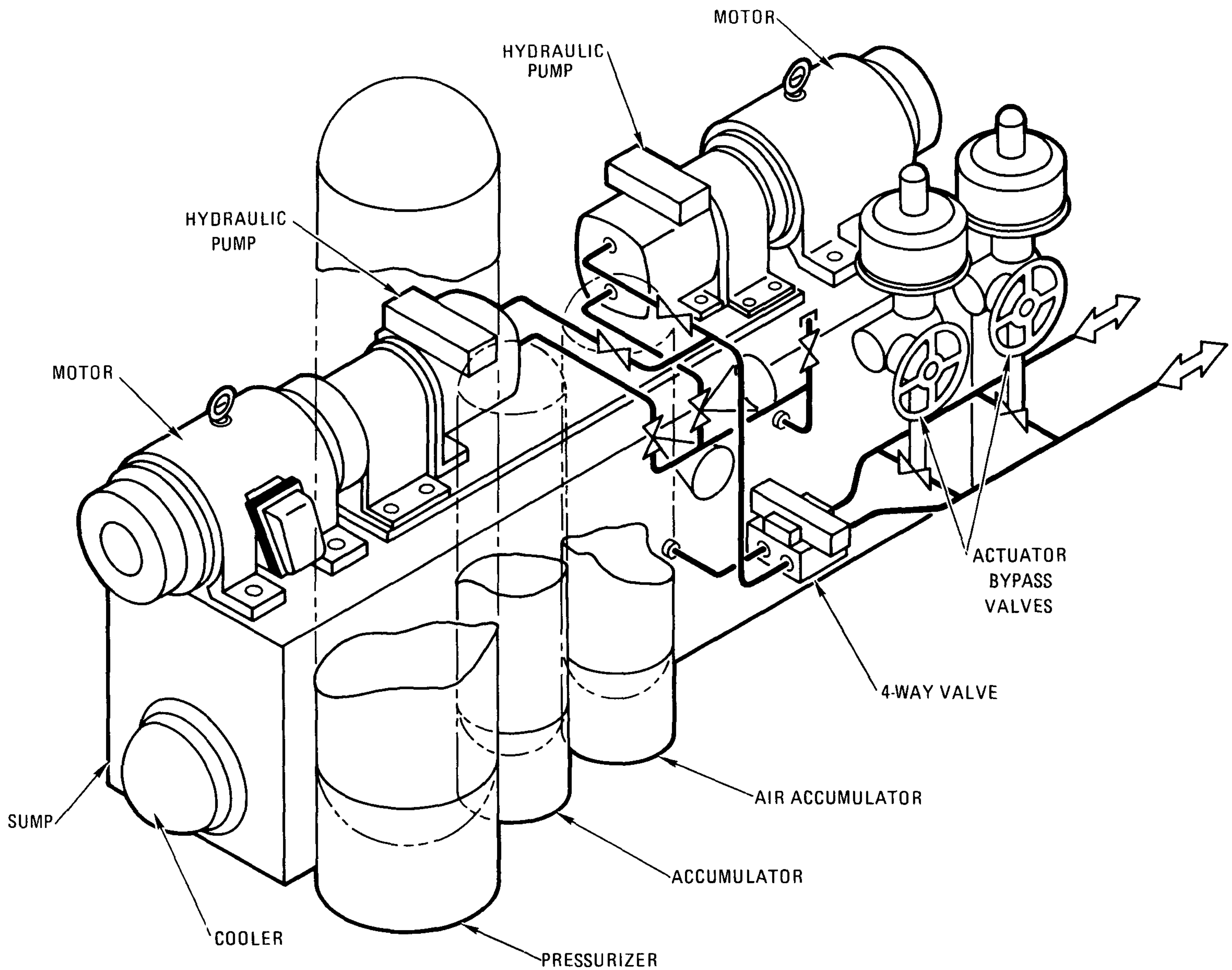


Fig. 6-2. Actuating system for helium bypass valve

7. STRUCTURAL MECHANICS (630103)

7.1. SCOPE

The purpose of this task in FY-79 is to evaluate the effect of core barrel restraints on the seismic response of the core for the warm liner concept and perform PCRV/containment seismic analysis for the HHT plant.

7.2. SUMMARY

An evaluation of proposed GT-HTGR core seismic support and restraint systems has been completed. Two designs were investigated: the GA one-loop GT-HTGR demonstration plant and the German HHT 1637-MW(t) pebble bed demonstration plant. In both designs, the graphite core is contained within a core barrel. The GA design utilizes spring packs attached to the PCRV liner and penetrating the core barrel to restrain the graphite core. The HHT design has no lateral restraint for the core or the core barrel except at the lower end of the barrel. The evaluation was based on the design being compatible with a range of soil sites from soft soil to rock with a safe shutdown earthquake level of 0.3 g. The conclusions are as follows:

1. The GA design with spring packs passing through the core barrel provides essentially the same restraint to the core as the present HTGR lateral core restraint system. The core barrel is separately supported by radial keys to the PCRV liner and should not affect the seismic response of the core.

2. The HHT core barrel design presented to GA for analysis may not provide an adequate seismic restraint for the graphite pebble bed core. The vertical support design is such that the vertical weight of the core does not appear to be connected to the core

barrel. The stiffness of the core barrel has not been evaluated. However, even if it is very rigid, the design of the "dampers" separating the core barrel and the outer ring of reflector blocks cannot be counted on to keep the core totally tight. Even a small amount of gap or looseness is enough to produce a frequency response in the range of amplification which tends to cause uplift of the core barrel. Thus, a sophisticated three-dimensional analysis of the core and core barrel may be required to perform an accurate analysis. A horizontal restraint at the top of the core barrel would alleviate this potential problem.

A mathematical dynamic model of the reactor containment building (RCB)/PCRV for the HHT demonstration plant has been initiated. The mass and stiffness properties of the RCB dome and shell have been calculated. Effort is continuing to generate the mass and stiffness properties of the PCRV, PCRV support, and internal structures and components. Development of the overall dynamic model has been delayed due to the rapidly changing HHT design.

8. SHIELDING ANALYSIS AND DESIGN (630104)

8.1. SCOPE

The purpose of this task is to provide radiation protection and shielding analysis and design support to the GT-HTGR Project.

8.2. SUMMARY

8.2.1. Core Barrel Study

The neutron irradiation effects on the core barrel of the warm liner concept were evaluated. Calculations were made of the neutron fluxes in the side regions of the one-loop demonstration plant. Both the CTD one-dimensional transport/diffusion code and the 2DB two-dimensional diffusion code were used in order to verify the results.

Major differences between the GT-HTGR demonstration plant and the conventional SC-HTGR include the addition of a third row of replaceable side reflector blocks and the attachment of a core barrel around the side thermal shield. An average side reflector thickness of 159 cm (62.6 in.) was calculated for the GT-HTGR.

The material proposed for the core barrel is A-387 steel. At present, there is no neutron damage function for A-387 steel. The initial nil ductility temperature is also unavailable for this material.

Because the thermal flux at the core barrel (1.5×10^{10} n/cm^2-s) is considerably higher than the intermediate or fast flux, the damage to the core barrel will be caused almost entirely by thermal neutrons. The corresponding fluence for 40 yr is $\sim 1.5 \times 10^{19}$ nvt (without uncertainty factor), probably resulting in a differential nil ductility temperature of less than 37.8°C (100°F) for low-temperature irradiation. Irradiation tests would be required to determine the irradiated nil ductility temperature and the damage functions for A-387 steel and to qualify the material for core barrel applications.

8.2.2. Fission Product Study

Basic fission product plateout assumptions were made for the purpose of radiation analysis, shielding design, and decontamination evaluation in the GT-HTGR plant maintenance study.

Total plateout activities were obtained by scaling the values for the SC-HTGR with MEU fuel, except that cesium and strontium releases were calculated expressly for the GT-HTGR. A total plateout surface area of 2.2×10^9 cm^2 was used to obtain uniform surface activities. Plateout distributions through the primary circuit were based on analyses performed earlier. In the interest of conservatism, modifications were made to these earlier distribution curves so that the plateout activity would not drop below the uniform level at any point in the primary circuit.

Preferential plateout factors ranged all the way up to 95 for Ag-110m at the turbine inlet. Only iodine and cesium have low preferential plateout at the turbine inlet.

Computer printouts were prepared tabulating the $\mu Ci/cm^2$ of plateout for 18 important nuclides at eight different locations in the primary circuit for 1-, 3-, 6-, 10-, and 40-yr operation and 0-, 1-, 10-, and 100-day shutdown times. Both Level A and Level B values were generated.

Each power conversion loop (PCL) consists of four helium control valves: the safety, bypass, attemperation, and trim valves. The safety valve is normally closed. Hence, the plateout on the safety valve should be negligible. The helium flow to other valves is expected to be small. For design purposes, the plateout on the bypass, attemperation, and trim valves is assumed to be 10% of the uniform plateout. Similarly, the CAHE system, which is not in the normal flow path of primary helium during operation, is also assumed to contain 10% of the uniform plateout.

9. LICENSING (6302)

9.1. SCOPE

The purpose of this task is to provide design review and licensing positions to ensure that design features comply with regulatory requirements.

9.2. SUMMARY

Various NRC documents were reviewed for identifiable requirements that would make a hot test facility (HTF) mandatory. Various licensing questions or issues which an HTF could resolve were also considered. The following conclusions were drawn:

1. A facility of the magnitude of an HTF is not a requirement. While licensability could be greatly enhanced by data from full-scale tests, NRC requirements can be satisfied by rigorous analyses, component and/or scale tests, and final testing in the reactor facility.

2. It is not prudent to begin the design of the HTF without performing a thorough failure effects and modes analysis (FEMA) of the turbomachine, its control system, and related systems. Without a FEMA, it may be difficult to establish confidence that the needed tests have been identified and the facility will be adequate to perform these tests.

9.3. DISCUSSION

9.3.1. HTF Evaluation

Despite Conclusion 1 above, it should be noted that the NRC places great stock in testing and is particularly cautious about prototype systems and components. Primarily, the NRC is concerned with safety-related equipment, but at this stage of the design, it cannot be said with certainty which systems and components will be designated safety class and/or Seismic Category 1. The NRC is also interested in the ability of the plant to be a reliable source of generating capacity; lack of confidence in plant reliability could have a negative impact on licensability through the NRC's Environmental Report. Furthermore, it has become common at public hearings for intervenors to challenge applications on the basis of environmental evaluations that assume overly optimistic plant availability estimates.

Areas of major importance for facility capability are:

1. Measurement of the transient response of systems and components and the validation of transient response computer codes.

2. Experience with the rotating seal and verification of its performance under transient conditions.

3. Measurement of acoustic forces developed within the operating loop.

4. Evaluation of control system response and stability and machine operating characteristics.

Other facility capability could include the following items, which probably could be accomplished in lesser facilities:

5. Testing of the overspeed protection system.
6. Non-destructive testing of overspeed integrity.

7. Measurement of the flow resistance of the idle machine.
8. Experience with early operational failures.
9. Experience with in-service inspection (ISI).

10. SAFETY AND RELIABILITY (6307)

10.1. SCOPE

The purpose of this task is to:

1. Provide safety assessment of design issues and support the HHT program by reviewing Federal Republic of Germany (FRG) general safety design criteria and HHT safety criteria.

2. Provide availability input to aid in major feature selections, generate preliminary estimates of overall plant availability, provide availability input to incentives reports, and perform availability assessment of intercooling features.

10.2. SUMMARY

As input to the "Primary System Parameter Review Design Report," some qualitative observations relative to temperature and pressure parameters have been prepared. Higher fuel and graphite temperatures for the same fuel system will result in larger fractions of failed fuel and greater release of fission products to the primary coolant system, increasing the duty of the helium purification system and source term of the design basis depressurization accident (DBDA). However, the off-site doses resulting from a DBDA are relatively small compared with postulated accidents, and the increased fuel temperatures are not expected to alter the conclusion regarding the DBDA. Increased fuel temperatures will reduce the margins to fuel particle failure during core heatup events. Higher graphite temperatures will accelerate the reaction with any steam/water which is introduced into the core.

An increase or decrease in the turbine differential temperature will result in a corresponding increase or decrease in the maximum depressurization rate for the design of the CACS ducts.

More detailed information will be provided in the design report referred to above.

Recent RATSAM studies indicate that maximum core outlet plenum and CACS lower duct depressurization rates resulting from a turbine deblading event could be as high as 17.24 MPa/s (2500 psi/sec) (maximum average over any 10-ms period) for the three-loop commercial plant. Depressurization rates were found to be relatively sensitive to the assumed deblading time, the method used for simulating the deblading, and the core outlet plenum volume.

The FRG safety design criteria have been reviewed and comments have been sent to Hochtemperatur Reaktorbau (HRB). The criteria reviewed included testability, improvement measures, radiation exposure guidelines, general definitions, and penetration and closures. Comments on the German criteria were presented, along with information regarding the U.S. criteria pertaining to the subject German criteria.

Work was initiated to establish a preliminary availability model for the reference 3000-MW(t) three-loop plant and to assemble data on component and system reliability, maintenance, and operational constraints. The model and data will be employed to generate a preliminary estimate of GT-HTGR plant availability which may be used to compare the relative availability of competing systems (e.g., SC-HTGR) and to aid in the selection of major design features. In addition to the model work, further investigation of failure rates and repair times, to be used in evaluating the model, is being accomplished. Maintenance philosophies and preliminary procedures are being developed, separately from the availability analyses, which will be used as they become available. New failure rate sources for the GT-HTGR unique turbomachines, heat exchangers, and valves are being sought, and evaluations of data currently in hand are being reviewed.

10.3. DISCUSSION

10.3.1. Depressurization Rates During Turbine Deblading Accident

Depressurization rates for the three-loop commercial plant core outlet plenum have been calculated to range from 5.52 to 24.13 MPa/s (800 to 3500 psi/sec), depending on the assumptions and forcing functions used in the analysis (see Table 10-1). The RATSAM forcing functions are believed to be more realistic than the earlier TUBE forcing functions and appear to be consistent with the Swiss methods of analysis. RATSAM depressurization rates are higher and therefore more conservative than prior rates predicted by TUBE. Thus, it is recommended that for a three-loop commercial plant, a maximum depressurization rate in the core outlet plenum of 17.24 MPa/s (2500 psi/sec) be used for the conceptual plant evaluation. This rate corresponds to an assumed three-loop core outlet plenum volume of 257.7 m^3 (9100 ft^3), a forcing function approximating a sequential stage-by-stage deblading, and a deblading time of 150 msec. As a first approximation this value should be used as the depressurization rate in the CACS lower ducts.

Currently the deblading time is assumed to be 150 ms, so results based on that time are being quoted for use. However, for the HHT project a deblading time of 20 ms is being assumed (one turbomachine revolution). Additional information is being requested from turbomachine vendors to determine the correct assumption.

It should be noted that the maximum depressurization rate will persist for a very short period of time. Lesser rates result when longer time intervals are considered, as shown in Table 10-1, where the maximum depressurization rates in the core outlet plenum of a three-loop plant are given for deblading times of 20 and 150 ms.

Since core outlet plenum (COP) depressurization rates appear to be very sensitive to COP volume, designers have been advised to incorporate appropriate margins to allow for design changes and uncertainties.

TABLE 10-1
DEPRESSURIZATION RATES IN CORE OUTLET PLENUM AND CORE OUTLET DUCT ENTRANCE[a]

Deblading Time (ms)	Forcing Function	Core Outlet Plenum Volume [m^3 (ft^3)]	Core Outlet Plenum Maximum Depressurization Rate [MPa/s (psi/sec)]	Core Outlet Duct Entrance Maximum Depressurization Rate [MPa/s (psi/sec)]
20	K = f(t)	206.7 (7,300)	55.16 (8,000)	208.57 (30,250)
		1,415.9 (50,000)	11.93 (1,730)	148.57 (21,570)
	K = f(t^2)	206.7 (7,300)	51.37 (7,450)	364.88 (52,920)
		339.8 (12,000)	38.13 (5,530)	364.06 (52,800)
		1,415.9 (50,000)	11.93 (1,730)	363.16 (52,670)
150	K = f(t)	206.7 (7,300)	44.82 (6,500)	58.75 (8,520)
		339.8 (12,000)	32.75 (4,750)	54.22 (7,863)
		1,415.9 (50,000)	11.24 (1,630)	45.85 (6,650)
		28,317.0 (10^6)	0.67 (97)	44.61 (6,470)
	K = f(t^2)	206.7 (7,300)	25.24 (3,660)	94.46 (13,700)
		339.8 (12,000)	19.72 (2,860)	94.46 (13,700)
		1,415.9 (50,000)	7.45 (1,109)	93.77 (13,600)
		28,317.0 (10^6)	0.55 (80)	93.77 (13,600)

(a) From RATSAM.

It has been observed that an actual progressive deblading would result in a stage-by-stage decrease in flow resistance. Thus, a sequential deblading model which considers the effect of increased mass flow during the deblading should be used in future analyses. A typical resistance change where $K = f(\dot{m}, t)$ has been calculated. This representation is also amenable to simulating partial deblading accidents, which were found to be more probable than total deblading. It is expected that using such a forcing function will result in COP maximum depressurization rates less than those resulting from the liner deblading approximation $K = f(t)$.

Future work will include a reanalysis of the three-loop plant using dynamic models of the turbomachinery, which have been incorporated in the GT-HTGR version of RATSAM currently under development. Such an analysis should show the effects of continued operation or rundown of the intact loops.

Other depressurization accident sequences will also be analyzed, including compressor deblading and recuperator failure.

11. REACTOR TURBINE SYSTEM/BALANCE OF PLANT (RTS/BOP) INTEGRATION (631001)

11.1. SCOPE

The purpose of this task is to develop the reference plant layout, develop the conceptual design for major BOP systems, and issue a package of information for cost basis.

11.2. SUMMARY

Work commenced on this task in March 1979, and the layout drawings for the 3000-MW(t) reference plant were about 50% complete by the end of the first half of the fiscal year. This work revealed a problem in routing precooler water lines through pipe chases in the PCRV. Modifications to the PCRV design were made to accommodate the piping requirements.

The present design configuration places the main generators inside the containment building and introduces a requirement for high-voltage, high-current electrical penetration of the containment. A review of current designs and discussions with manufacturers are in progress, with initial indications being that a new design will be required. A review is also being made of the applicability of NRC Regulatory Guides on fire protection to these penetrations.

Work items identified for the second half of the fiscal year include completion of the subtasks discussed above and the maintenance/contamination evaluation. The objective of the latter subtask is to establish the relationship between plant maintenance and the contamination level resulting from varying fuel release levels. A program plan was prepared and distributed to the interfacing organizations.

12. SYSTEM DESIGN (631002)

12.1. SCOPE

The purpose of this task is to establish the CACS design criteria and major features for the GT-HTGR reference design, develop and maintain design information as required for the primary coolant system, evaluate the reference plant design primary system parameters holding the turbomachine inlet temperature to 850°C, and perform CODER evaluations of cycle performance versus cost for various plant configurations.

12.2. SUMMARY

12.2.1. CACS Design Criteria

Design criteria for the CACS for a multiloop commercial GT-HTGR plant have been proposed. As one part of this effort, the unique GT-HTGR issues of (1) the design basis for overall core cooling and (2) criteria imposed by the CACS on main loop components were examined. The other part consisted of outlining the remaining CACS criteria in the same format as the equivalent criteria developed for the SC-HTGR plant designs.

12.2.2. One-Loop Demonstration Plant

The performance parameters for the one-loop demonstration plant based on updated $\Delta P/P$ losses were calculated using CODER-2. The results of the calculations show the plant efficiency to be reduced from 40.96% to 39.65%. The one-loop demonstration plant is intercooled and uses the warm liner concept. Comparison with the reference commercial plant (no intercooling and with conventional PCRV liners) efficiency of 39.55% indicates the performance penalty of the warm liner as intercooling is worth an approximately 1.0 to 1.5 percentage point increase in efficiency.

12.2.3. One-Loop Demonstration Plant Parametric Study To Reduce Potential Cesium Release

Fission product release studies showed that cesium release rate is sensitive to power to flow ratio and that dropping the helium inlet temperature from approximately 499°C (930°F) to approximately 443°C (830°F) for the intercooled cycle effectively doubled the cesium release rate. This impacts turbomachinery maintenance capability in particular, since cesium concentrates in the turbine and the lower portion of the recuperator. A parametric study was performed using CODER-2 to determine the impact of forcing the core inlet temperature up to approximately 499°C (930°F) and reducing the power/flow ratio to an adequate level. The parameter study shows that in order to maintain the base core efficiency and to boost the core inlet temperature to 499°C (930°F), the high-pressure compressor pressure ratio must be reduced from 1.75 to approximately 1.45 and the recuperator effectiveness increased from 0.898 to approximately 0.927. The cost effect was not quantified, but it will be significant since the PCRV size is sensitive to the recuperator size.

12.2.4. Two-Loop Intercooled Plant

The performance parameters for the two-loop intercooled plant with conventional liner based on updated $\Delta P/P$ losses were calculated using CODER-2. The results of the calculations show the plant efficiency to be reduced from 41.70% to 41.17%.

12.2.5. Three-Loop Non-Intercooled Plant

The performance parameters for the two-loop intercooled plant with conventional liners based on updated $\Delta P/P$ losses were calculated using updated duct and cavity dimensions and CODER-2. The results showed the plant efficiency to increase slightly from 39.55% to 39.61%.

12.2.6. Primary System Parameter Design Report

The turbine inlet temperature for the GT-HTGR has been at or near 850°C (1562°F) for over 5 yr, during which time considerable discussion has taken place regarding high-temperature component and material limitations for temperatures even lower than 850°C. With the impetus given the GT-HTGR program, a design report has been initiated to accomplish the following:

1. Review the design and material problems associated with the 850°C temperature.

2. Establish a course of action for required design verification and support.

3. Determine economic sensitivity to varying primary system parameters.

13. SYSTEM DYNAMICS (631003)

13.1. SCOPE

The purpose of this task is to analyze transient performance, develop control and PPS requirements, assess and develop system operational requirements, and provide transient requirements for component and subsystem design.

13.2. SUMMARY

Documentation of the 12 critical transients for the two-loop 300-MW(t) alternate commercial plant design has been completed.

Preliminary analysis of the critical component limiting transient is nearly complete for the 300-MW(t) reference commercial plant design.

Analysis of rapid load recovery capability following a drop load event was performed. The results showed an 80% load pickup capability in about 5 s for an early reload and a 60% load pickup capability in about 5 s if the reload is delayed longer than several minutes.

Turbomachine overspeed potential was identified in the GT-HTGR Accident Initiation and Progression Analysis (AIPA) study as being on the GT-HTGR safety risk envelope. This event was also investigated using REALY2 with the objective of reducing the risk by improving the overspeed protection reliability. It was found that the use of attemperation valves as backup for overspeed protection cannot control overspeed, at least in the intercooled turbomachine design because the valves are neither big enough nor located properly in the loop to provide the overspeed protection function. Efforts to improve the plant safety via reduction of turbomachine overspeed potential are expected to continue.

13.3. DISCUSSION

13.3.1. Alternate Commercial Plant Transient Analyses

The GT-HTGR transient performance analysis program, REALY2, was used to investigate the two-loop 3000-MW(t) alternate commercial plant design. Twelve plant transients were analyzed to evaluate plant component design requirements and assess the expected plant operational requirements under current preliminary plant control system (PCS) and PPS specifications. In addition, these transients were reviewed and tentatively classified in accordance with the Nuclear Safety Event Classification System. The results showed that all the analyzed PPS setpoints and plant control and plant component protective actions provide adequate margins for normal as well as upset plant operation. However, detailed evaluation of individual plant component loadings will require further analysis as the plant design is optimized.

13.3.2. Reference Commercial Plant Transient Analysis

The transient analysis program REALY2 is being utilized to simulate six plant transients for the 3000-MW(t) GT-HTGR reference commercial plant design. These transients were selected because they provide some of the expected "worst case" transient loadings on plant components under current preliminary PCS and PPS specifications. Limiting design requirements for several plant components are set by these transient loadings. Table 13-1 presents a summary of plant parameters selected to comparatively describe the transients in this study.

The specific modeling assumptions used for these transients were reviewed and accepted by cognizant organizations for correctness and accuracy and thus provide a set of preliminary design reference transients for inclusion in the Primary System Parameter Review Study.

TABLE 13-1
PARAMETERS USED IN COMMERCIAL PLANT TRANSIENT ANALYSIS[(a)]

Transient Description	Plant Parameters: Max. Core Inlet/Outlet Temp.		Max. Power-to-Flow Ratio	Max. Press./Temp. at LPR[(b)] Inlet		Max. Press. at Compressor Inlet	Max. Rate of Pressure Increase	Max. Flow Rate at LPR Inlet	Max. Turbine Speed	Max. Rate of Pressure Decrease	Max. Precooler Inlet/Outlet Temp.		Max. Recuperator Hot End Metal Temp.	Max. Recuperator Cold End Metal Temp.
	°C (°F)	°C (°F)		kPa (psia)	°C (°F)	kPa (psia)	kPa/s (psi/sec)	10^6 kg/h (10^6 lbm/hr)	rpm	kPa/s (psi/sec)	°C (°F)	°C (°F)	°C (°F)	°C (°F)
Full load - 100% nominal	500 (932)	850 (1562)	1.00	3248 (471)	537 (999)	3185 (462)	--	2.05 (4.53)	3600	--	224 (436)	27 (80)	513 (955)	207 (405)
Single-loop loss of load with overspeed	503 (938)	866 (1590)	1.29	6405 (929)	540 (1004)	6405 (929)	1303 (189)	5.73 (12.65)	4165	434 (63)	241 (466)	32 (90)	527 (980)	221 (430)
Single-loop turbomachine shaft break	503 (938)	864 (1588)	1.32	6433 (933)	541 (1005)	6433 (933)	1262 (183)	5.72 (12.60)	4200	434 (63)	242 (467)	31 (88)	524 (975)	220 (428)
Single-loop total loss of precooling water flow	506 (943)	863 (1586)	1.18	6998 (1015)	582 (1080)	6998 (1051)	1048 (152)	4.58 (10.10)	3600	331 (48)	309 (589)	149 (300)	563 (1045)	288 (550)
Plant loss of load with overspeed	519 (966)	871 (1600)	1.00	4758 (690)	667 (1232)	4758 (690)	1255 (182)	5.45 (12.02)	4170	1179 (171)	237 (458)	31 (87)	546 (1015)	218 (425)
Plant loss of cooling water flow	560 (1040)	872 (1602)	1.07	6102 (885)	704 (1300)	6102 (885)	814 (118)	3.81 (8.40)	3604	531 (77)	299 (570)	147 (297)	596 (1105)	277 (530)
Slow rod runout at design	512 (954)	904 (1660)	1.29	3468 (503)	537 (999)	3399 (493)	--	2.26 (4.99)	3600	--	245 (473)	30 (87)	520 (968)	218 (425)

(a) Underlined numbers are the highest values predicted for given parameters.

(b) Low-pressure recuperator.

13.3.3. Rapid Recovery from Drop Load Conditions

In the event of total load rejection (TLR), such as might occur during a grid upset, the ability of a plant to quickly reload (even in minutes) could be of significant benefit. The GT-HTGR initially responds to TLR by rapid opening of bypass valves and, after a short-duration overspeed, a recovery and hold at design speed (standby condition). Subsequent actions would be a function of the particular utility and power grid needs. These actions might include automatic or manually initiated reduction of system temperature and/or system inventory. The options may be chosen dependent on time and/or other parameters to provide flexibility of action. Until longer-term (temperature or inventory change) action has been taken, the GT-HTGR is, in its standby condition, potentially able to rapidly reload once synchronization and breaker reclosure have been accomplished. Full temperature, synchronous speed, and relatively high thermal power provide a unique condition for fast reloading.

While normal load reductions will automatically introduce reduction of temperature and/or inventory to obtain efficient operation, the response to TLR may be maintained by bypass control alone. If the standby condition is maintained strictly by bypass control, the rejected heat load will be relatively high. It is, in part, the transfer of the power going into rejected heat which enhances load pickup. After several minutes of operation, the higher rejected heat load will be reflected in a slightly elevated return water temperature to the precooler inlet. If load is recovered prior to the increase in return water temperature, a capacity to attain 100% load will exist. Some subsequent temporary reduction in maximum load will occur as the higher level of rejected heat is finally reflected in the return water temperature. Conversely, if the reloading is not done until the return water temperature has risen, the initial maximum load will be limited by the cycle bottom temperature Again, as the reloading (and hence lower rejected heat) is reflected around the CWS loop, the return temperature will drop and allow a recovery to full load.

Two cases were evaluated for a preliminary assessment of the rapid reloading condition. The first case was a sequence of TLR to standby condition and, after resynchronization at approximately 2.5 min, a step load demand to full load. The second case was an approximation of reload from conditions which would exist (higher return water temperature and hence higher gas temperatures in the low-temperature region of the loop) following a prolonged hold in standby. Both cases represent extreme reload rates to assess maximum capability, and it would not be anticipated that load demand would simply be stepped to 100%. The actual reload rate would probably depend to a large degree on the existing grid conditions and would be controlled by the operator.

The results of the study show that a unique and potentially very useful capability may be provided by the GT-HTGR. The initial load pickup of $\sim$80% can be obtained in 5 s for an early reload, and $\sim$60% load pickup can be achieved in 5 s for the late reload subsequent to resynchronization and followed by a somewhat slower recovery to full load.

13.3.4. Turbomachine Overspeed Study

The AIPA study considered a sequence of events initiated by loss of offsite power (LOSP), followed by failure of the control system to keep all machines at normal speed and then failure of the primary and backup protective valves to prevent overspeed. In the present study, the PPS was considered to fail as in the AIPA sequence. However, a backup to the redundant safety bypass valve was provided. It was assumed that a backup overspeed detection would be used to open the attemperation valve in a further effort to prevent overspeed. Improvement of the detection reliability has also been investigated.

The attemperation valve is smaller and has one-quarter the flow area of the safety bypass valve. Also, it is located downstream of the low-pressure compressor for other control purposes. Therefore, it has about one-third the pressure drop of the safety bypass valve, which is located downstream of the high-pressure compressor.

The effects of the backup measure were not enough to prevent overspeed, which was nearly 130% within 5 s. Table 13-2 shows the results and compares them with the results that are expected when the safety bypass valve functions normally. The rapid reduction in recuperator temperature, which is an added problem, is due to the turbine exit temperature not increasing as well as the cold flow introduced by the attemperation valve opening. The improper bypass action is again responsible.

TABLE 13-2
RESULTS OF TURBOMACHINE OVERSPEED STUDY

	Event 1 (Safety Bypass Valve Operates Correctly as Overspeed Backup)	Event 2 (Safety Bypass Fails and Attemperation Valve Only Provides Overspeed Backup)
Bypass flow rate (2.5 s)	1000 kg/s (8×10^6 lb/hr)	164 kg/s (1.3×10^6 lb/hr)
Overspeed condition (2.5 s)	113% (peak)	119% (rising)
Overspeed condition (5 s)	100%	128% (rising)
Low-pressure recuperator inlet temp.	486°C (906°F)	410°C (770°F)
Turbine exit temp. (5 s)	648°C (1199°F)	472°C (881°F)

14. PCRV LINERS, PENETRATIONS, AND CLOSURES (631104)

14.1. SCOPE

The purpose of this task is to provide the conceptual design of the two-loop and three-loop reference plants and to evaluate the warm liner concept.

14.2. SUMMARY

Drawings of the conceptual liner design for the two-loop and three-loop reference plants were completed. These drawings, shown in Figs. 14-1 through 14-4, were used to define the technical basis for initial conceptual liner cost estimates. The design was based on existing liner configurations except for the turbomachine cavity liner in the two-loop plant. The turbomachine cavity liner configuration was developed during the reporting period, and a separate drawing was made to depict its arrangement.

As part of a continuing effort on the use of warm liners, problems associated with the design and installation of cold concentric ducts were identified. These ducts are used between the turbomachine and intercooler and, in the GA warm liner design, between the precooler and recuperator. A design of a possible leak detection-collection system was also initiated as an alternative to the warm liner.

A thermal analysis was performed to investigate the consequences of a dead flow condition for the gas which is channeled to flow between the core barrel lid and the top cap liner in the warm liner alternate version of the GT-HTGR. Since there is a tendency for dead flow conditions to develop in the head regions and since it was expected that the elevated liner temperatures, which result from a dead flow condition, would cause

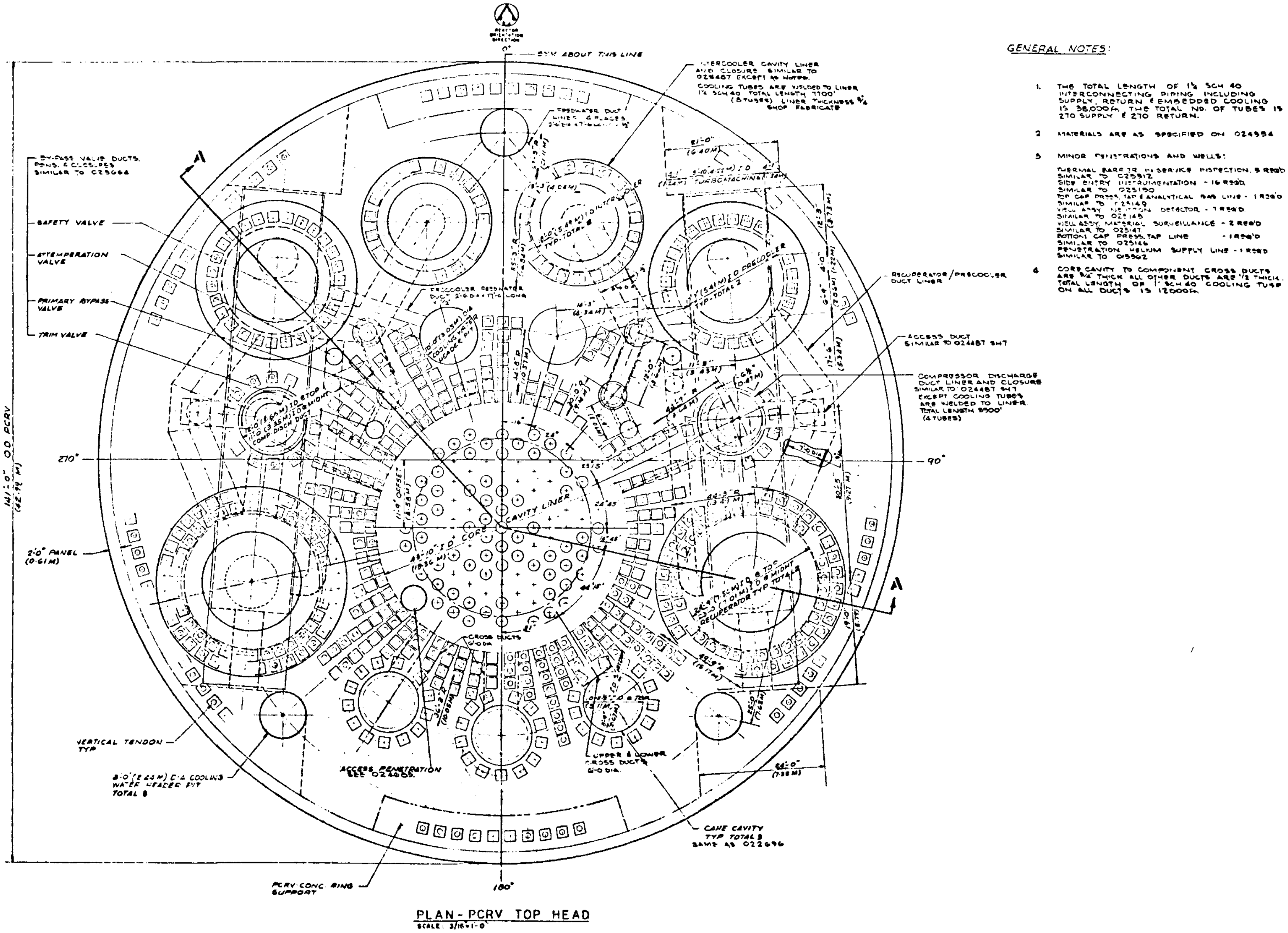


Fig. 14-1. Conceptual design of PCRV liner for two-loop GT-HTGR

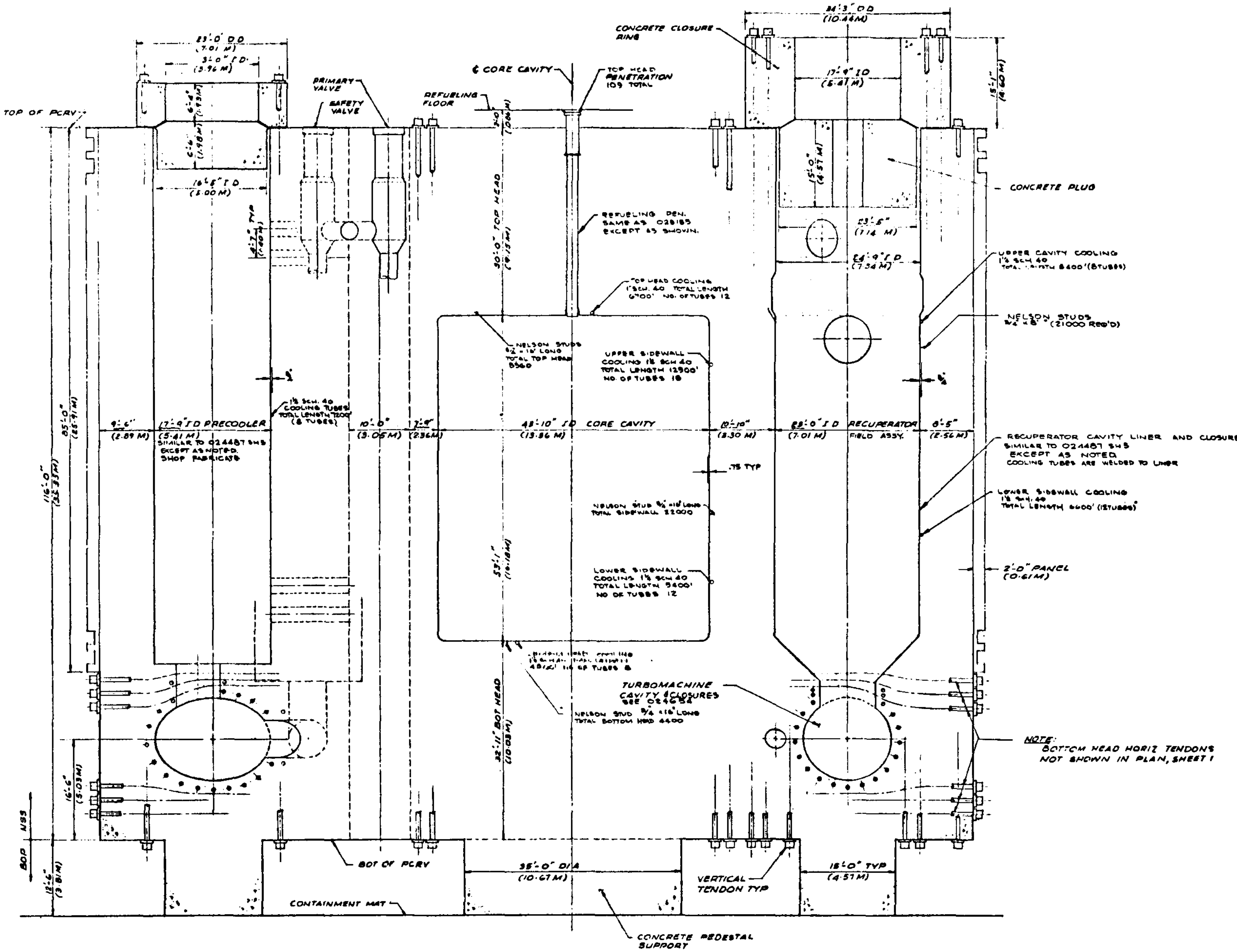


Fig. 14-2. Section A-A of Fig. 14-1

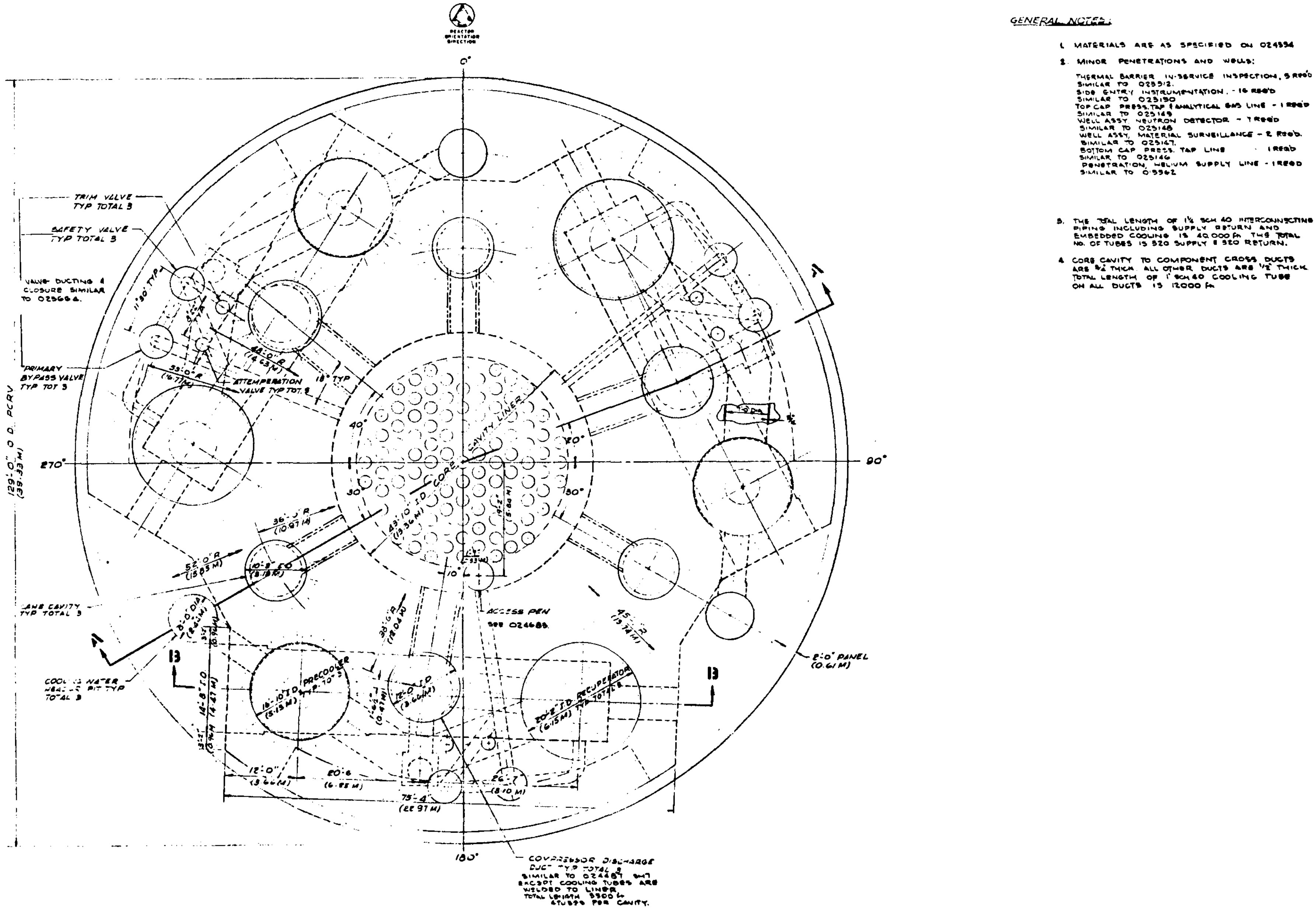


Fig. 14-3. Conceptual design of PCRV liner for three-loop GT-HTGR

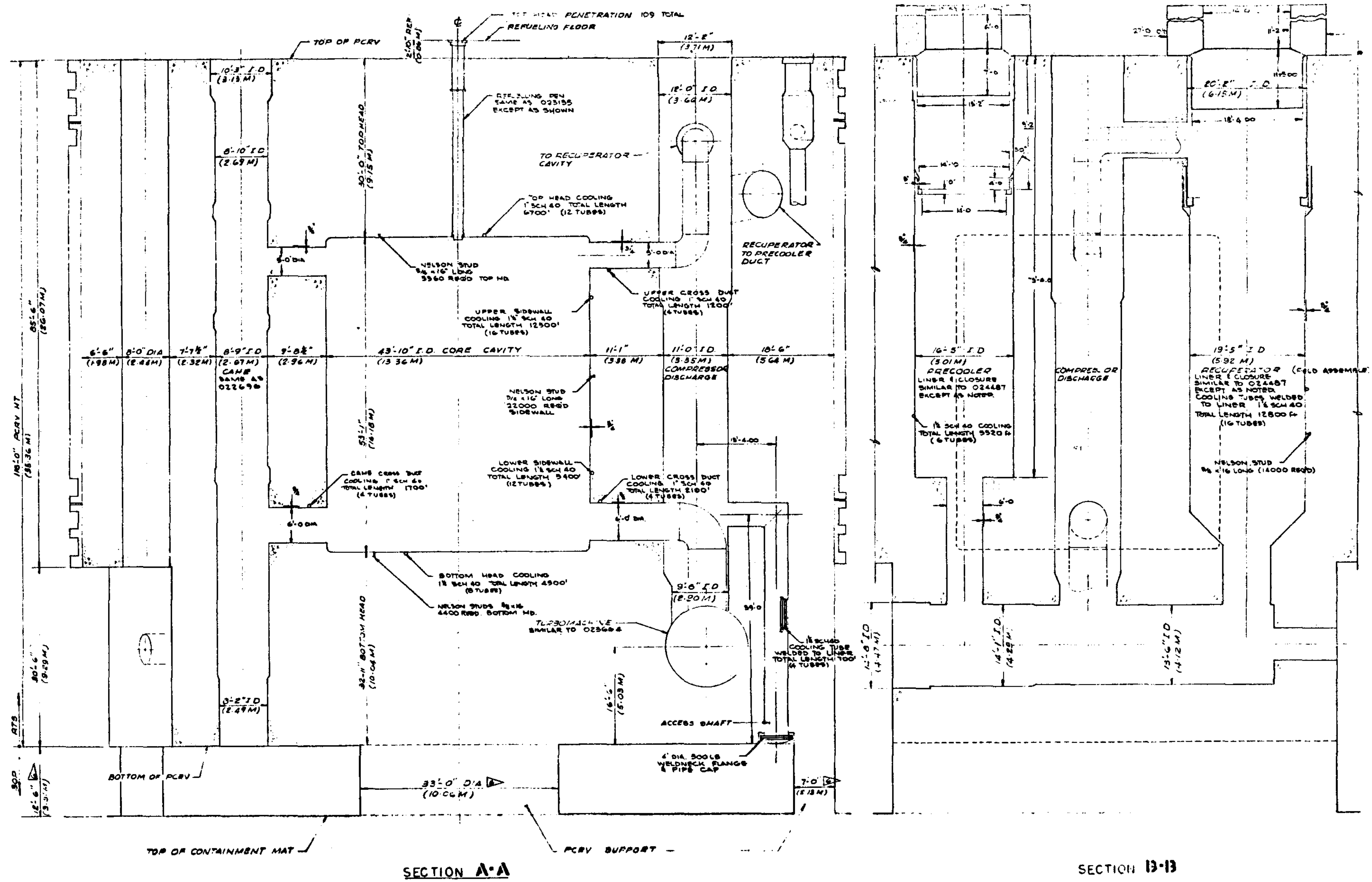


Fig. 14-4. Sections A-A and B-B of Fig. 14-3

more severe consequences at a penetration/liner junction, the intersection of the core cavity liner with a refueling penetration was chosen for this analysis. The results of the analysis indicate that dead flow conditions would cause significantly higher liner and insulating concrete temperatures than would occur under expected flow conditions. The bulk concrete temperatures would exceed the allowables of the ASME Code, Section III, Division 2. A stress analysis of the liner/penetration junction for cyclic conditions has been initiated to predict fatigue life under postulated dead flow conditions.

15. PCRV STRUCTURES (631105)

15.1. SCOPE

The purpose of this task is to develop PCRV design criteria, review the HHT PCRV design, and develop the two-loop/three-loop PCRV conceptual arrangement.

15.2. SUMMARY

A decision has made that the GT-HTGR PCRV will be designed on the basis of a multi-pressure vessel to minimize the PCRV size.

The HHT design reveals that the bottom section of the PCRV makes it very difficult to accommodate the required prestressing. Detailed tendon interface evaluation is required to establish the feasibility of the proposed concept.

Conceptual PCRV arrangement drawings of the two-loop and three-loop GT-HTGR reference plant for cost estimating purposes were completed.

15.3. DISCUSSION

The PCRV size and prestressing requirements are strongly influenced by the design conditions, the complexity of component arrangement, and the resulting tendon layout. The multi-pressure conditions in the PCRV cavities, the effect of differential pressure on inner ligaments, and the proof test pressure specification were major considerations in establishing the PCRV design requirements for the initial sizing of the vessel. The effect of pressure relief settings on the PCRV cavity pressures was also addressed. Pressure relief settings as proposed for the GT-HTGR have been based on the maximum operating pressure at the compressor outlet with pressure margins included for the maximum turbine overspeed transient. Pressure relief caused by low-probability events at the compressor outlet does not provide a design basis representative of the overall structural response of the PCRV. From the standpoint of PCRV design, more realistic overpressure relief set points can be established from the maximum operating pressure in the core cavity for the high-pressure region and from the equilibrium pressure in the side cavities for the low-pressure region. A reassessment of the pressure relief setting should be made when the internal pressure relief system for the GT-HTGR reference plant is better defined. With available information on system pressures, it would appear that a single uniform proof test pressure corresponding to the highest equilibrium pressure can satisfy the purpose of the structural acceptance test and meet the intent of the ASME Code, Section CB-6000.

The complexity of component arrangement in the PCRV bottom head for the GT-HTGR introduces problems in the layout of tendons to produce the required prestressing. A preliminary study of the HHT tendon and component layouts in the turbomachine region of the PCRV bottom head showed that it is extremely difficult, if not practically infeasible, to provide the required prestressing tendons based on the HHT tendon layout. An alternative layout scheme for the bottom head prestressing for the HHT 600-MW(e) plant was completed as shown in Fig. 15-1. The proposed scheme incorporates sufficient horizontal straight and circumferential tendons in the bottom head to resist the turbine and other heat exchanger cavity pressures. Detailed tendon interference study and analysis to confirm the effectiveness of the proposed prestressing scheme are required before complete feasibility can be established.

Considerable technical support was provided to establish an up-to-date version of the PCRV subroutine in the CODER program for the reference three-loop non-intercooled plant. The PCRV algorithm for the delta arrangement was generated for use in the primary system parameter review study for a reactor outlet temperature of 850°C (1562°F).

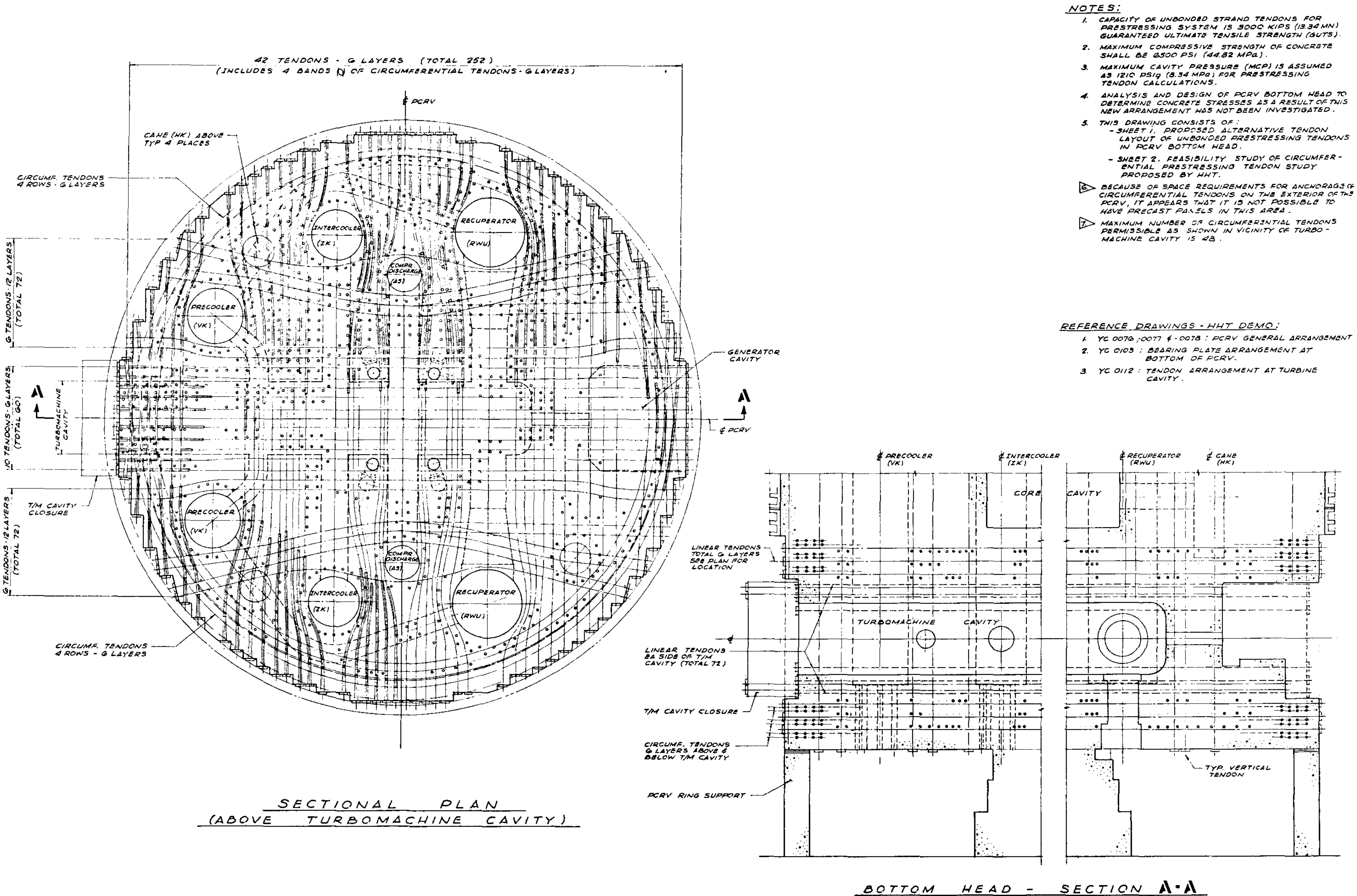


Fig. 15-1. Alternative tendon layout for 600-MW(e) HHT demonstration plant (sheet 1 of 2)

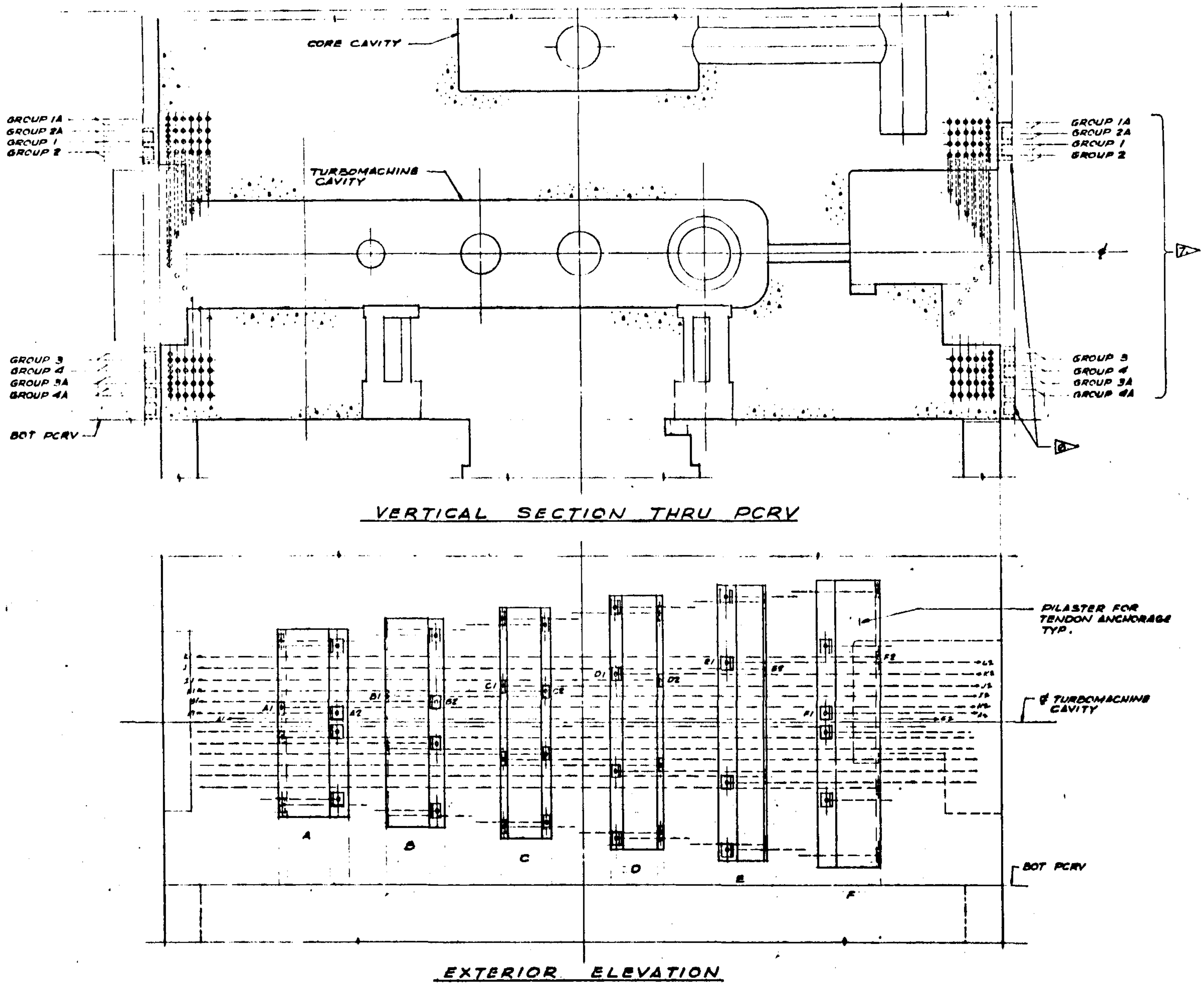


Fig. 15-1. Alternative tendon layout for 600-MW(e) HHT demonstration plant (sheet 2 of 2)

The conceptual PCRV for the two-loop GT-HTGR, starting with the "B-2A" configuration, resulted in the final layout shown in Fig. 15-2. Compared with the demonstration plant design, the two-loop plant PCRV requires generally smaller component cavity diameters and has considerably simpler primary system gas flow paths because the need to provide flow annuli around the components for the warm liner concept has been obviated. An additional PCRV simplification resulted from the smaller size and straightforwardness of the CACS as compared with that which would have been required to accommodate core cooling associated with the warm liner concept. The collective impact of these considerations is a two-loop plant PCRV with a diameter of 42.7 m (140 ft) and a height of 35.4 m (116 ft). Although the two-loop plant PCRV size reflects considerable economy of scale relative to the PCRV for the GA demonstration plant, it does not yet compare favorably with its competing concept, the three-loop, non-intercooled reference commercial plant, which has a PCRV diameter of 39.3 m (129 ft). Consistent with other GT-HTGR plant PCRV designs, the two-loop plant PCRV height was governed by the recuperator height combined with the elevation of the turbomachine in the bottom head, where space must be provided for horizontal tendons surrounding the turbomachine cavity and for diagonal tensioning between the turbomachine cavity diameter and the bottom of the PCRV. The orientation of the vertical compressor discharge cavity also has a potential influence on PCRV diameter. For this study a vertical, straight compressor discharge cavity orientation was adopted from considerations of core hot duct replaceability. The PCRV support concept is based on a central concrete foundation under the core cavity, two partial ring supports at the outer periphery, and two support pads underneath the turbomachine cavities. This approach provides the space required underneath the PCRV for tendon stressing while acknowledging ASME Code requirements for accessibility. Three 2.44-m (8-ft) diameter reactor plant cooling water system pits and two 3.05-m (10-ft) diameter pipe chase cavities are provided in this PCRV design to accommodate the required water plumbing. It is intended that each pipe chase pit will contain the inlet and outlet water piping for both a precooler and an intercooler. While current layout work has not identified any infeasible aspects of this "shared pit" approach, it should be regarded

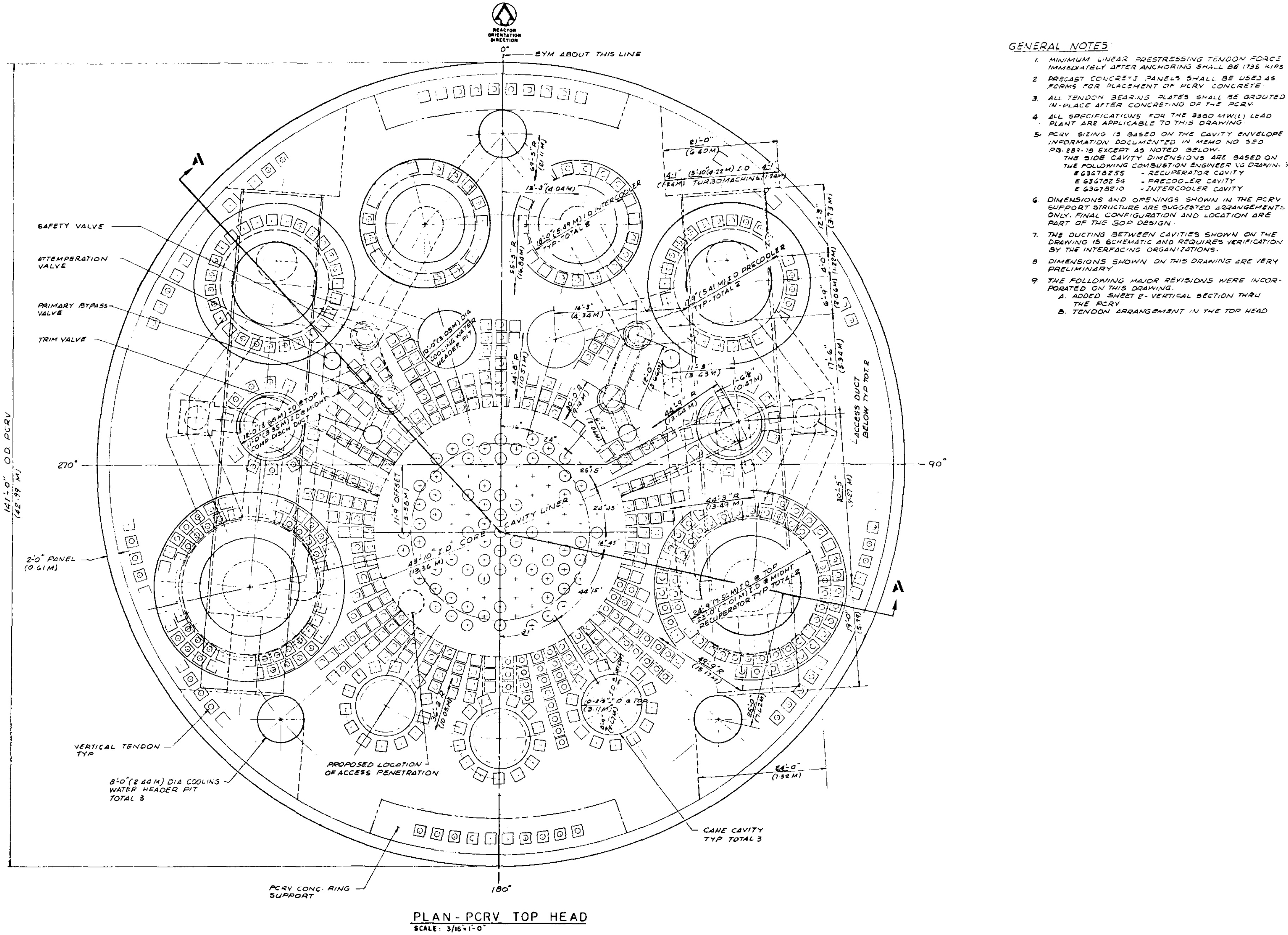
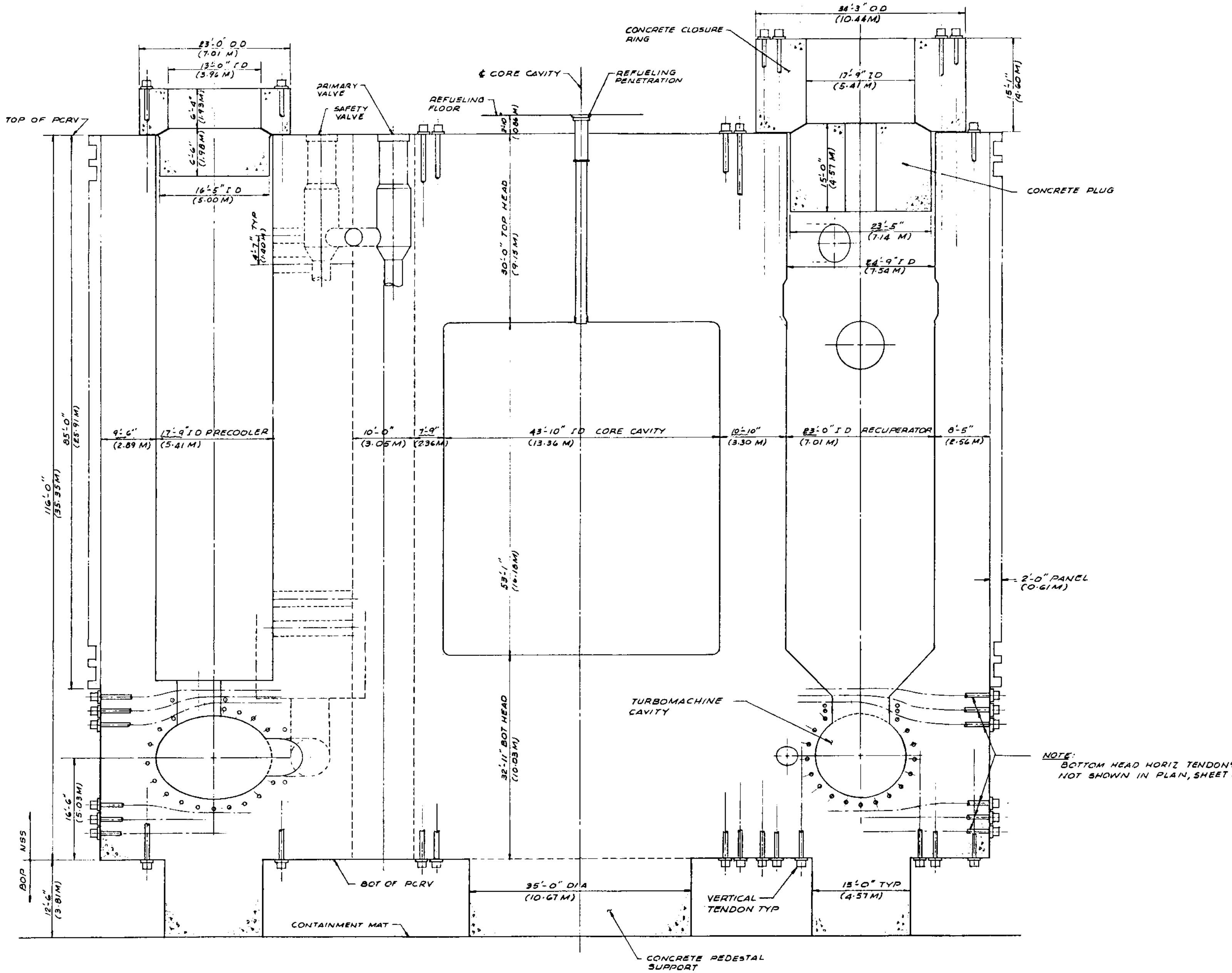


Fig. 15-2. Conceptual PCRV for two-loop GT-HTGR commercial plant (sheet 1 of 2)

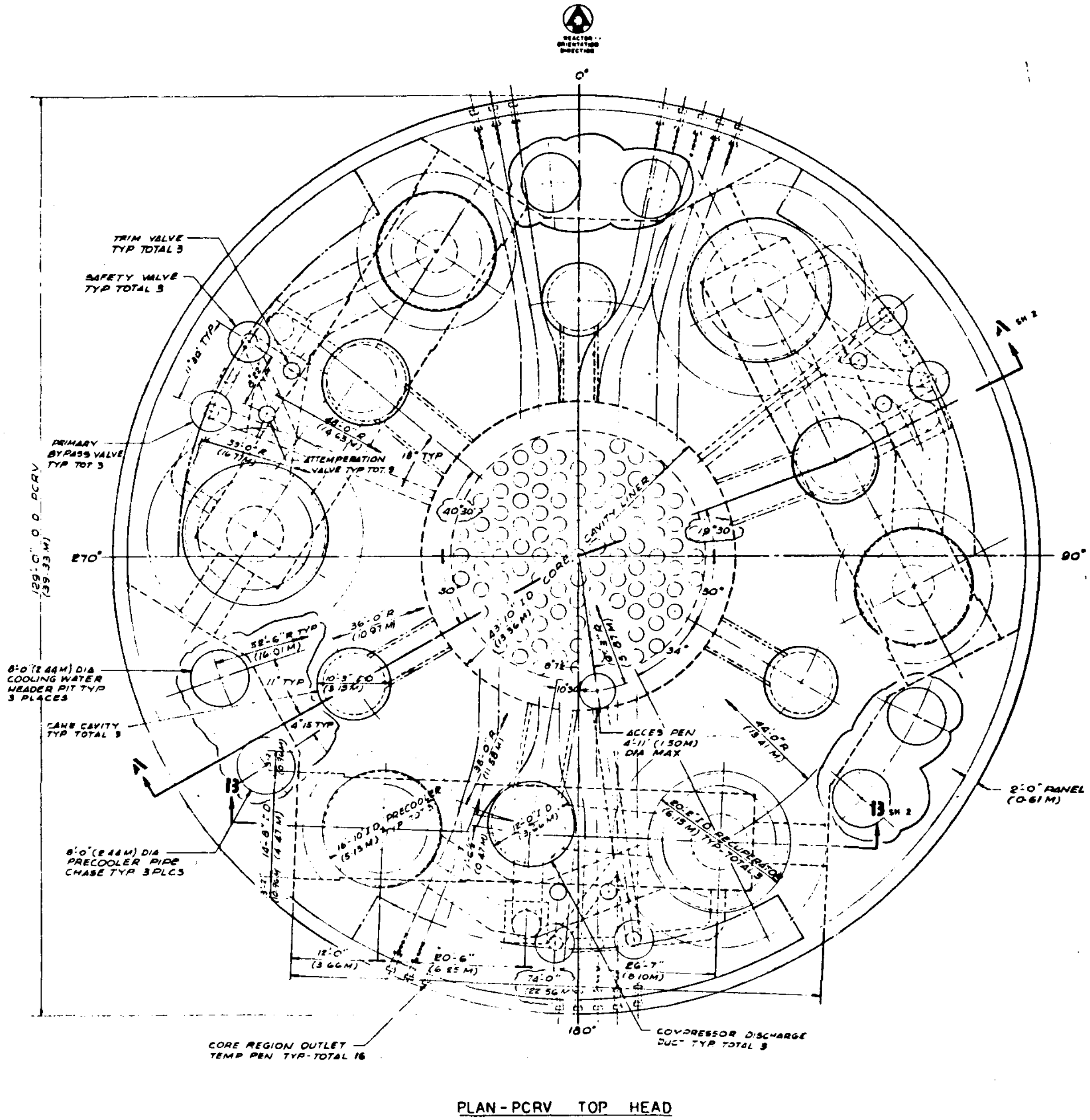


VERTICAL SECTION A-A

Fig. 15-2. Conceptual PCRV for two-loop GT-HTGR commercial plant (sheet 2 of 2) as a provisional possibility until specific design/mockup studies establish its viability.

The PCRV concept for the three-loop, non-intercooled reference commercial plant embodies conventionally insulated and water-cooled liners throughout the entire primary system and is based on a PCRV design approach generally similar to that employed for SC-HTGR PCRV design, except that circumferential prestressing has been replaced by linear horizontal tendons in the area of the turbomachine cavities. Structural analysis and PCRV layout work were performed to confirm the general PCRV conceptual design, culminating in the arrangement shown in Fig. 15-3.

Compared with other commercial GT-HTGR plant arrangements studied previously, including the two-loop intercooled alternate concept investigated in 1978, this three-loop plant arrangement results in the smallest PCRV. This efficient utilization of PCRV structure arises mainly from the tangential orientation of the three turbomachines about the centrally located core to achieve the so-called "delta" configuration, which is ideally suited to accommodate the combination of the 1000-MW(t) loop rating with non-intercooled turbomachines. The collective impact of these considerations is a three-loop plant PCRV with a diameter of 39.3 m (129 ft) and a height of 35.4 m (116 ft), a structure that is 3.35 m (11 ft) smaller than that of the competing alternate two-loop intercooled plant concept. Consistent with other GT-HTGR plant designs, similar layout and stress requirements control the PCRV sizing.



PLAN - PCRV TOP HEAD

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(t) LEAD PLANT ARE APPLICABLE TO THIS DRAWING.
5. DIMENSIONS SHOWN ON THIS DRAWING ARE VERY PRELIMINARY.
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 THE DRAWING IS SCHEMATIC AND REQUIRES VERIFICATION BY THE INTERFACING ORGANIZATIONS.
8. PCRV SIZING IS BASED ON THE INFORMATION AS FOLLOWS:
   - A CORE CAVITY (DIA ONLY) — GA 024490
   - B RECUPERATOR CAVITY (DIA ONLY) CE E63678260-1
   - C PRECOOLER CAVITY (DIA ONLY) - CE E63678287-1
   - D COMP DISCHARGE DUCT (DIA ONLY) - GA 024573
   - E CAHE CAVITY (DIA ONLY) — GA 023651
   - F TURBOMACHINE CAVITY (DIA ONLY) - GA 023664
   - G MAXIMUM CAVITY PRESSURE (PSIG) USED IN THE DESIGN:

	INDIVIDUAL	EQUILIBRIUM
CORE, CAHE, COMP DISCHARGE DUCT & RECUPERATOR (HP)	1185	900
PRECOOLER, RECUPERATOR (LP) & TURBOMACHINE CAVITY	385	900

9. ADDITIONAL PRECOOLER PIPE CHASES (3) WERE INCORPORATED AS PER MEMO NO SED PB 16 78

QUANTITIES:

1. CONCRETE (6500 PSI)
   - PCRV — 38,850 CY
   - PRECAST PANELS — 1,590 CY

	CLOS PLUG	CLOS RING
RECUPERATOR	340 CY	710 CY
PRECOOLER	122 CY	167 CY

2. REINFORCEMENT (ASTM A 615 GR 60):
   - PCRV — 2,202 TONS
   - PRECAST PANELS — 223 TONS

	CLOS PLUG	CLOS RING
RECUPERATOR	33 TONS	118 TONS
PRECOOLER	12 TONS	28 TONS

   NO OF MECHANICAL SPLICES IN PCRV — 8,800

3. PRESTRESSING-LPS:
   NO OF 60-½ Ø STRAND TENDONS — 814
   LENGTH OF TENDONS (BETWEEN CONCRETE SURFACES)*

	QUANTITY	LENGTH
VERTICAL TENDONS:		
1 CORE CAHE & COMP. DISCH	367	118 LF
2 RECUPERATOR	150	132 LF
3 PRECOOLER	33	125 LF
4 PERIPHERAL	60	126 LF
HORIZONTAL TENDONS:		
1 TURBOMACHINE	60	65 LF
2 BOTTOM HEAD	144	124 LF (AVG)

   * ALLOW 5 LF PER TENDON FOR TENSIONING

4. PRESSTRESSING - CPS.
   - A PCRV
     - ½" Ø STRAND - TOTAL LENGTH — 8.3 x 10^6 LF
     - NO. OF STRAND LAYERS — 391
     - NO OF 18" PRECAST PANELS — 214
     - NO. OF LOAD MONITOR PANELS — 6
   - B CLOSURE RINGS
     - ⅜" Ø STRAND - TOTAL LENGTH — 1.5 x 10^5 LF (INCLUDES STRAND FOR RECUPERATOR AND PRECOOLER)

5. INSTRUMENTATION:
   - NO OF LPS LOAD CELLS:
     - 1 VERTICALS — 30
     - 2 HORIZONTALS — 10
   - NO OF CPS LOAD CELLS — 6
   - NO OF SENSORS:
     - 1. PCRV — 450
     - 2. LINER — 460
   - ADD SPARE SENSORS TO ABOVE: PCRV - 10%; LINER - 15%

Fig. 15-3. Conceptual PCRV for three-loop non-intercooled GT-HTGR commercial plant (sheet 1 of 2)

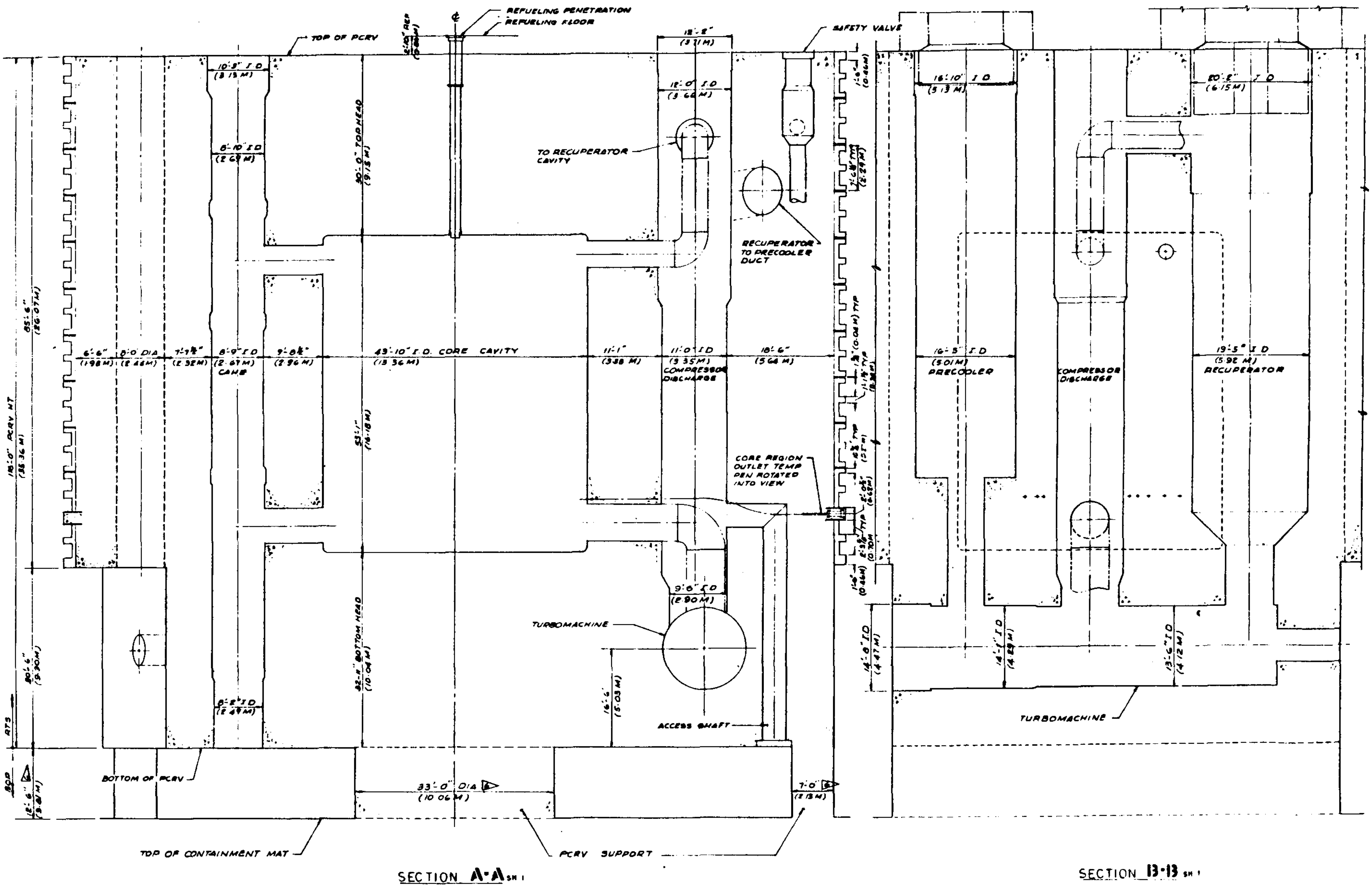


Fig. 15-3. Conceptual PCRV for three-loop non-intercooled GT-HTGR commercial plant (sheet 2 of 2)

16. THERMAL BARRIER (631106)

16.1. SCOPE

The purpose of this task is to provide the two-loop and three-loop reference GT-HTGR conceptual layout, evaluate the HHT hot duct design, and conduct higher-temperature thermal barrier design.

16.2. SUMMARY

Two-loop and three-loop reference GT-HTGR plant conceptual layout drawings for cost estimating purposes were completed.

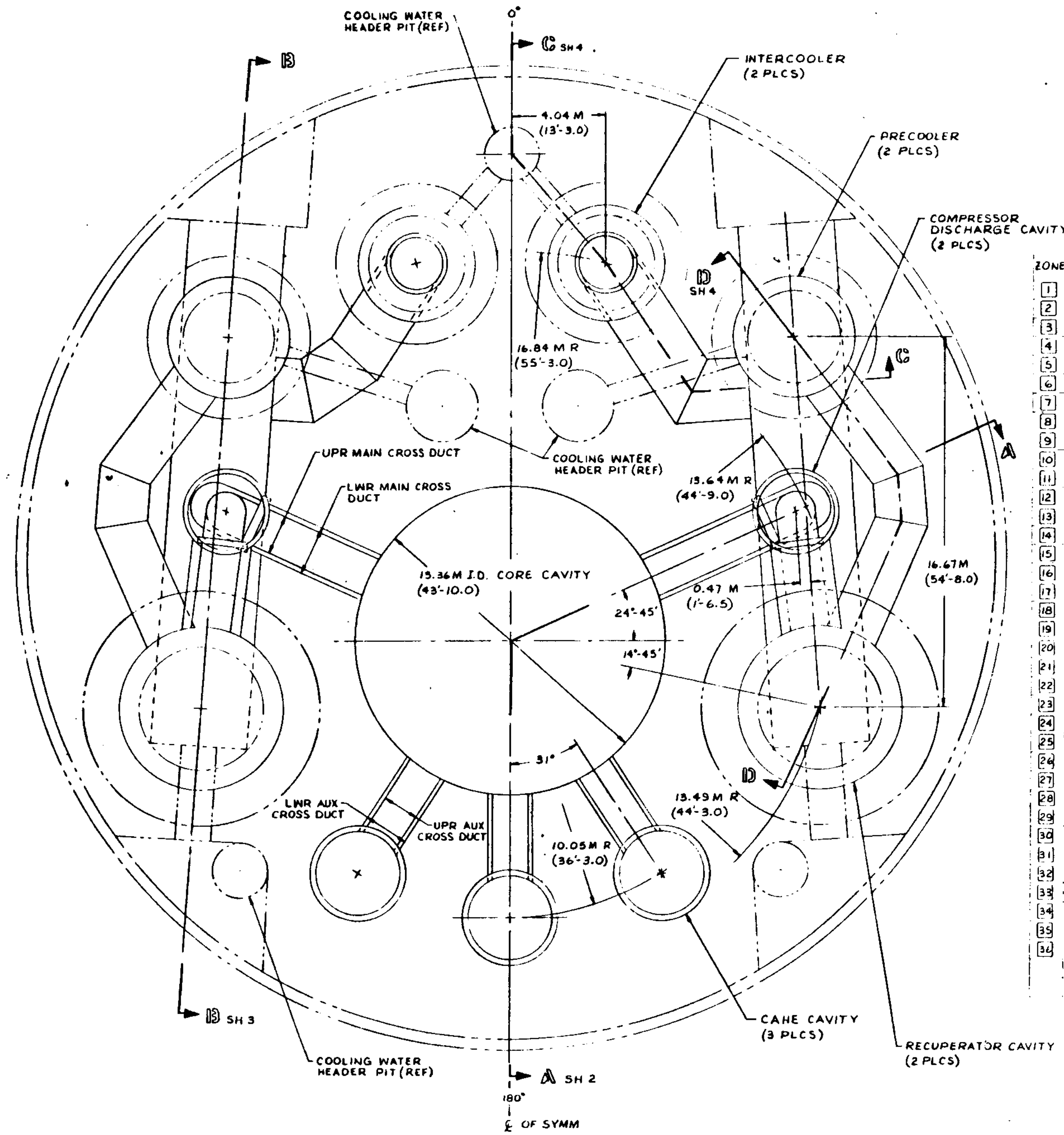
The HHT hot duct evaluation was continued with special emphasis on the ring seals, gimballed sections, expansion joints, and supports.

Different insulation systems and materials are being investigated to accommodate higher-temperature application. In addition to thermal problems, vibration and venting problems are also being considered in the design.

16.3. DISCUSSION

16.3.1. Two- and Three-Loop Layouts

Thermal barrier general arrangements were completed as a basis for cost estimating. These drawings (Figs. 16-1 and 16-2) define the thermal barrier zones, classes, thicknesses, and areas for each of the plants. Technically, from a thermal barrier point of view, there is very little difference between the two-loop and three-loop plants. The preliminary coverplate material selection is:



PLAN VIEW

NOTES:

1. INSTALLATION PER G.A. SPEC NO. 900008.
2. DIMENSIONS ARE IN METRIC UNITS WITH BRITISH UNITS ENCLOSED THUS ( ).
3. SYMBOLS:
   [x] INDICATES THERMAL BARRIER-ZONE NO.
   (LATER) (x)← NO. INDICATES SEQUENCE OF THERMAL BARRIER INSTALLATION. ARROW INDICATES DIRECTION OF PROGRESSIVE LAY-UP.

4 TOTAL AREA PER REACTOR (MEASURED AT LINER SURFACE)

NOTES CONT'D ON SH 2

ZONE	SEE SH	CLASS	QTY	THICKNESS 5 MM ±2.0	IN ±.12	AREA 4 M^2	FT^2	DRAWING NO.	ZONE DESCRIPTION
1	3	A	2	51	2.0	231	2489	(LATER)	TURBOMACHINE CAVITY, CENTER
2	3	B_1	2	76	3.0	89	956		" " , TURBINE END
3	2	B_1	2	76	3.0	164	1764		UPPER MAIN CROSS DUCT
4	2	B_1	1	76	3.0	140	1506		CORE CAVITY, TOP HEAD
5	2	B_1	1	127	5.0	174	1870		" " , UPPER SIDEWALL
6	2	B_1	1	127	5.0	390	4190		" " , CENTER SIDEWALL
7	2	B_2	1	127	5.0	116	1246		" " , LOWER SIDEWALL
8	2	C	1	378	14.88	140	1506		" " , BOTTOM HEAD
9	2	B_2	2	178	7.0	132	1420		LWR MAIN CROSS DUCT
10	3	B_1	2	76	3.0	28	301		TURBOMACHINE CAVITY, END WALL
11	3	B_1	2	76	3.0	13	142		TURBINE TO RECUPERATOR DUCT
12	2	B_1	3	76	3.0	51	544		UPPER AUXILIARY CROSS DUCT
13	2	B_1	3	127	5.0	349	3751		CAHE CAVITY, UPPER
14	2	B_2	3	178	7.0	60	648		LWR AUXILIARY CROSS DUCT
15	2	B_2	3	127	5.0	263	2832		CAHE CAVITY, LOWER
16	2	NOT REQUIRED							AUXILIARY CIRC. PENETRATION
17	3	B_1	2	76	3.0	70	757		RECUPERATOR CAVITY, LWR HEAD
18	3	B_1	2	76	3.0	351	3780		" " , LWR SIDEWALL
19	3	A	2	76	3.0	351	3780		" " , CTR SIDEWALL
20	3	A	2	51	2.0	340	3659		" " , UPR SIDEWALL
21	3	A	2	51	2.0	84	900		" " , CLOSURE
22	4	A	2	51	2.0	179	1930		" TO PRECOOLER DUCT
23	3	A	2	51	2.0	102	1101		PRECOOLER CAVITY, UPR SIDEWALL
24	3	A	2	51	2.0	42	457		" " , CLOSURE
25	3	A	2	51	2.0	723	7780		" " , LWR SIDEWALL
26	3	A	2	51	2.0	46	493		" " , LWR HEAD
27	3	A	2	51	2.0	48	521		" TO COMPR. DUCT
28	3	A	2	51	2.0	228	2458		TURBOMACHINE CAVITY, COMPR. END
29	2	A	2	51	2.0	747	8041		COMPR. DISCH CAVITY
30	2	NOT REQUIRED							" " " CLOSURE
31	3	A	2	51	2.0	145	1562		" TO RECUPERATOR DUCT
32	4	A	2	51	2.0	248	2670		LP COMPR OUTLET TO INTERCOOLER
33	4	A	2	51	2.0	625	6733		INTERCOOLER CAVITY
34	4	A	2	51	2.0	335	3607		" TO HP COMPR
35	—	NO INFORMATION							VALVE CAVITIES AND DUCTS
36	2	NO INFORMATION						(LATER)	PERIPHERAL SEAL

TOTAL = 7004 M^2 = 75,361 FT^2

CLASS A = 4474 M^2 = 48140 FT^2 = 64 %
" B_1 = 1819 M^2 = 19570 FT^2 = 26 %
" B_2 = 571 M^2 = 6144 FT^2 = 8 %
" C = 140 M^2 = 1506 FT^2 = 2 %

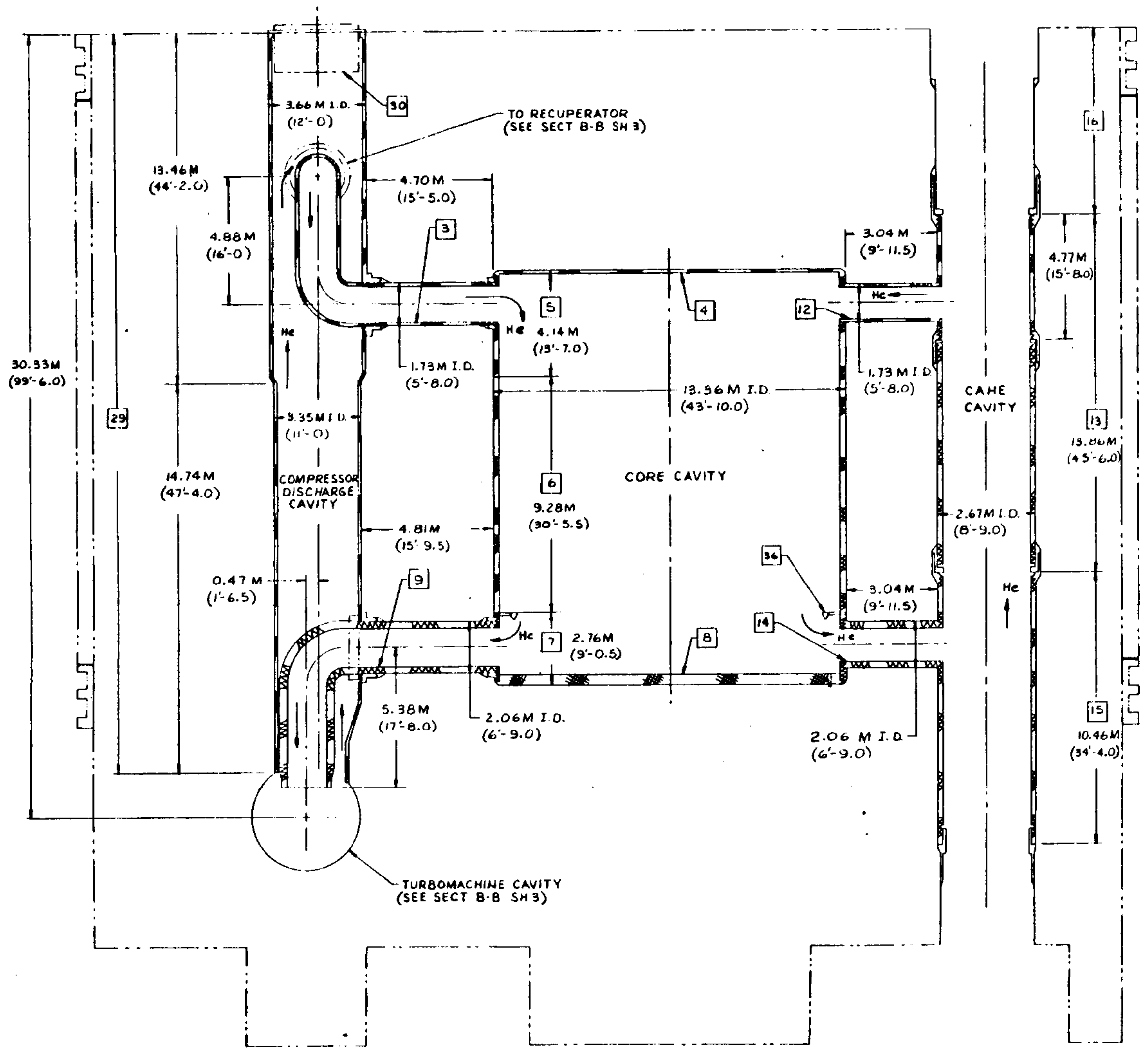
Fig. 16-1. Thermal barrier general arrangement for two-loop GT-HTGR (sheet 1 of 4)

NOTES: CONT'D FROM SH 1

5. THICKNESS AT INSTALLATION. DOES NOT INCLUDE ATTACHMENT FIXTURE PROTRUSION OF 25.4 MM (1.00) BEYOND COVER PLATE

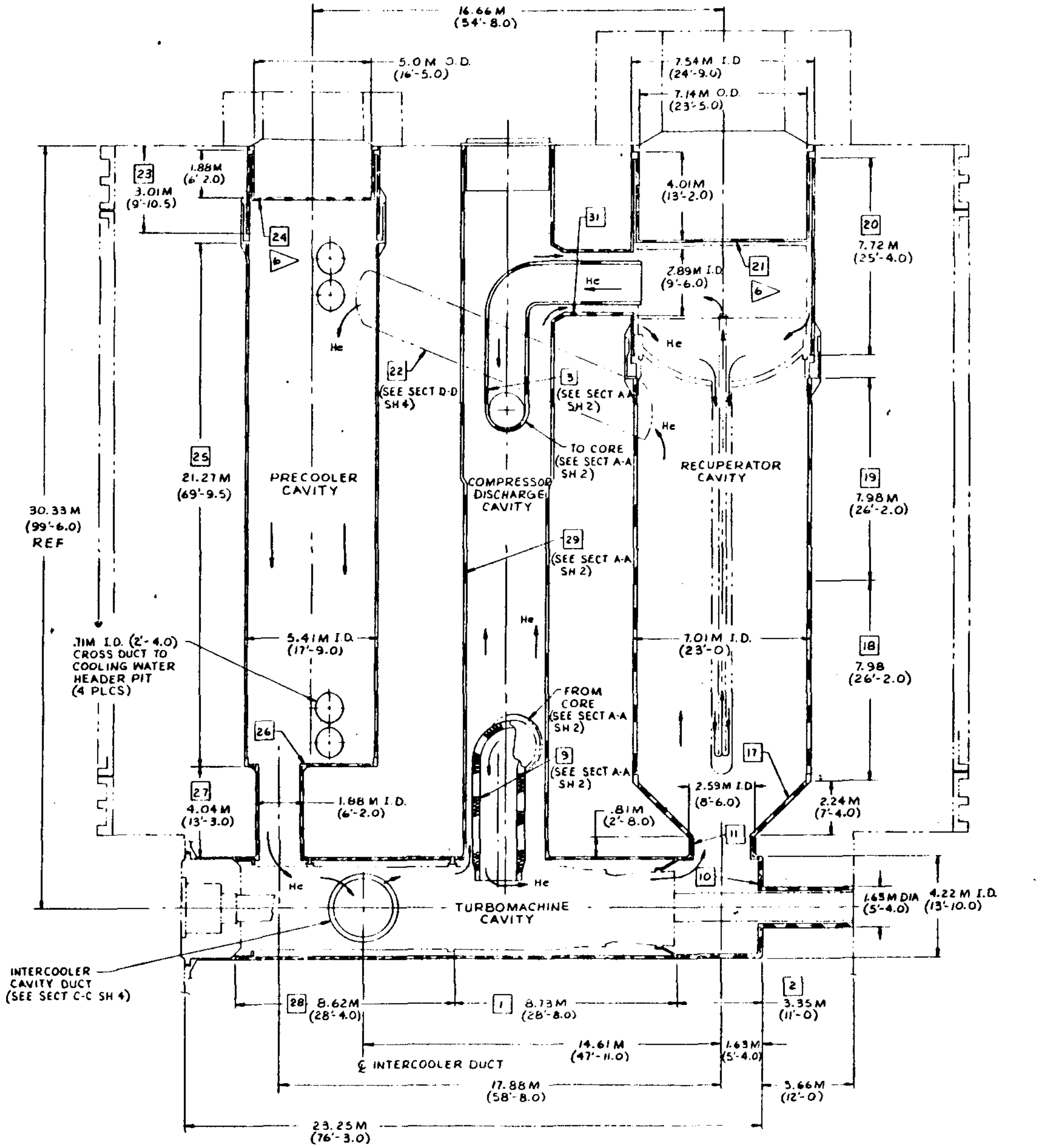
6. THERMAL BARRIER TO BE ASSEMBLED ONTO PLUG PRIOR TO INSTALLING PLUG INTO PCRV.

7. DIMENSIONS SHOW ARE PRELIMINARY AND ARE SCALED FROM PCRV DWG 024490.



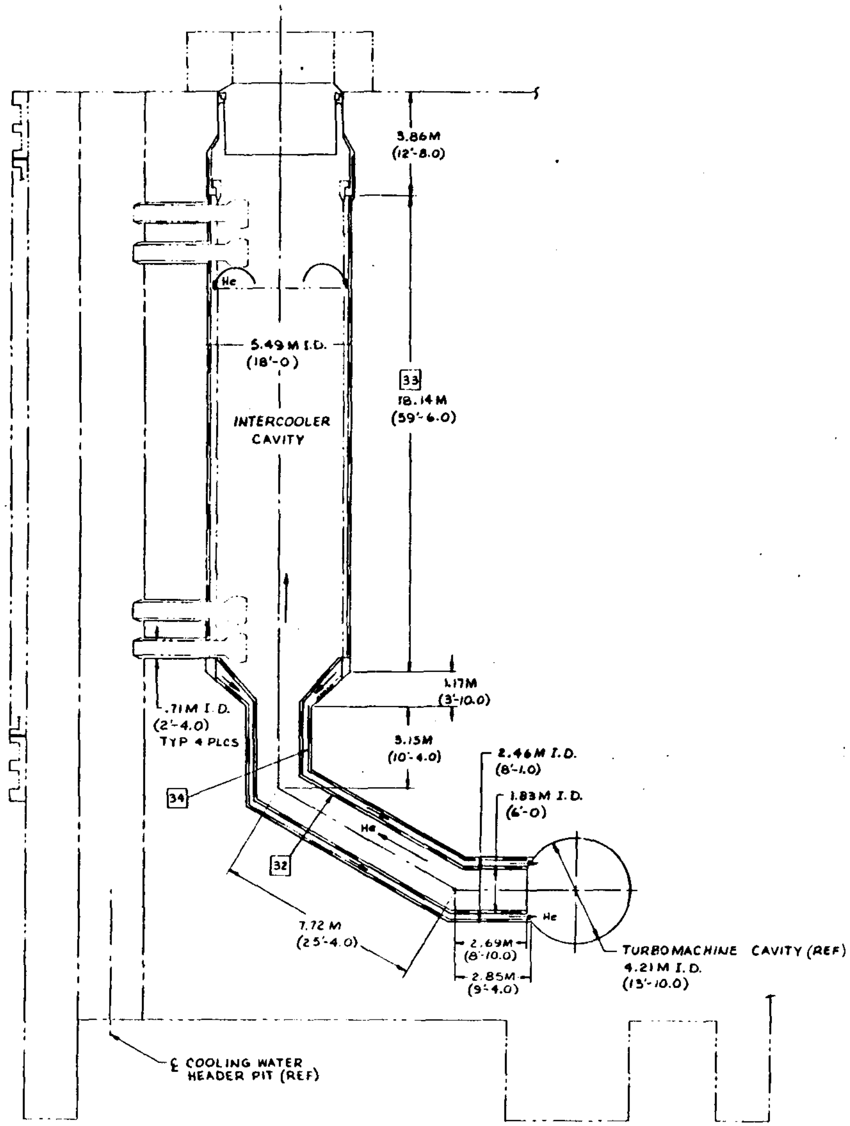
SECTION A-A SH 1
(ROTATED 90° CCW)

Fig. 16-1. Thermal barrier general arrangement for two-loop GT-HTGR (sheet 2 of 4)

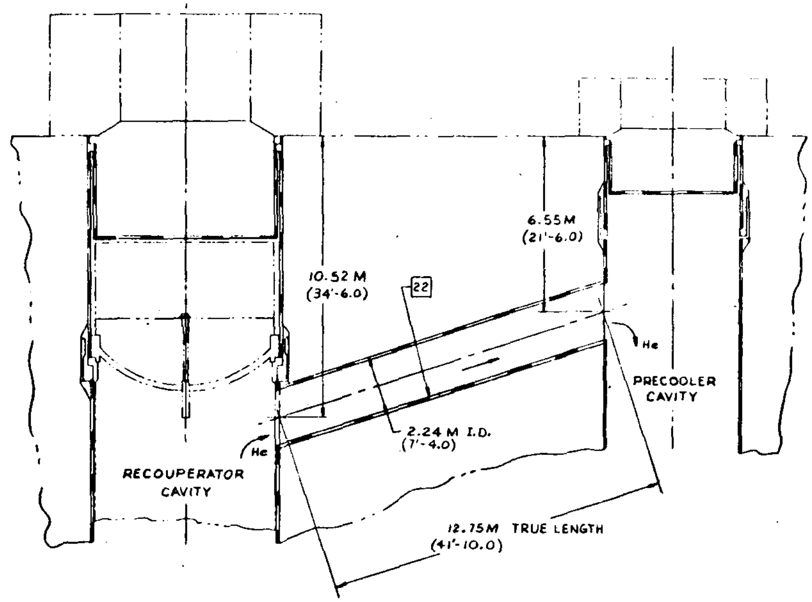


SECTION B-B SH 1
(ROTATED 90° CCW)

Fig. 16-1. Thermal barrier general arrangement for two-loop GT-HTGR (sheet 3 of 4)

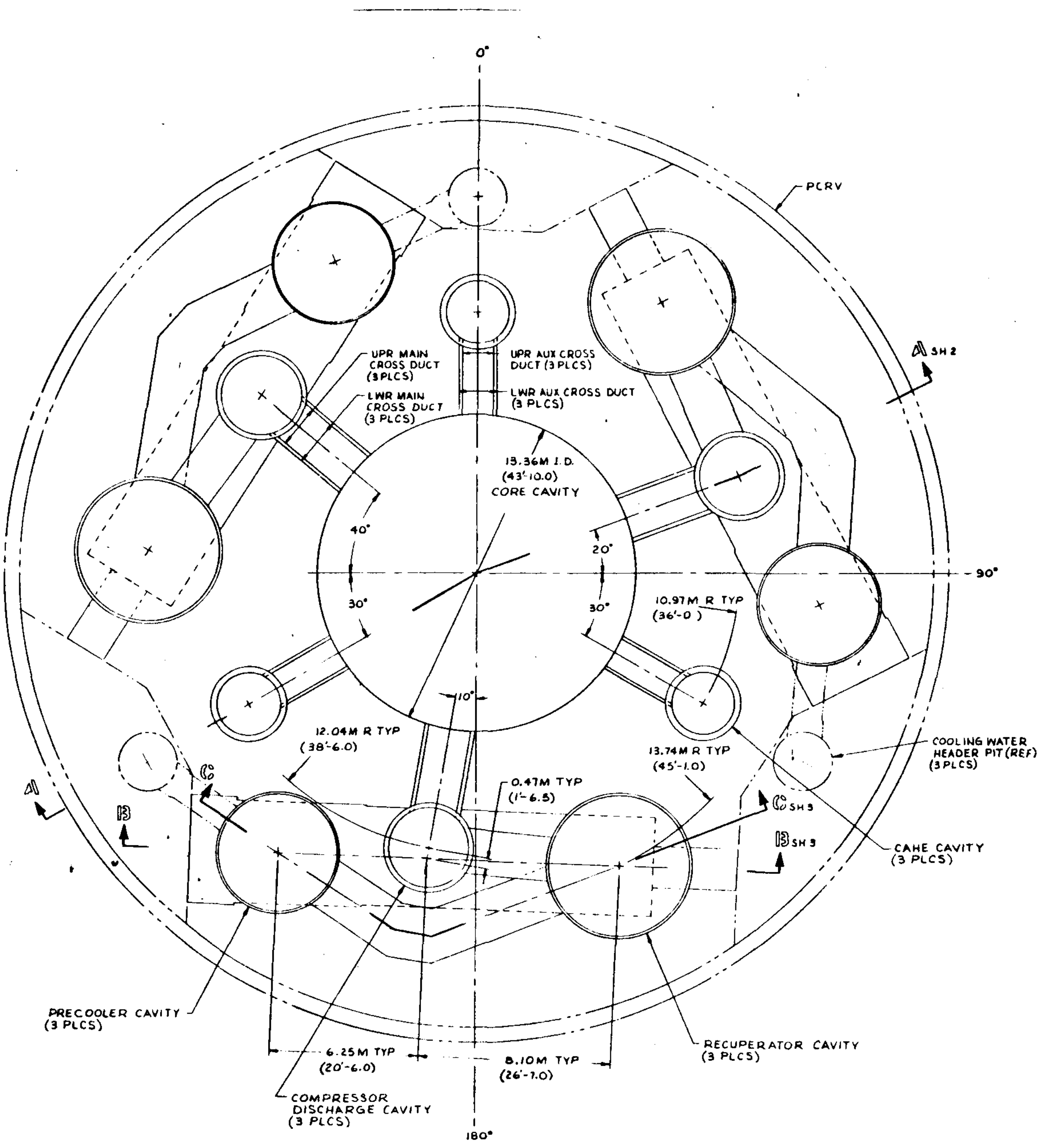


SECTION C-C SH I
(ROTATED INTO VIEW)



SECTION D-D SH I
(ROTATED INTO VIEW)

Fig. 16-1. Thermal barrier general arrangement for two-loop GT-HTGR (sheet 4 of 4)



PLAN VIEW

NOTES:

1. INSTALLATI[illegible] REF [illegible]
2. D[illegible]NSIONS ARE IN METRIC UNITS WITH BRITISH UNITS ENCLOSED THUS (   ).
3. SYMBOLS:
   [X] INDICATES THERMAL BARRIER ZONE NO.
   (LATER) (X) NO. INDICATES SEQUENCE OF THERMAL BARRIER INSTALLATION. ARROW INDICATES DIRECTION OF PROGRESSIVE LAY-UP.

4 TOTAL AREA PER REACTOR (MEASURED AT LINER SURFACE)

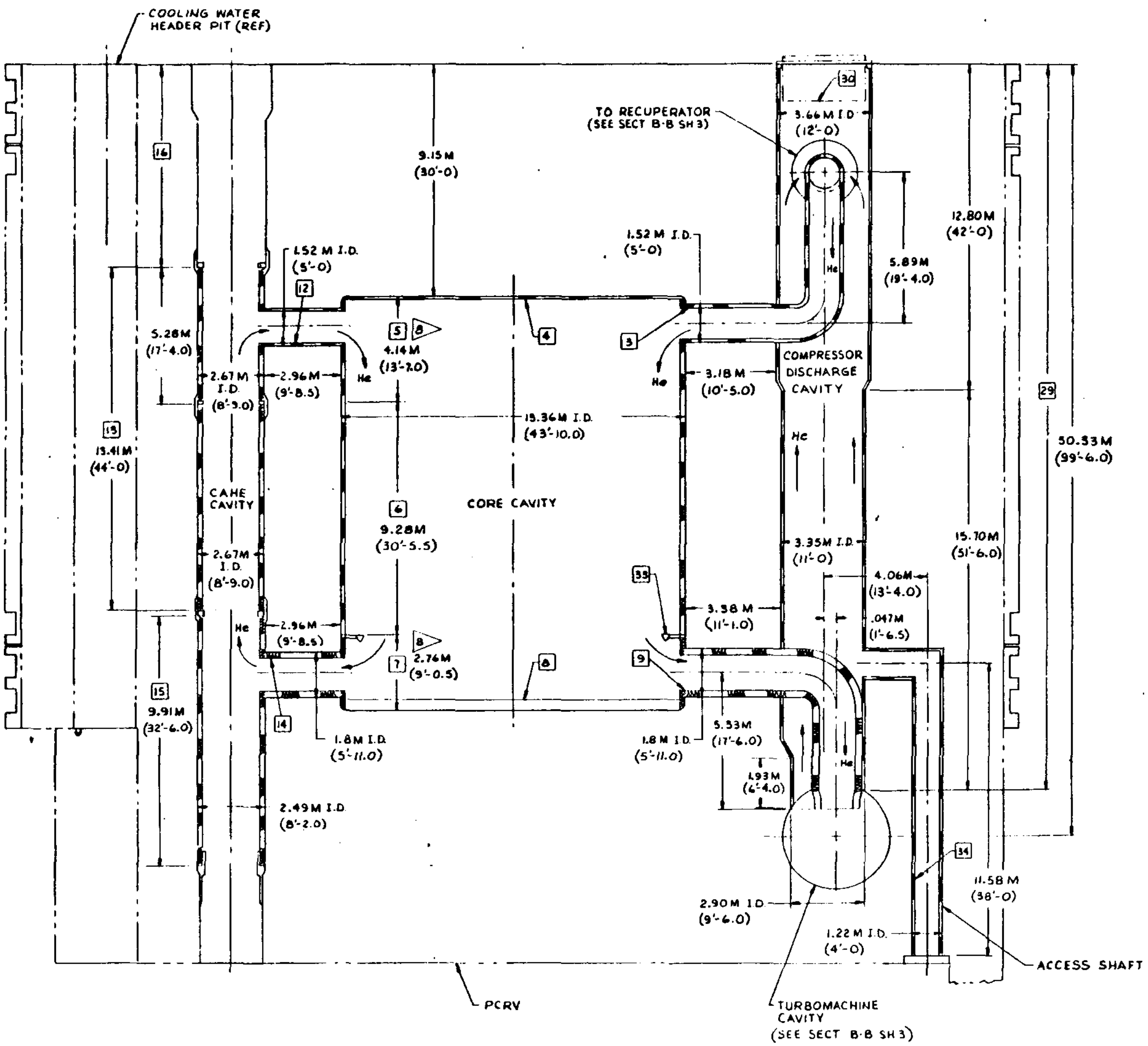
NOTES CONT'D ON SH 2

ZONE	SEE SH	CLASS	QTY	THICKNESS (5) MM ±3.0	IN ±.12	AREA (4) M²	FT²	DRAWING NO.	ZONE DESCRIPTION
1	3	A	3	51	2.0	310	3336	(LATER)	TURBOMACHINE CAVITY, CENTER
2	3	B1	3	76	3.0	126	1357		.. .., TURBINE END
3	2	B1	3	76	3.0	215	2313		UPPER MAIN CROSS DUCT
4	2	B1	1	76	3.0	140	1506		CORE CAVITY, TOP HEAD
5	2	B1	1	127	5.0	174	1872		.. .., UPPER SIDEWALL
6	2	B1	1	127	5.0	390	4196		.. .., CENTER SIDEWALL
7	2	B2	1	127	5.0	116	1248		.. .., LOWER SIDEWALL
8	2	C	1	378	14.88	140	1506		.. .., BOTTOM HEAD
9	2	B2	3	178	7.0	170	1829		LOWER MAIN CROSS DUCT
10	3	B1	3	76	3.0	91	979		TURBOMACHINE CAVITY, END WALL
11	3	B1	3	76	3.0	72	775		TURBINE TO RECUPERATOR DUCT
12	2	B1	3	76	3.0	43	463		UPPER AUXILIARY CROSS DUCT
13	2	B1	3	127	5.0	338	3637		CAHE CAVITY, UPPER
14	2	B2	3	178	7.0	54	581		LOWER AUXILIARY CROSS DUCT
15	2	B2	3	127	5.0	232	2496	(LATER)	CAHE CAVITY, LOWER
16	2	NOT REQUIRED							AUXILIARY CIRC. PENETRATION
17	3	B1	3	76	3.0	86	925	(LATER)	RECUPERATOR CAVITY, LWR HEAD
18	3	B1	3	76	3.0	442	4756		.. .., LWR SIDEWALL
19	3	A	3	76	3.0	442	4756		.. .., CTR SIDEWALL
20	3	A	3	51	2.0	391	4206		.. .., UPR SIDEWALL
21	3	A	3	51	2.0	207	2233		.. .., CLOSURE
22	2	A	3	51	2.0	237	2550		.. TO PRECOOLER DUCT
23	3	A	3	51	2.0	145	1560		PRECOOLER CAVITY, UPR SIDEWALL
24	3	A	3	51	2.0	135	1458		.. .., CLOSURE
25	3	A	3	51	2.0	924	9948		.. .., LWR SIDEWALL
26	3	A	3	51	2.0	59	635		.. .., LWR HEAD
27	3	A	3	51	2.0	98	1056		.. TO COMPR. DUCT
28	3	A	3	51	2.0	263	2830		TURBOMACHINE CAVITY, COMPR. END
29	2	A	3	51	2.0	1000	10760	(LATER)	COMPR. DISCH CAVITY
30	2	NOT REQUIRED							.. .. .., CLOSURE
31	3	A	3	51	2.0	113	1216	(LATER)	.. TO RECUPERATOR DUCT
32	—	NO INFORMATION							VALVE CAVITIES AND DUCTS
33	2	NO INFORMATION							PERIPHERAL SEAL
34	2	A	3	51	2.0	160	1721	(LATER)	ACCESS SHAFT

TOTAL = 7162 M² = 77063 FT²

CLASS A = 4533M² = 48823FT² = [illegible]
.. B1 = 2117M² = 22779FT² = 30[illegible]
.. B2 = 572M² = 6155FT² = 8%
.. C = 140M² = 1506FT² = 2[illegible]

Fig. 16-2. Thermal barrier general arrangement for thre-loop GT-HTGR (sheet 1 of 3)



NOTES:

5. THICKNESS AT INSTALLATION. DOES NOT INCLUDE ATTACHMENT FIXTURE PROTRUSION OF 25.4 MM (1.00) BEYOND COVER PLATE.

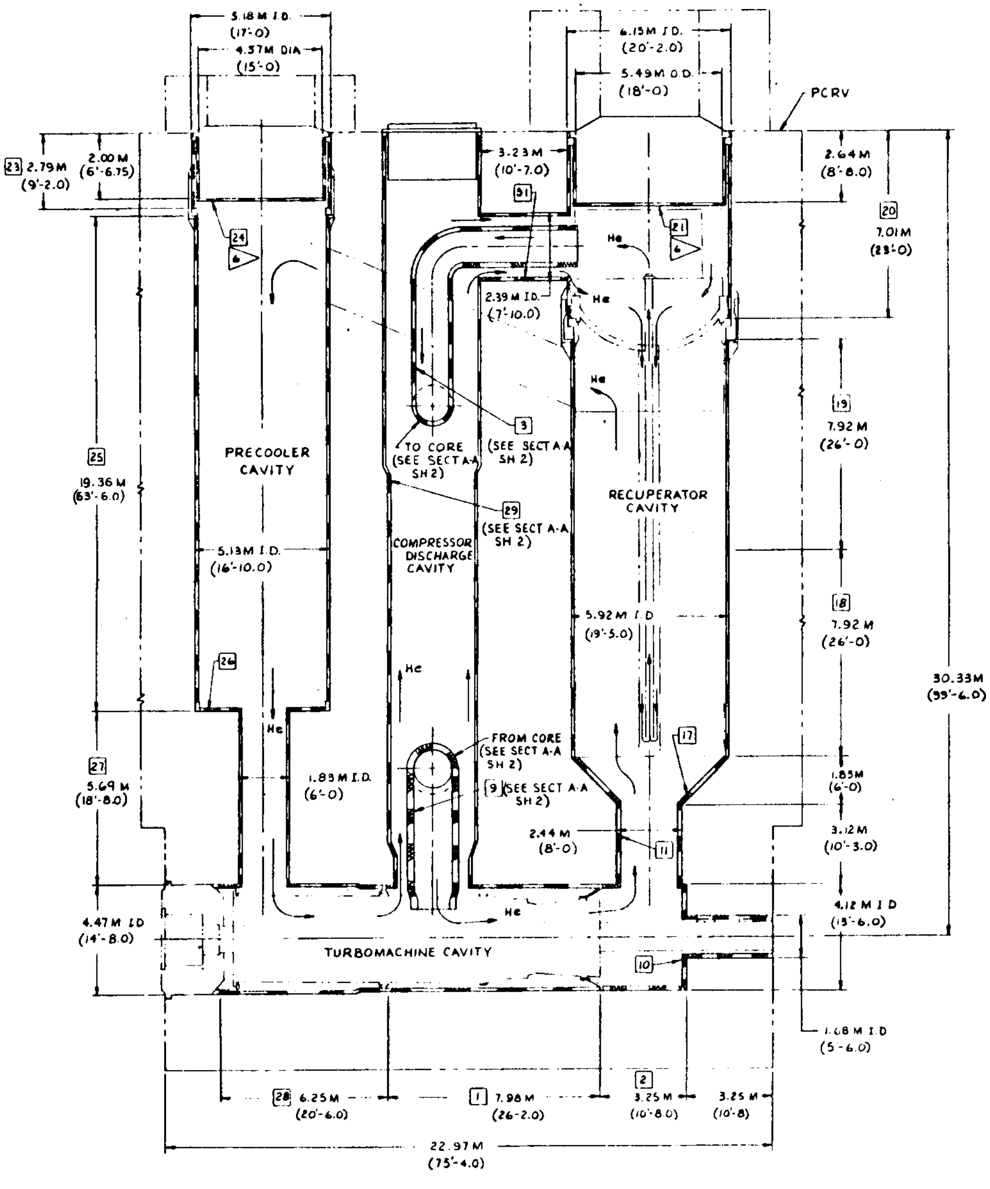
6. THERMAL BARRIER TO BE ASSEMBLED ONTO PLUG PRIOR TO INSTALLING PLUG INTO PCRV

7. DIMENSIONS SHOWN ARE PRELIMINARY AND ARE SCALED FROM PCRV CONCEPT DWG 024608

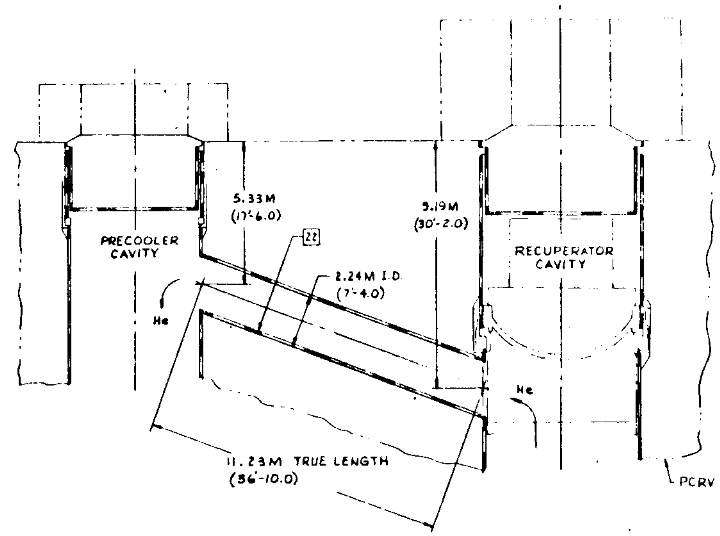
8. THE HEIGHT OF ZONES 5 & 7 ARE THE SAME AS THE 2 LOOP PLANT (REF DWG 024681). THESE HEIGHTS CAN BE REDUCED DUE TO THE REDUCED DIAMETERS OF ZONES 3, 9, 12 & 14.

SECTION A-A SH 1

Fig. 16-2. Thermal barrier general arrangement for three-loop GT-HTGR (sheet 2 of 3)



SECTION B-B SH 1



SECTION C-C SH 1

Fig. 16-2. Thermal barrier general arrangement for three-loop GT-HTGR (sheet 3 of 3)

Class	Normal Condition Temperature Range	Primary Material	Alternate Materials
A	To 370°C (700°F)	Carbon steel	--
B1	370°-650°C (700°-1200°F)	Type 316 SS	Incoloy 800-H, Hastelloy-X
B2	650°-927°C (1200°-1700°F)	Carbon-carbon (C-C)	Inconel 713LC, Inconel 100, Inconel 162
C	850°-1170°C (1562°-2140°F)	Alumina, silica, Type 316 SS	Graphite, SiC, Hastelloy X

16.3.2. HHT Conceptual Hot Duct Evaluation

This evaluation is still in progress. However, the consensus at this time is that the ducts are very complex. This is particularly true for the ring seals, gimballed sections, expansion joints, and supports. If replaceability is a criterion, then all ducts must be simplified.

16.3.3. High-Temperature Design Development

In conjunction with the parametric study, designs are being developed that, in particular, are to accommodate vibration and depressurization. Figure 16-3 shows three types of coverplates and insulation systems that have been submitted for detailed analyses. These coverplate designs, with modifications, can be fabricated from carbon-carbon or cast superalloys. Two approaches to the hot duct design have been investigated (Figs. 16-4 and 16-5). The first is an example of cylindrical carbon-carbon sections with toroidal insulation washers. Installation, removal, and replacement are relatively simple, but the concept is expensive because of material costs.

The other concept involves coverplates that permit rapid venting by virtue of insulation-side grooves and perforated cruciform seal sheets. Since vibration is expected to be a dominating parameter, the coverplates are shown with multiple intrapanel attachments for stability and the insulation is encapsulated and stitched by ceramic cloth.

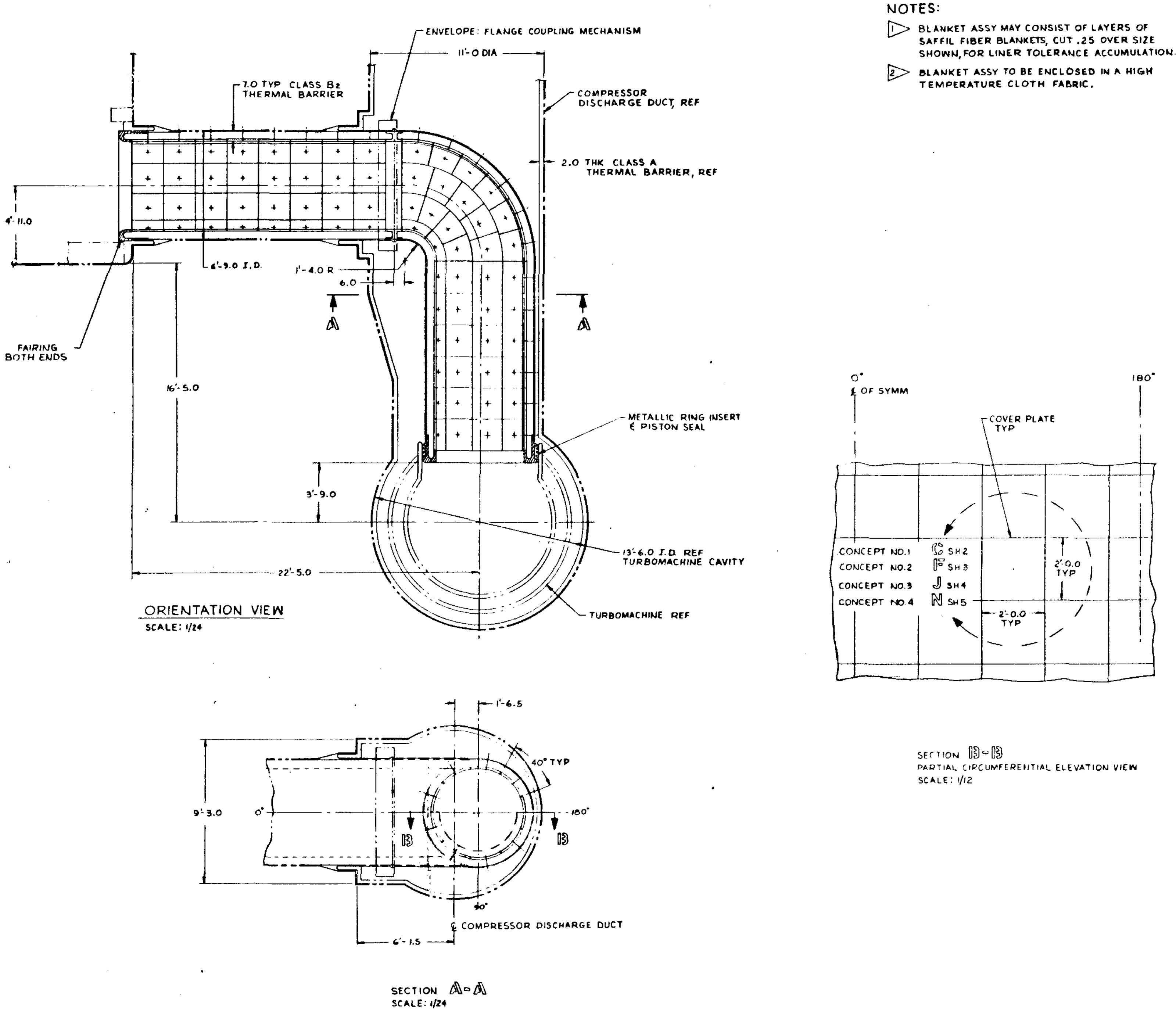


Fig. 16-3. High-temperature thermal barrier installation concept (sheet 1 of 5)

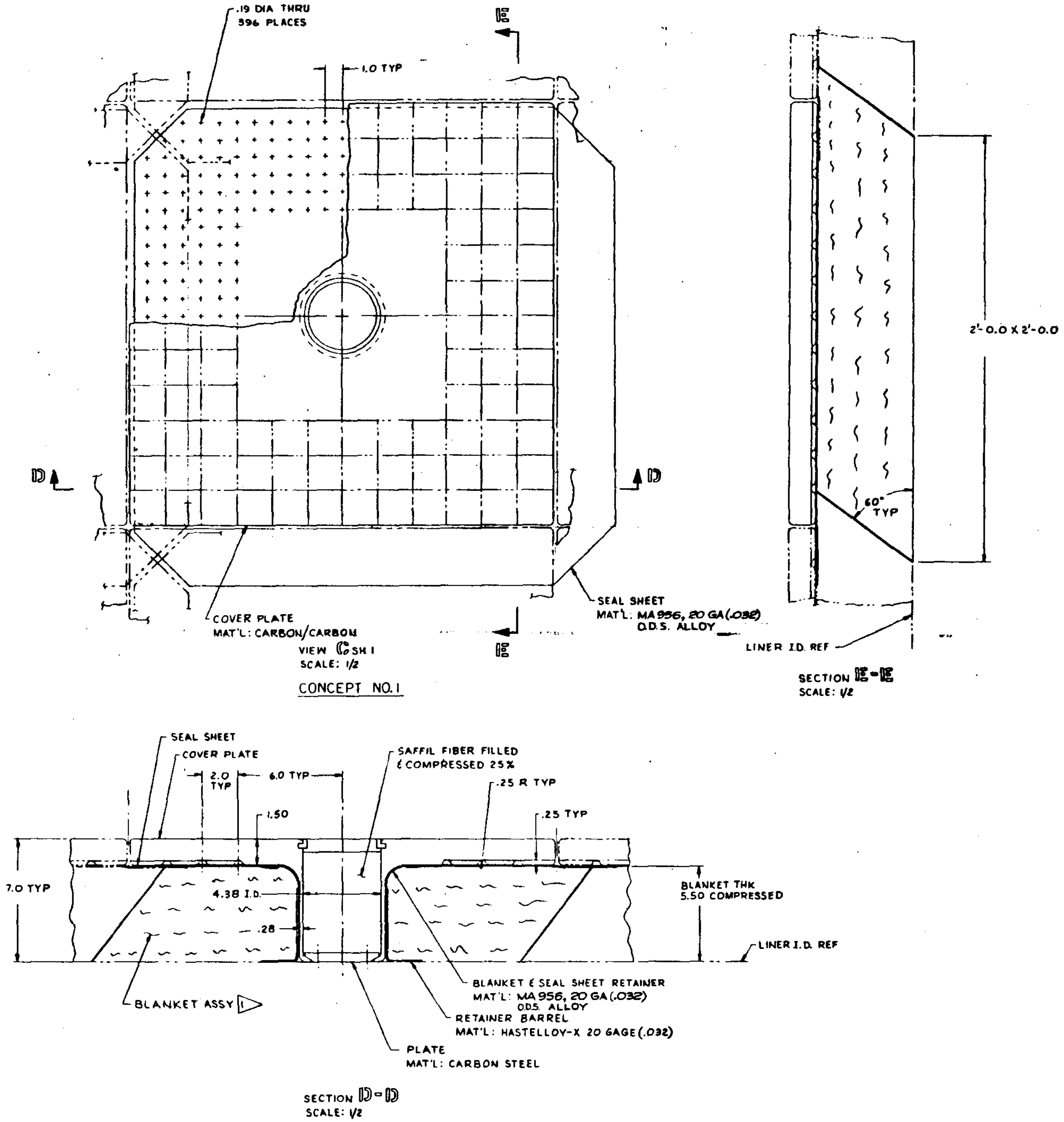


Fig. 16-3. High-temperature thermal barrier installation concept (sheet 2 of 5)

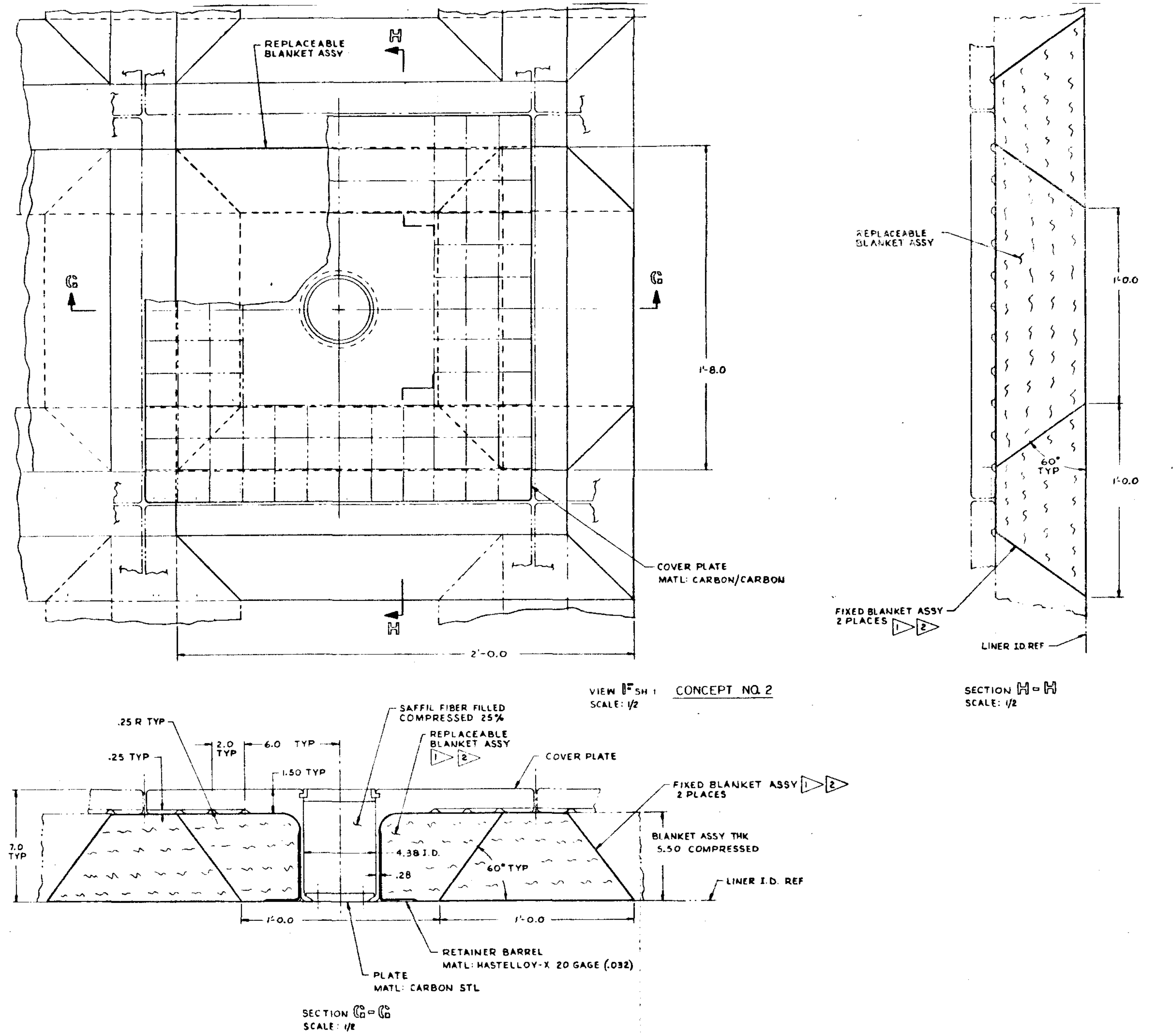


Fig. 16-3. High-temperature thermal barrier installation concept (sheet 3 of 5)

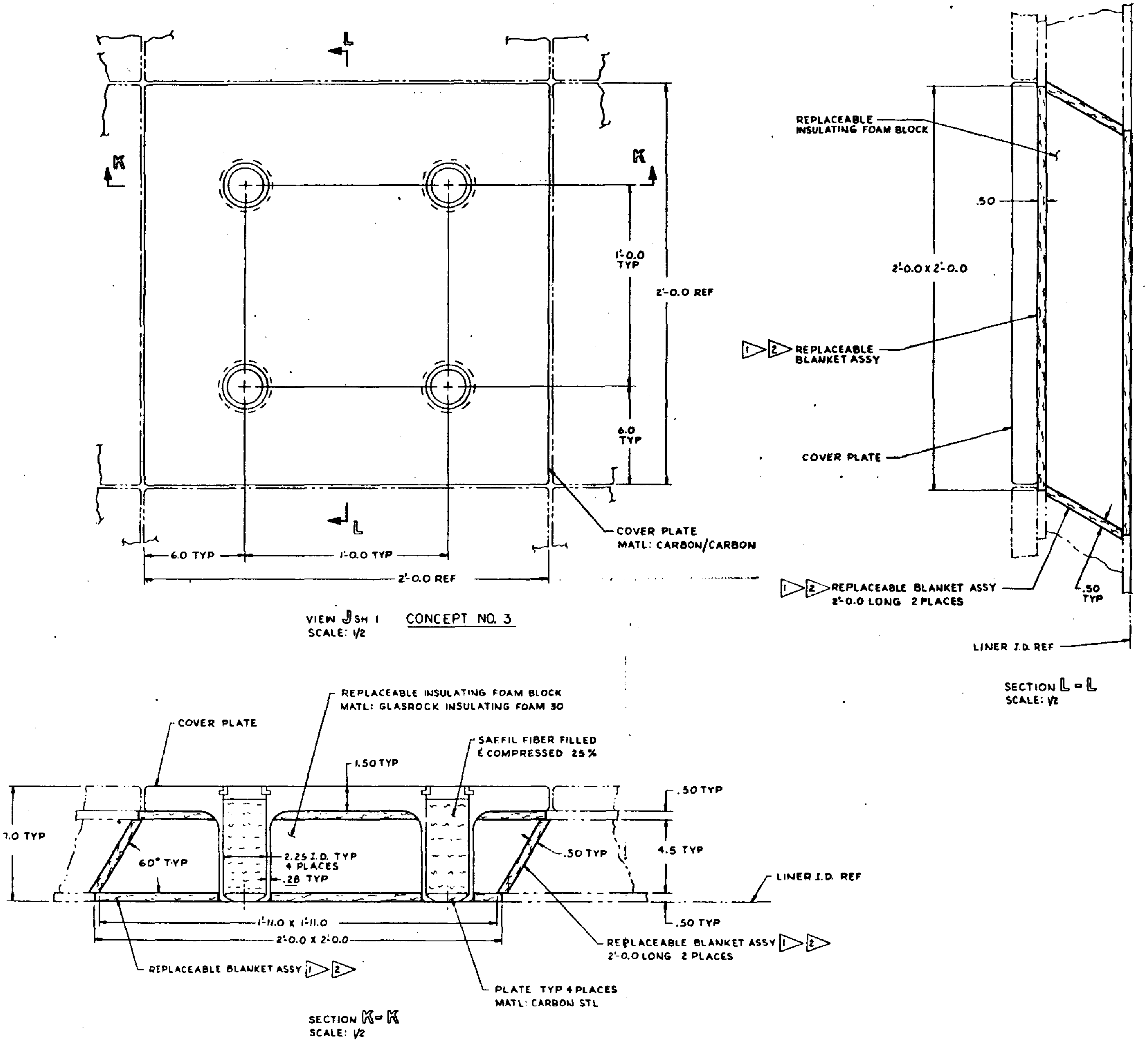


Fig. 16-3. High-temperature thermal barrier installation concept (sheet 4 of 5)

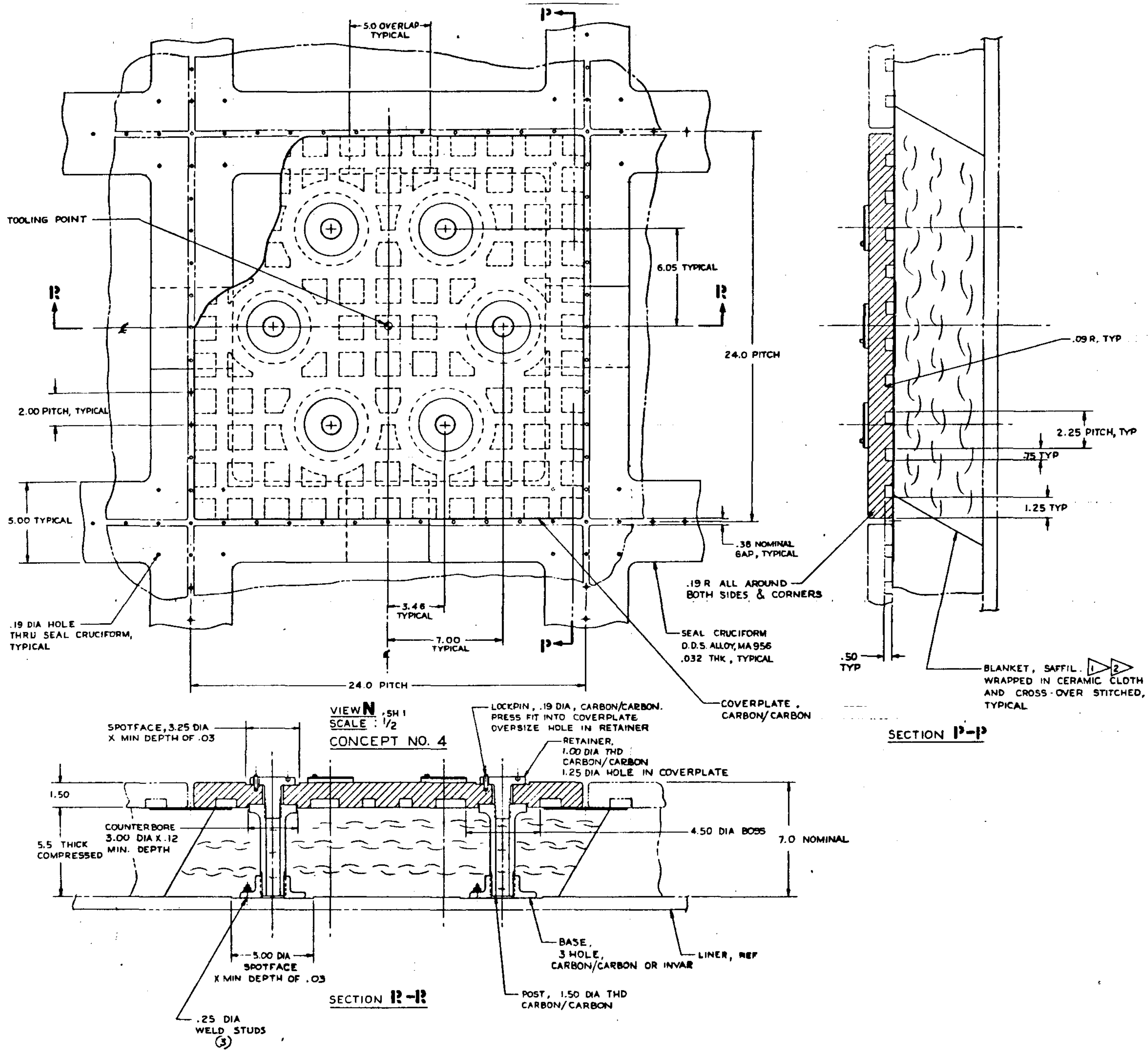


Fig. 16-3. High-temperature thermal barrier installation concept (sheet 5 of 5)

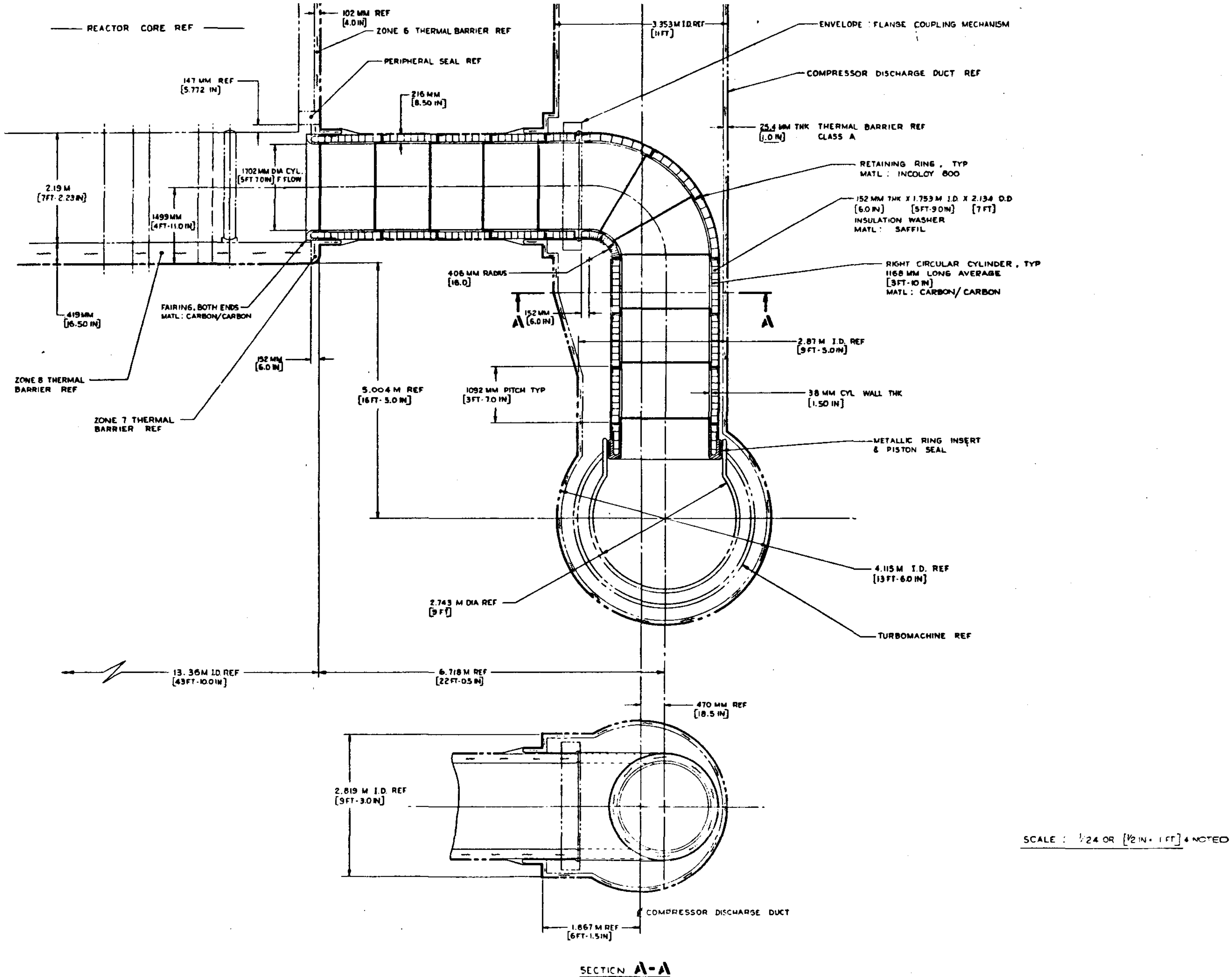


Fig. 16-4. Hot duct concept featuring cylindrical carbon-carbon sections with toroidal insulation washers

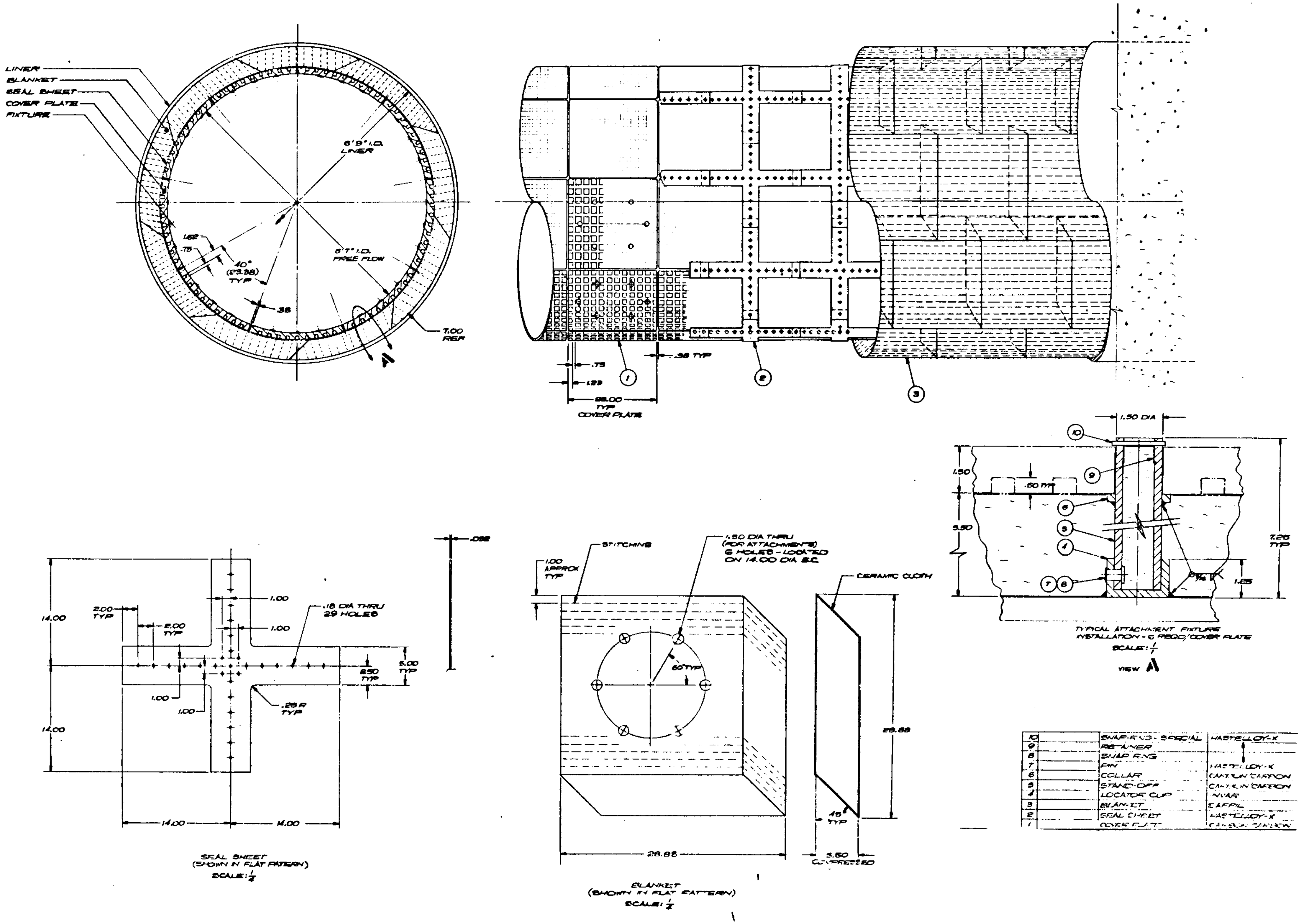


Fig. 16-5. Alternate hot duct concept (sheet 1 of 2)

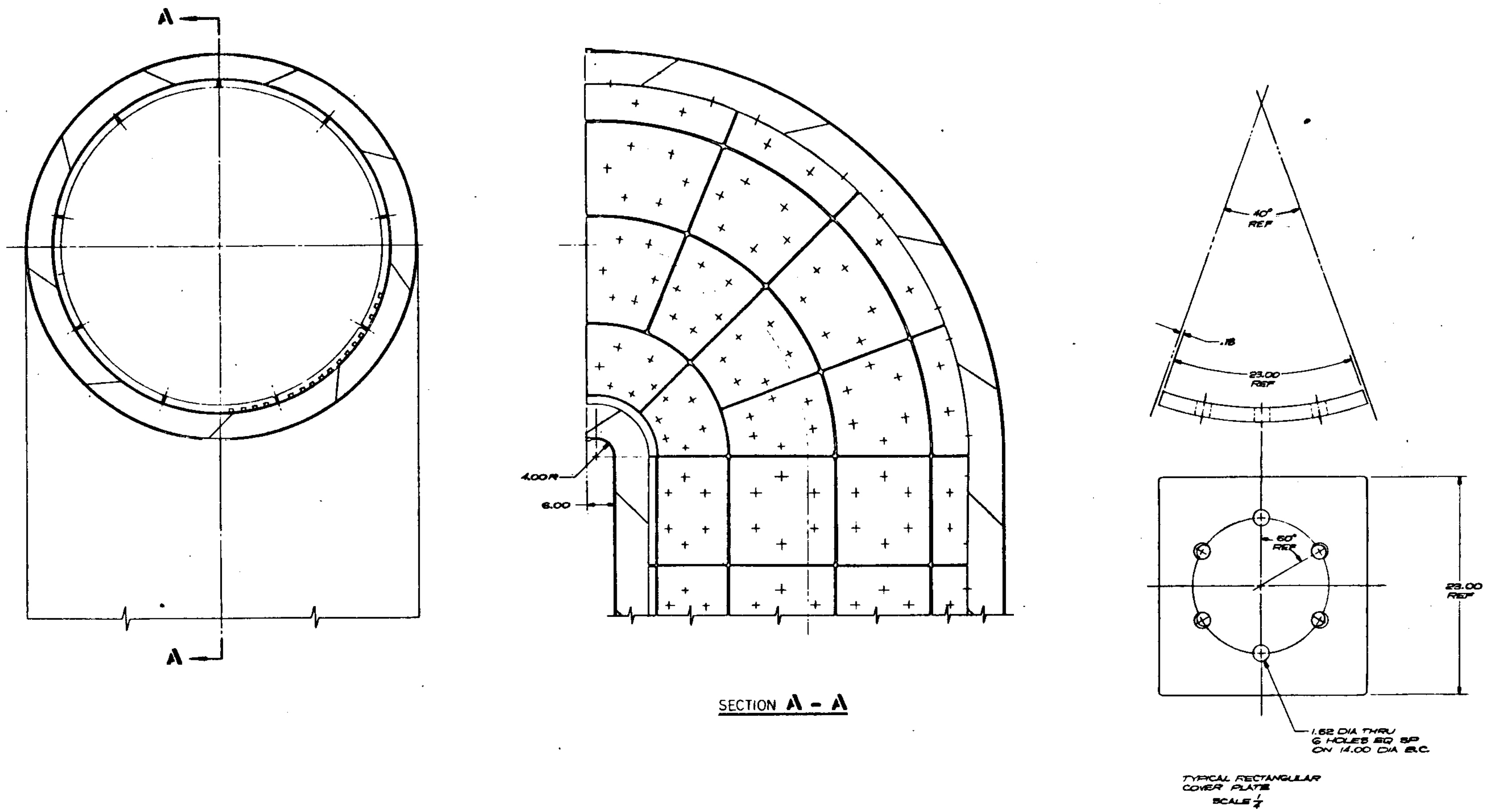


Fig. 16-5. Alternate hot duct concept (sheet 2 of 2)

16.3.4. Plant System Parameter Evaluation

Three major issues are of concern in developing reliable thermal barrier designs for the GT-HTGR: acoustic vibration, core outlet temperatures, and rapid depressurization rates. In general, except for uncertainties relating to the fatigue of fibrous insulation and some minor concern regarding depressurization, the regions having temperatures lower than 650°C (1200°F), that is, Classes A and B1, are not expected to pose significant technical problems. It is the lower core plenum and the hot ducts (Class B2) where the potential problems are sufficiently great to warrant a major design and design verification and support effort in order to demonstrate feasibility.

First, and most important, is the issue of vibration. The overall sound pressure levels generated by the turbomachine are projected to be in the range of 174 dB. Local narrow band levels could be even higher. At these levels the fatigue resistance of coverplates, attachments, and insulation is extremely tenuous. Of the material choices, metallic structures are given the best chance of survival, primarily due to a relatively high modulus of elasticity. However, the potential for damage to the fibrous insulation materials is very great if a "conventional" HTGR thermal barrier system is to be used. A possible solution to this problem is to encapsulate and quilt the insulation packages, thereby minimizing fiber damage.

Second, the core outlet temperature of 850°C (1562°F) narrows the choice of practical Class B2 material candidates to cast superalloys, the carbonaceous fiber-reinforced composites (C-C), and hard ceramics. The data base for these materials is very limited, especially in the areas of fatigue and long-term creep. Hence, the ability to properly analyze any of the proposed configurations is greatly lessened.

Finally, there are opposing requirements for controlled convection within the thermal barrier in order to minimize heat transfer and the necessity for rapid venting of the system during a depressurization accident. It is believed possible to design a coverplate/seal sheet arrangement that can accommodate the proposed depressurization rates.

In support of the aforementioned issues, design verification and support programs will be required to establish material properties, evaluate coverplate/attachment fixture fabricability, determine insulation damage tolerance, and assess the combined system performance.

16.3.5. Warm Versus Cold Liner Assessment

The thermal barrier is influenced by three factors that are unique to the warm liner concept: the core barrel design, concentric ducts, and hot duct/core barrel interface.

The core barrel is made up of segmented interlocking plates. This leads to a movable base for attachment of the high-temperature thermal barrier. This is particularly critical for the Class C thermal barrier, where hard ceramic blocks are part of the thermal barrier and core support design. In addition, the gaps between the core barrel segments increase the probability of bypass flow through the thermal barrier, which could greatly decrease the effectiveness of its function.

In order to allow replaceability of the concentric ducts and insulation, the ducts must be segmented. This makes the design of the thermal barrier more complex. In addition, the segmented duct presents a non-rigid foundation for the thermal barrier, which increases the probability of failure due to bypass flow.

Seismic movements and thermal expansion of the core barrel cause design problems at the core barrel/cross-duct interfaces. The current design concept uses a double bellows to accommodate these movements. Design of a flexible thermal barrier for this interface region will be extremely difficult and could affect the normal function of the thermal barrier owing to the possibility of gas bypass.

All the above problems would require extensive analysis and testing above that required for the cold liner design.

17. REACTOR INTERNALS (6317)

17.1. SCOPE

The purpose of this task is to provide two-loop and three-loop reference plant conceptual arrangement drawings and conduct a hot duct evaluation.

17.2. SUMMARY

17.2.1. Demonstration Plant

The revised conceptual layout drawing (Fig. 17-1) for the reactor internals of the GA version of the HHT demonstration plant was issued.

Reactor internals input to the evaluation report on a warm versus cold liner design was issued. The conclusion of the study is that for the reactor internals themselves, the only potential advantages identified to date for internals of the warm liner configuration are that a more limited zone of the core lateral restraint springpack assembly needs to be able to resist high temperatures (for which no materials are available in any case) and that access to the springpacks may be possible in service.

On the other hand, the very real disadvantages must be weighed carefully against any advantages of the warm liner configuration which may be identified for the plant system other than the reactor internals.

For the concentric, high-temperature internal ducts, the problems are essentially similar in nature for the cold and the warm liner configurations (Figs. 17-2 and 17-3). However, the number of these ducts and their lengths are greater in the warm liner configuration considering the additional

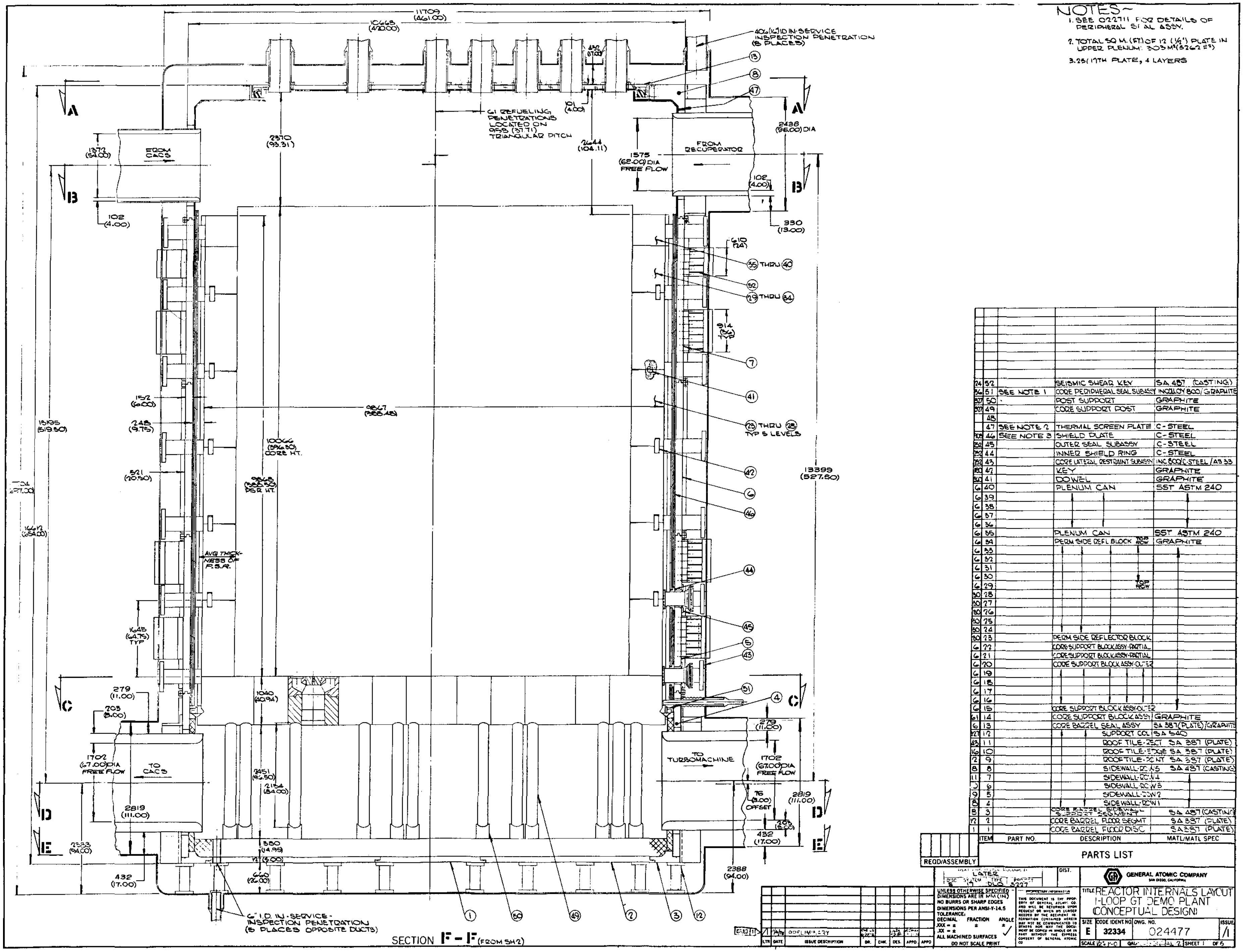


Fig. 17-1. Conceptual design of reactor internals for demonstration plant (sheet 1 of 6)

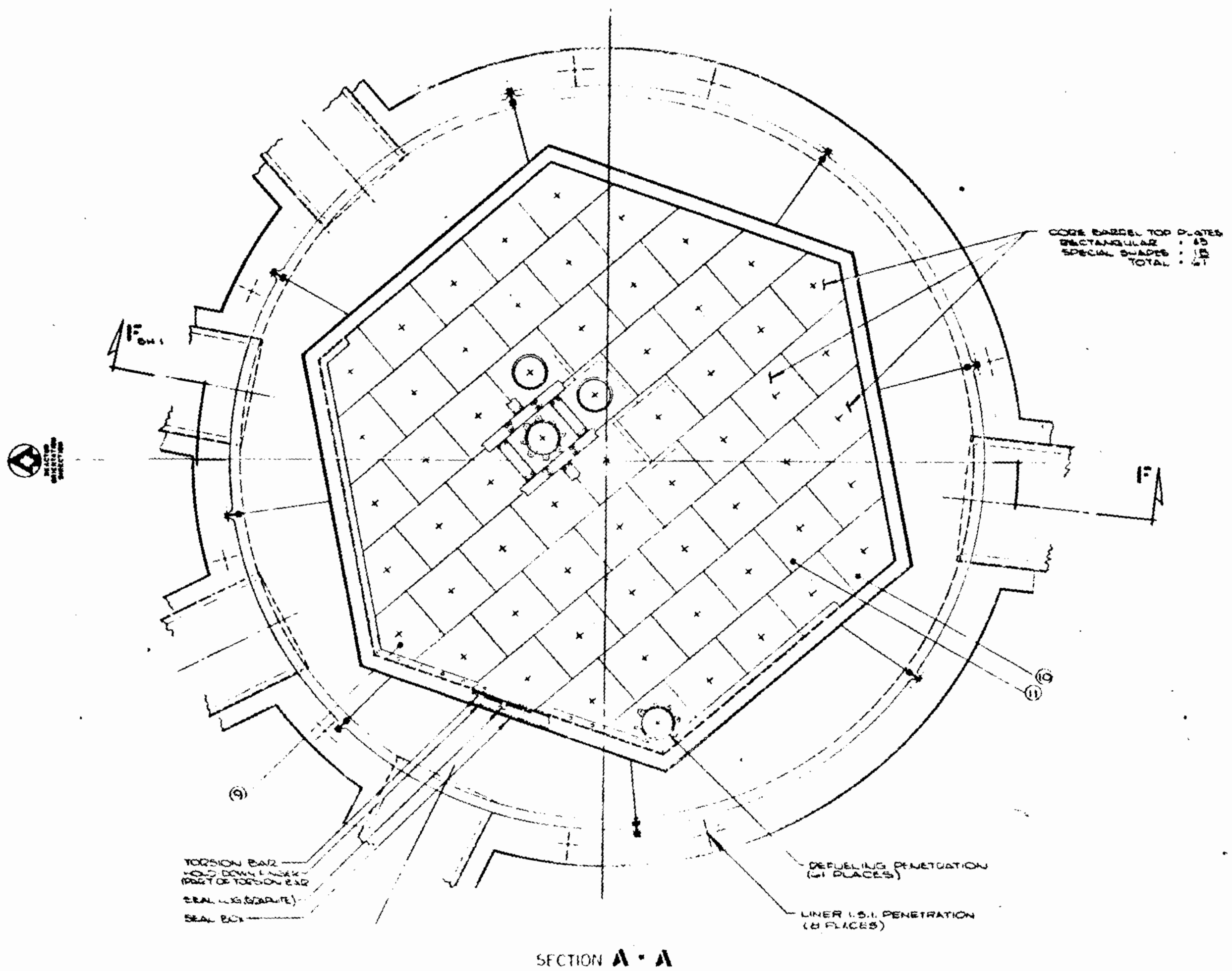


Fig. 17-1. Conceptual design of reactor internals for demonstration plant (sheet 2 of 6)

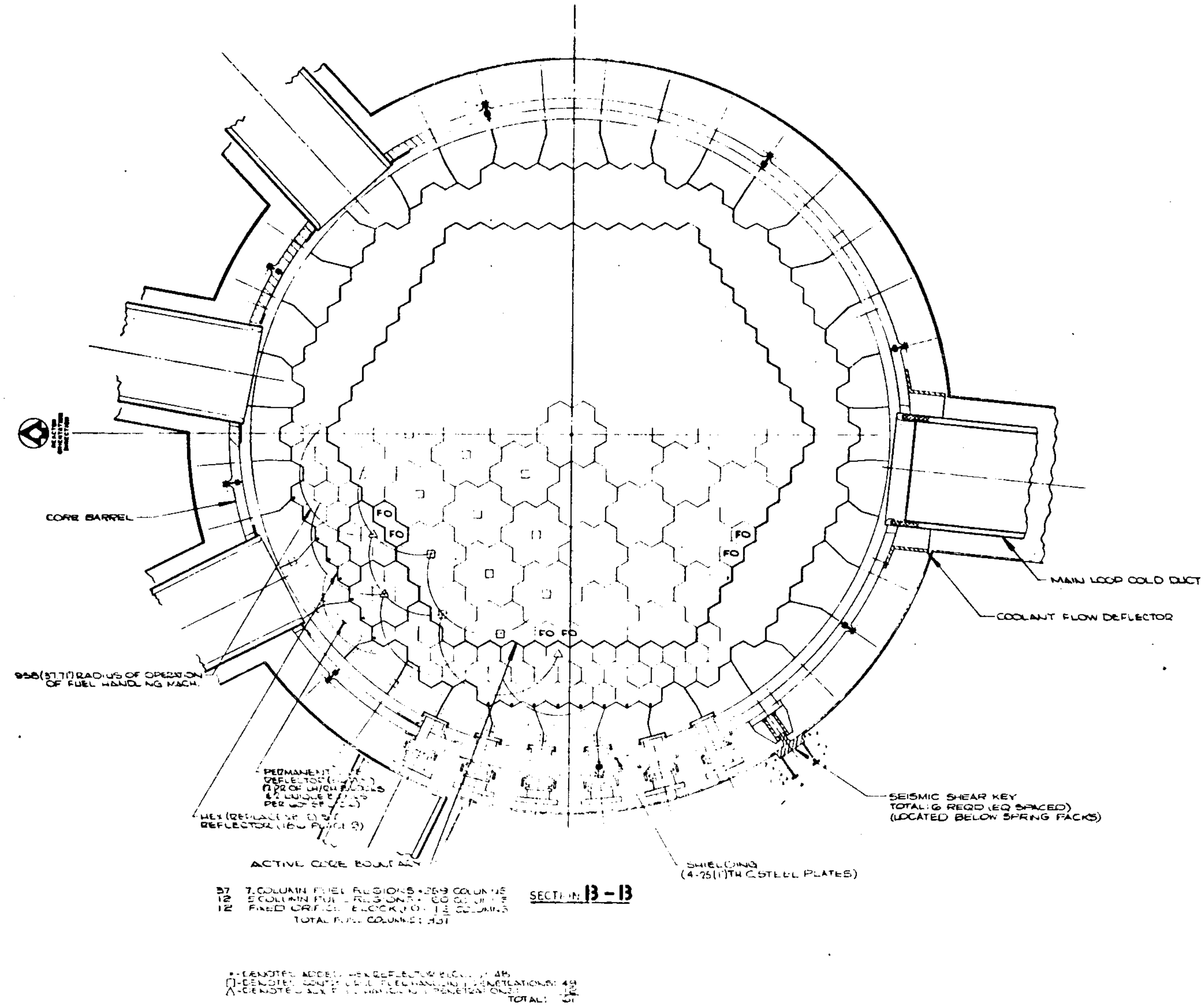


Fig. 17-1. Conceptual design of reactor internals for demonstration plant (sheet 3 of 6)

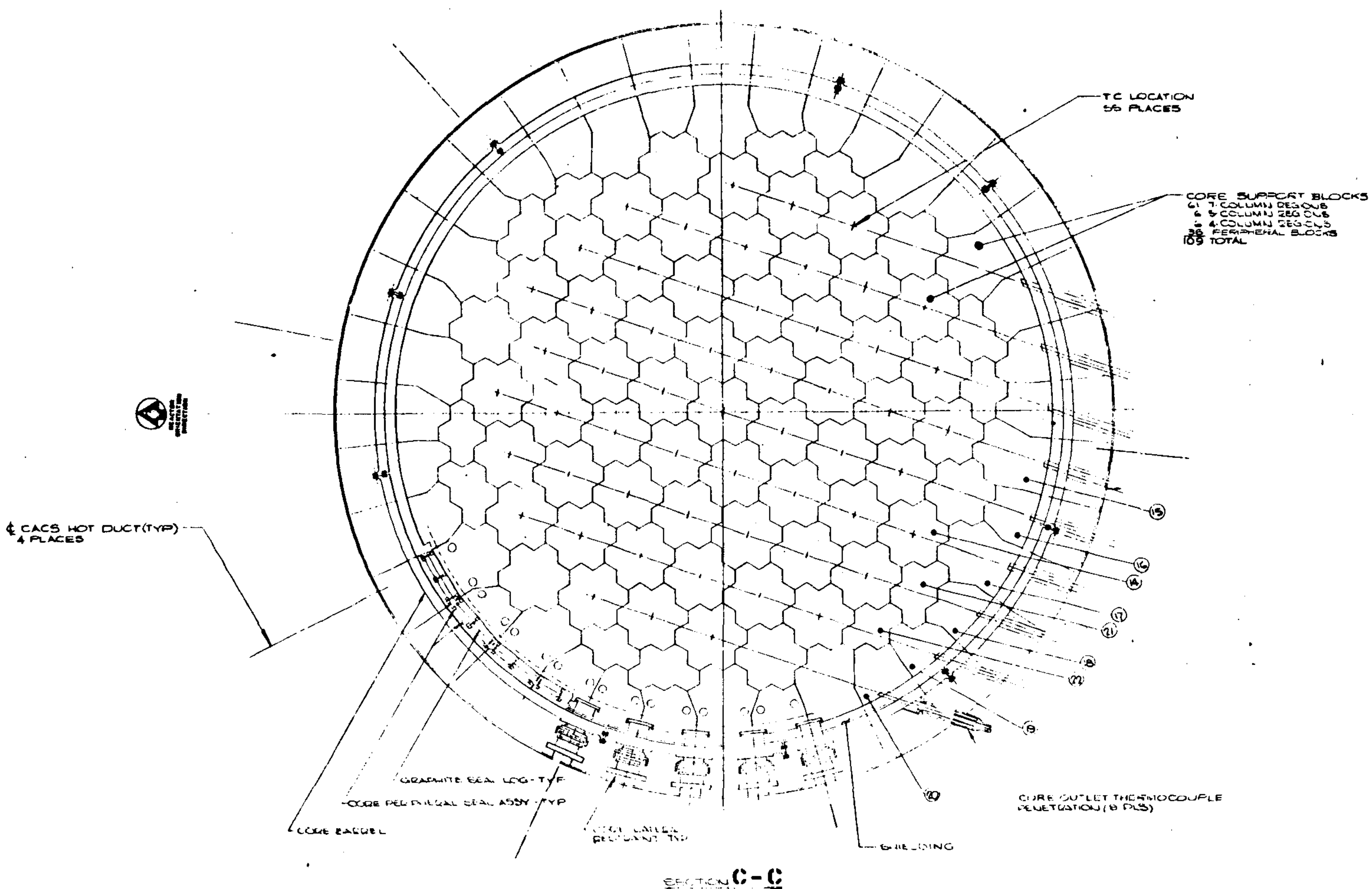
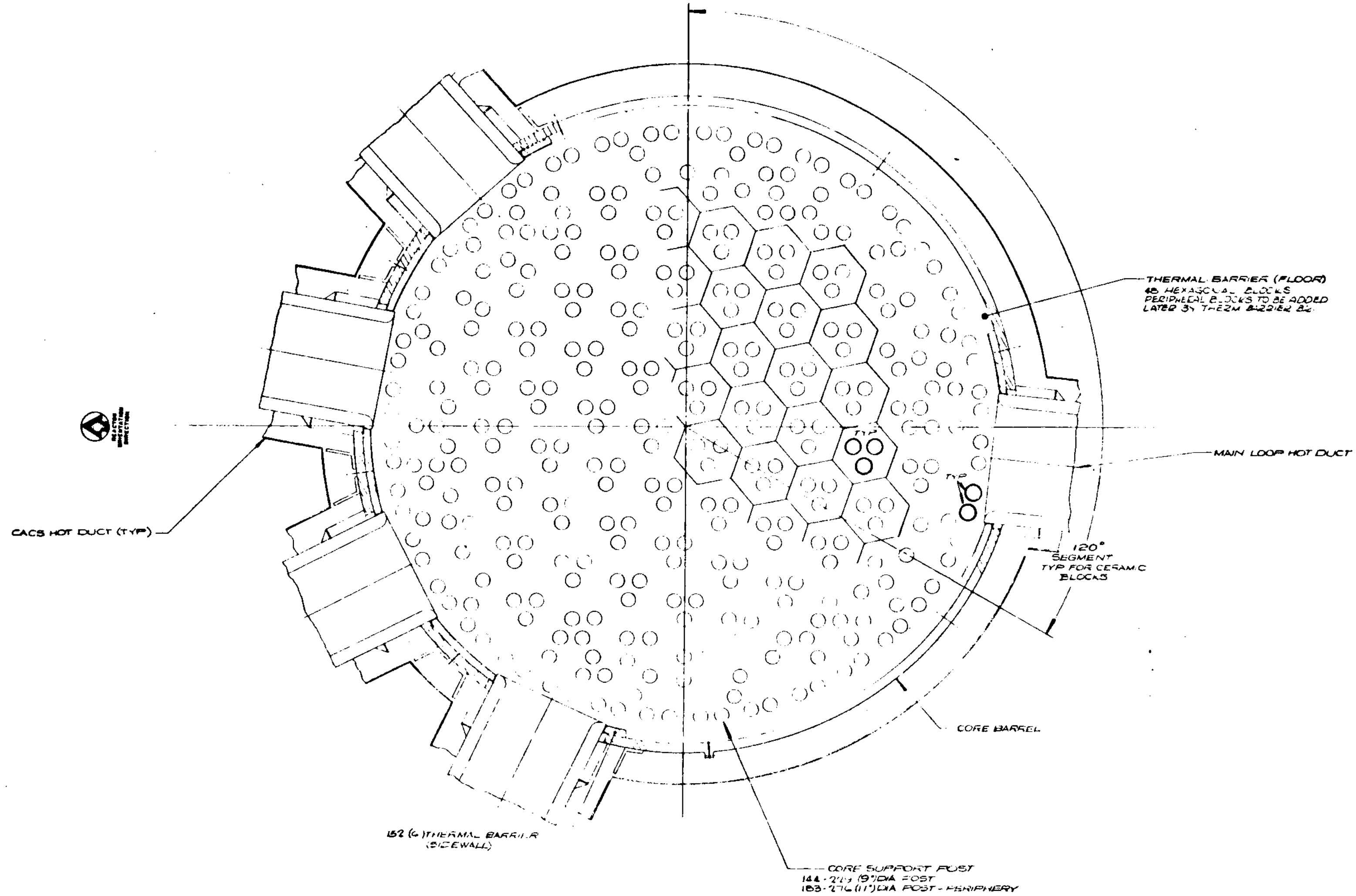


Fig. 17-1. Conceptual design of reactor internals for demonstration plant (sheet 4 of 6)



SECTION D - D

Fig. 17-1. Conceptual design of reactor internals for demonstration plant (sheet 5 of 6)

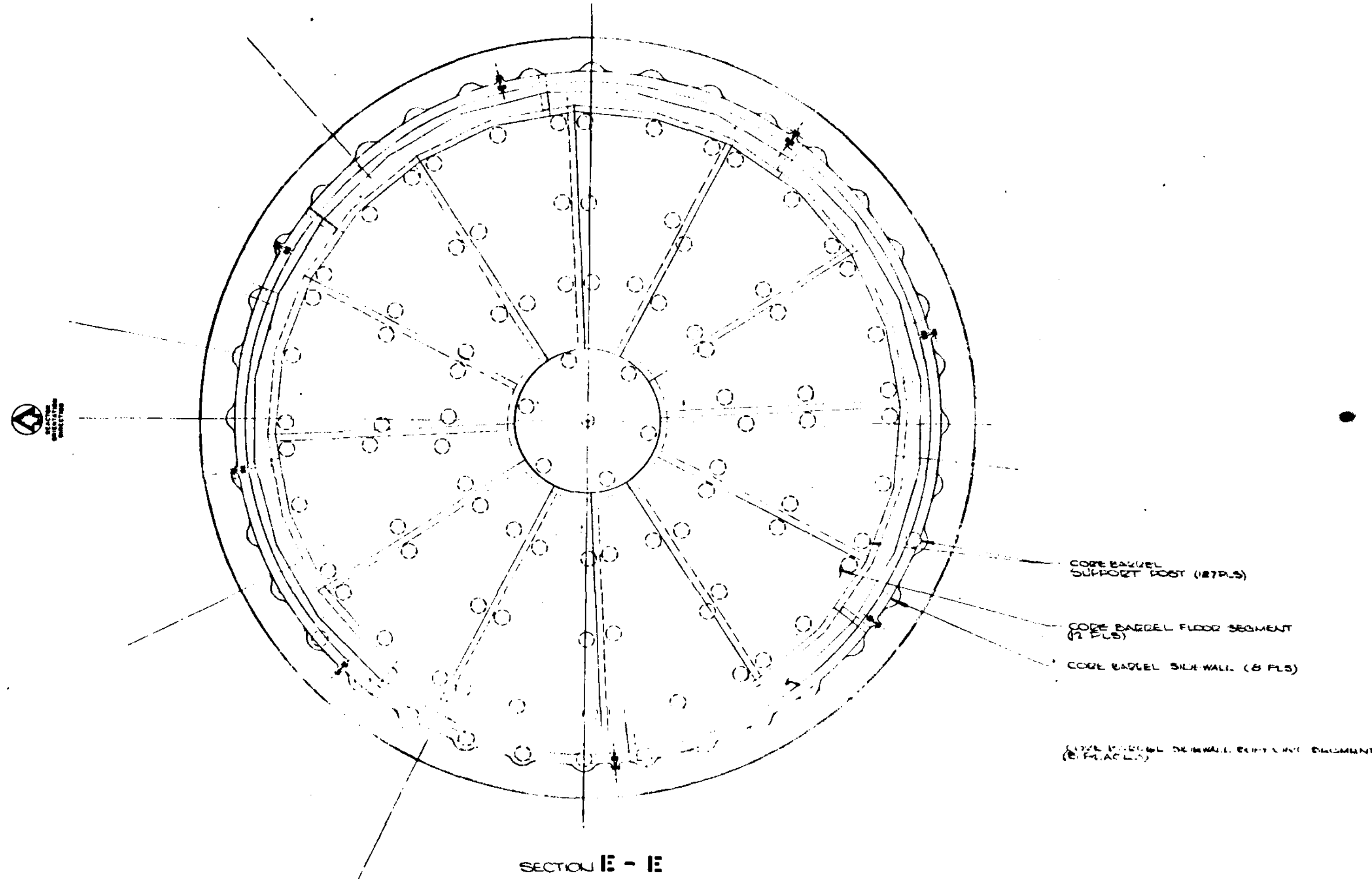
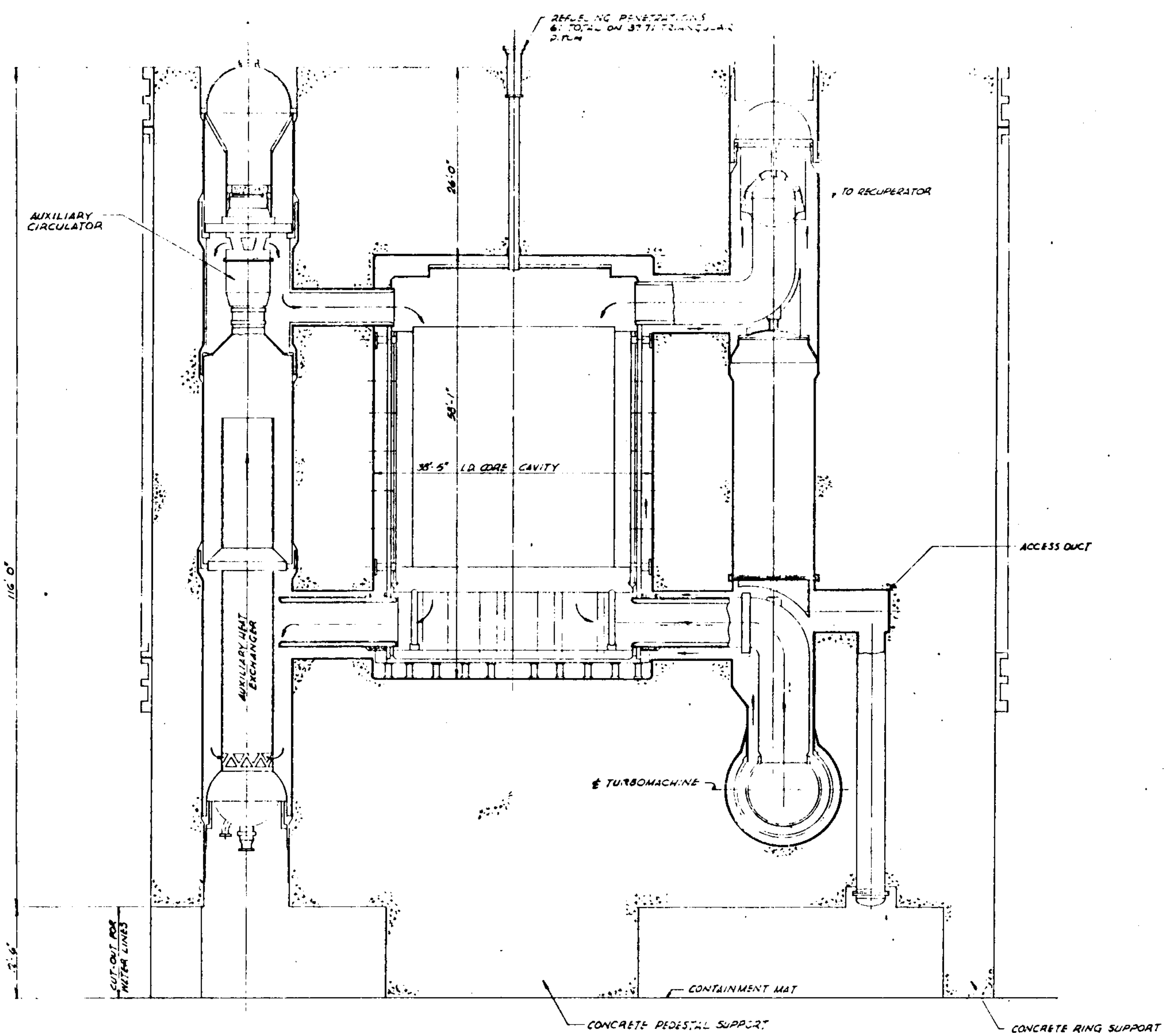


Fig. 17-1. Conceptual design of reactor internals for demonstration plant (sheet 6 of 6)



SECTION A – A

Fig. 17-2. Warm liner concentric ducts (sheet 1 of 2)

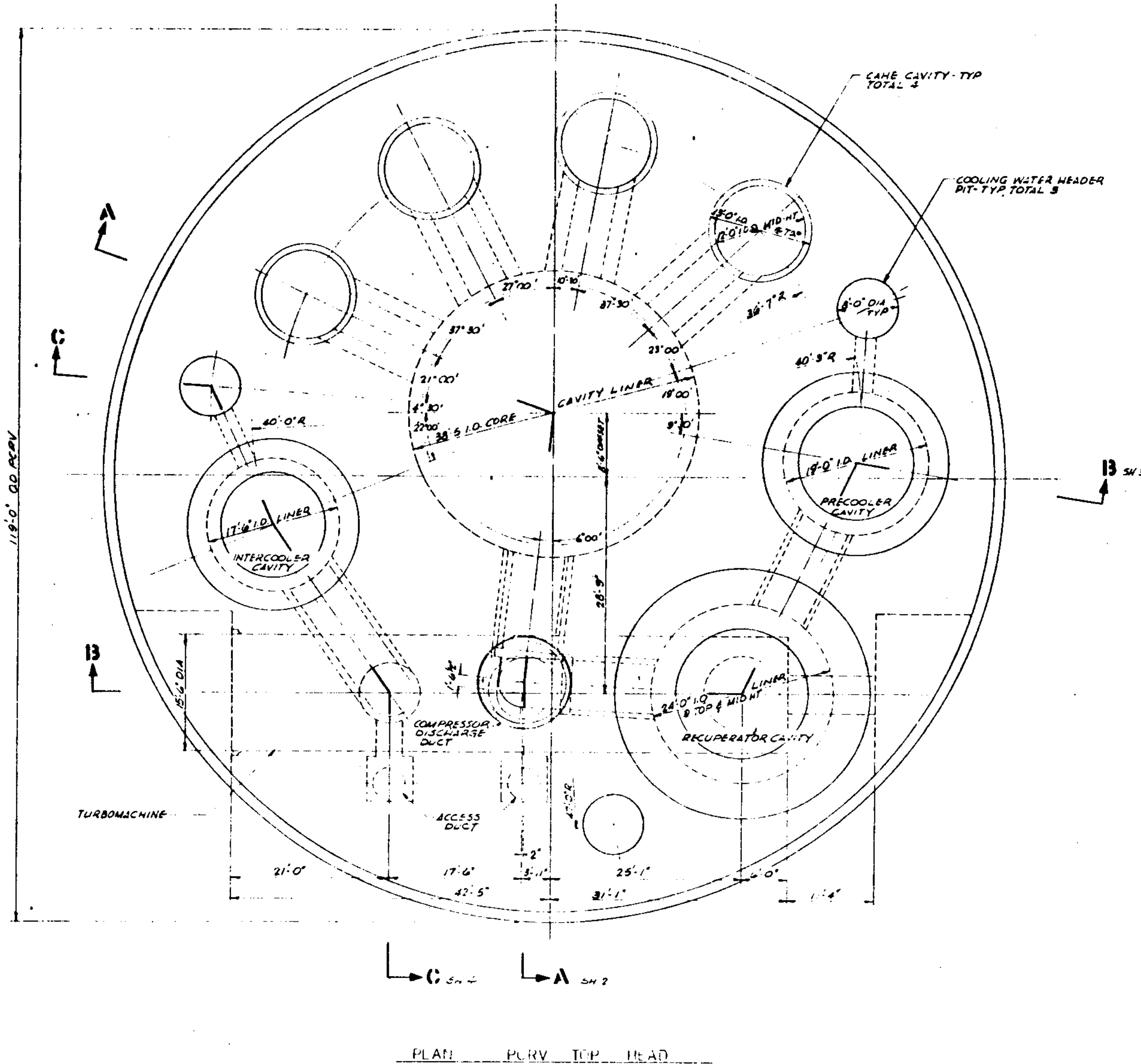
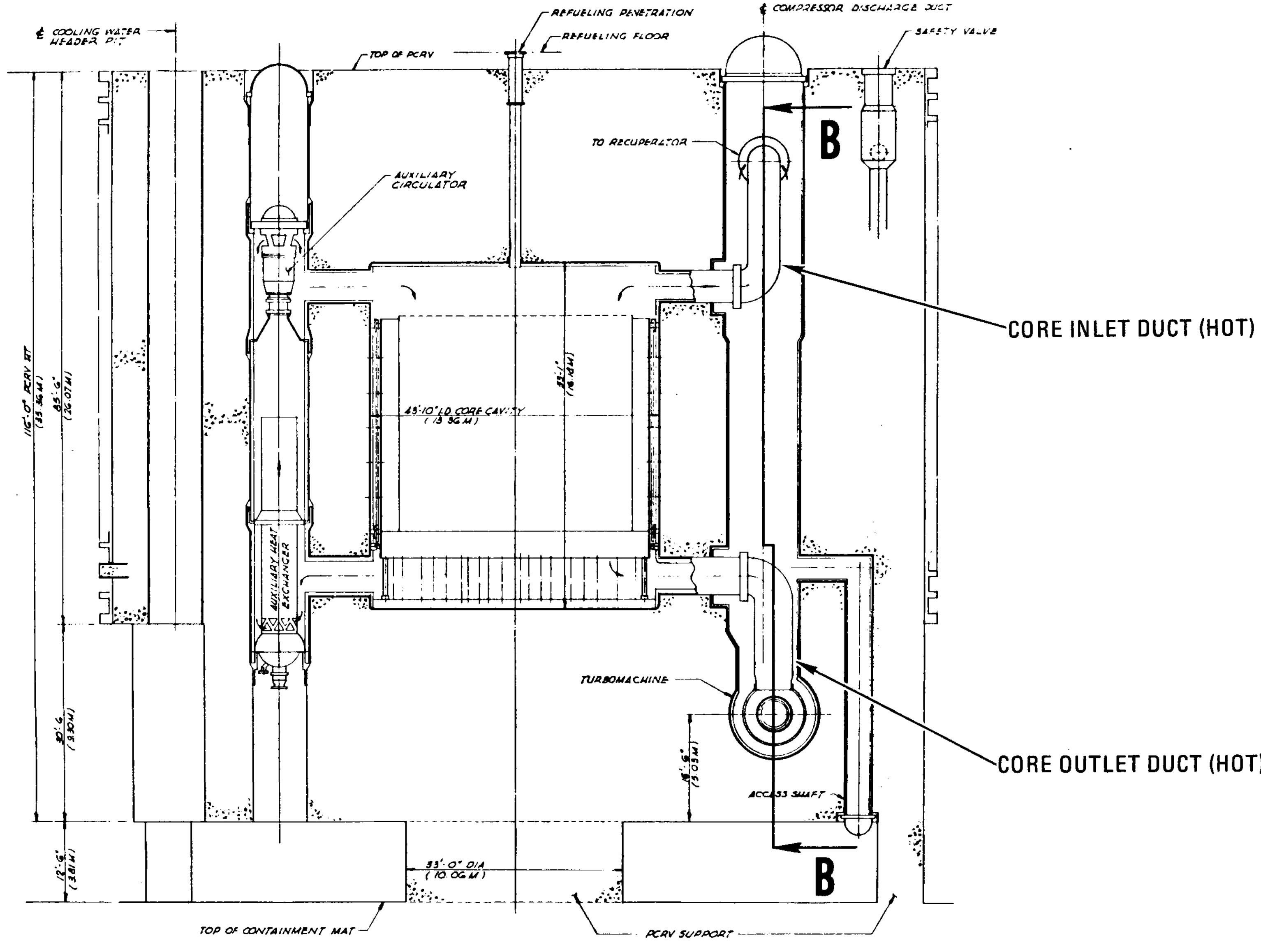
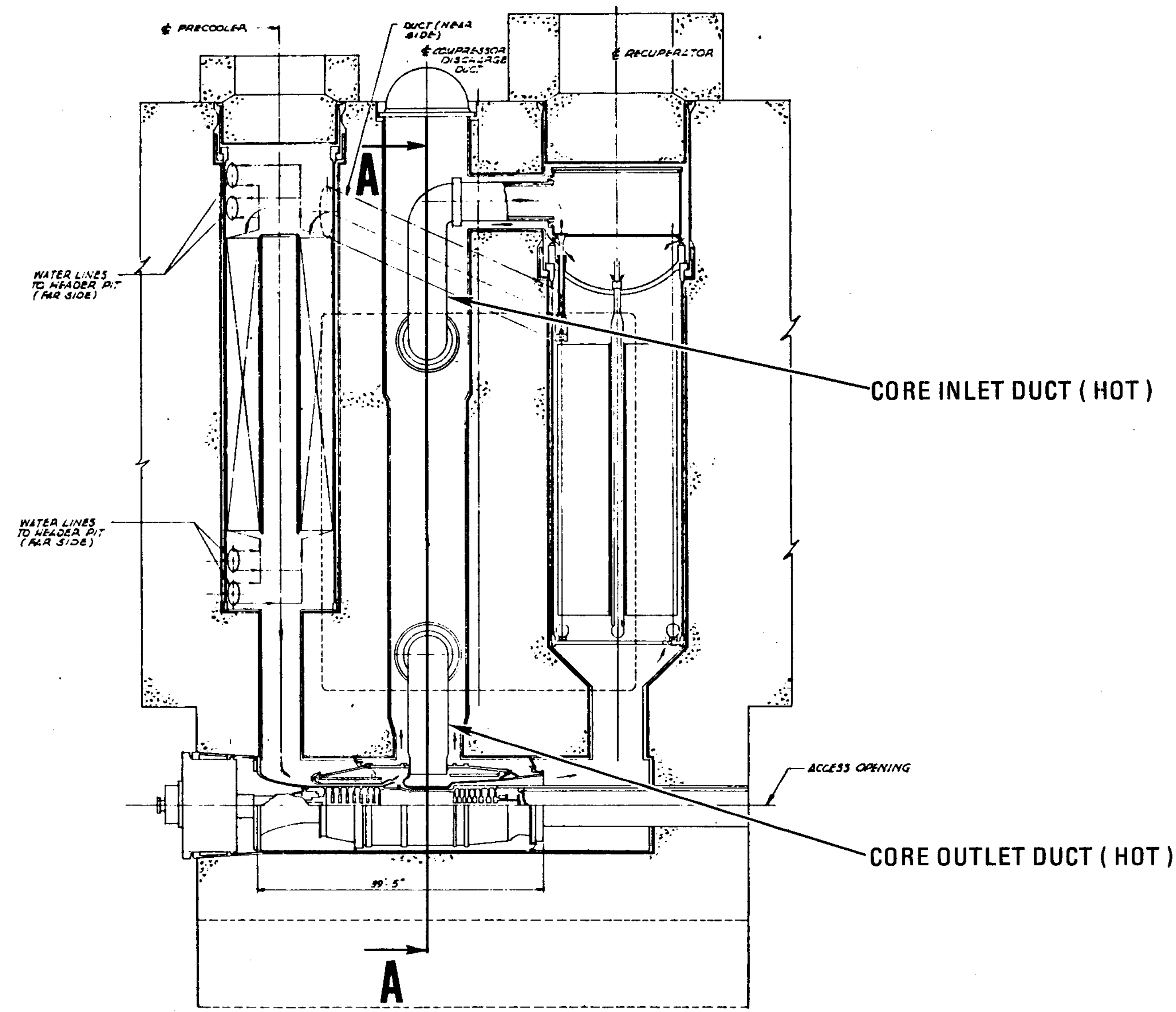


Fig. 17-2. Warm liner concentric ducts (sheet 2 of 2)



SECTION A-A

Fig. 17-3. Cold liner concentric ducts (sheet 1 of 2)



SECTION B – B

Fig. 17-3 Cold liner concentric ducts (sheet 2 of 2) horizontal, free-standing sections connecting the ducts to the core barrel. This results in a penalty for the warm liner configuration. This penalty must be weighed against the potential advantages of the entire free-standing ducts of the warm liner configuration with respect to fabrication, installation, and replaceability.

To summarize, in light of the incomplete information available from the HHT Project and the limited time and manpower allocated to this study, it was not possible to reach a definite conclusion regarding the feasibility of the present design. Instead potential problem areas were identified and recommendations for needed analytical and experimental programs were made.

17.2.2. Commercial Plant

Reactor internals arrangement drawings (Figs. 17-4 and 17-5) for the two- and three-loop GA commercial plants were issued.

Conceptual structural and mechanical designs for the core outlet and inlet ducts were being developed for both two- and three-loop versions until redirection was received, when this task was closed.

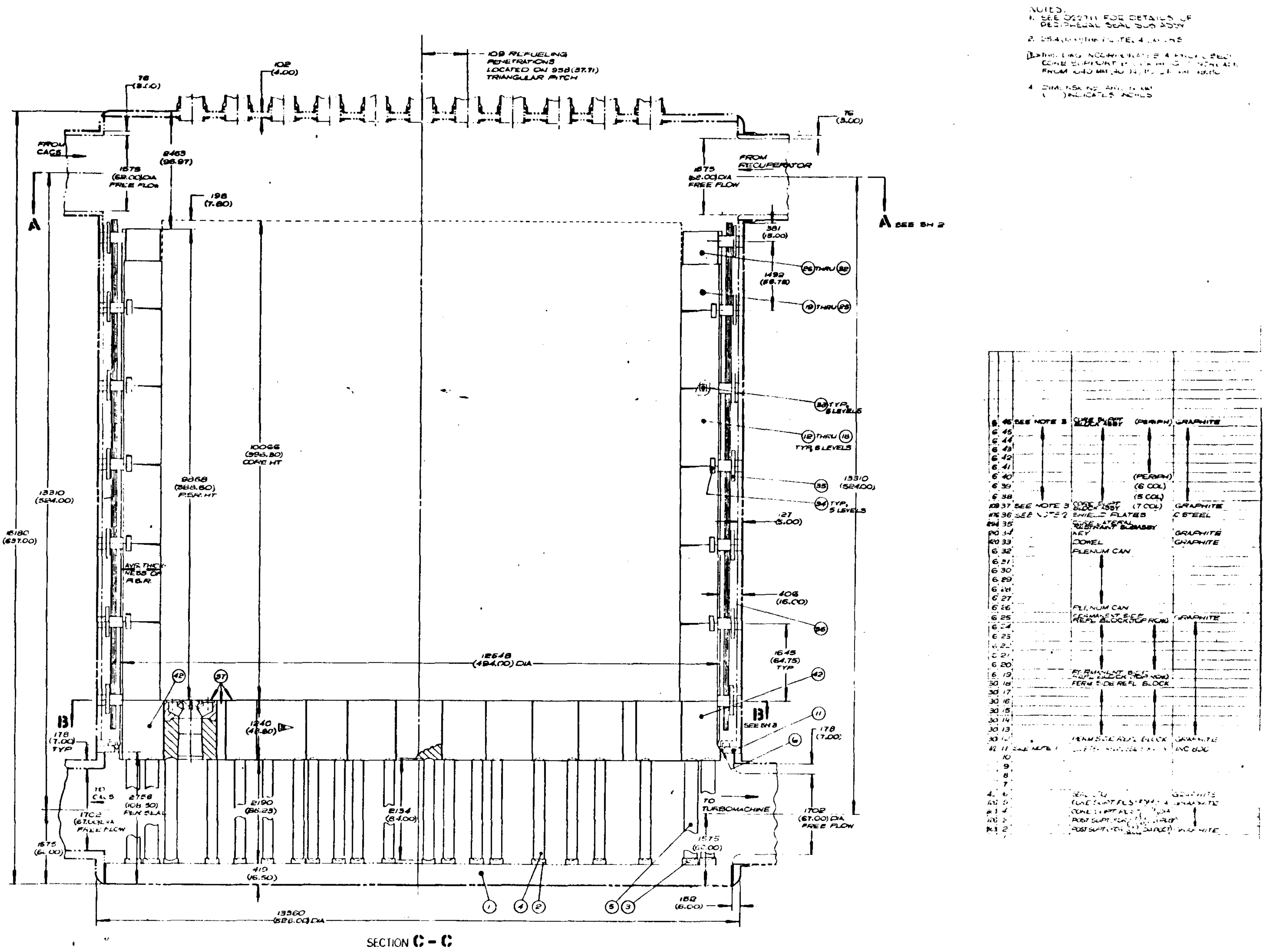


Fig. 17-4. Reactor internals arrangement for two-loop GT-HTGR commercial plant (sheet 1 of 3)

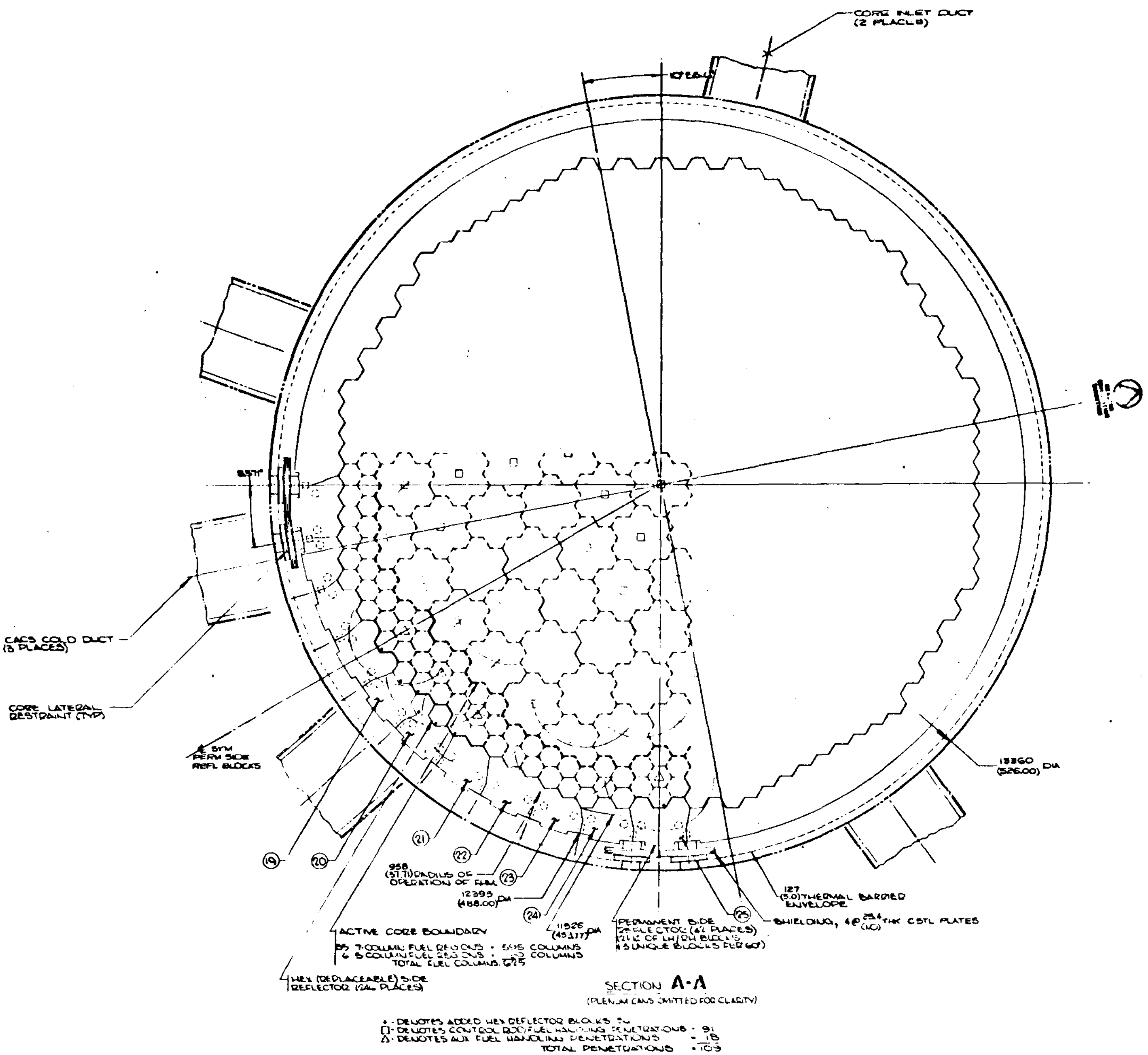


Fig. 17-4. Reactor internals arrangement for two-loop GT-HTGR commercial plant (sheet 2 of 3)

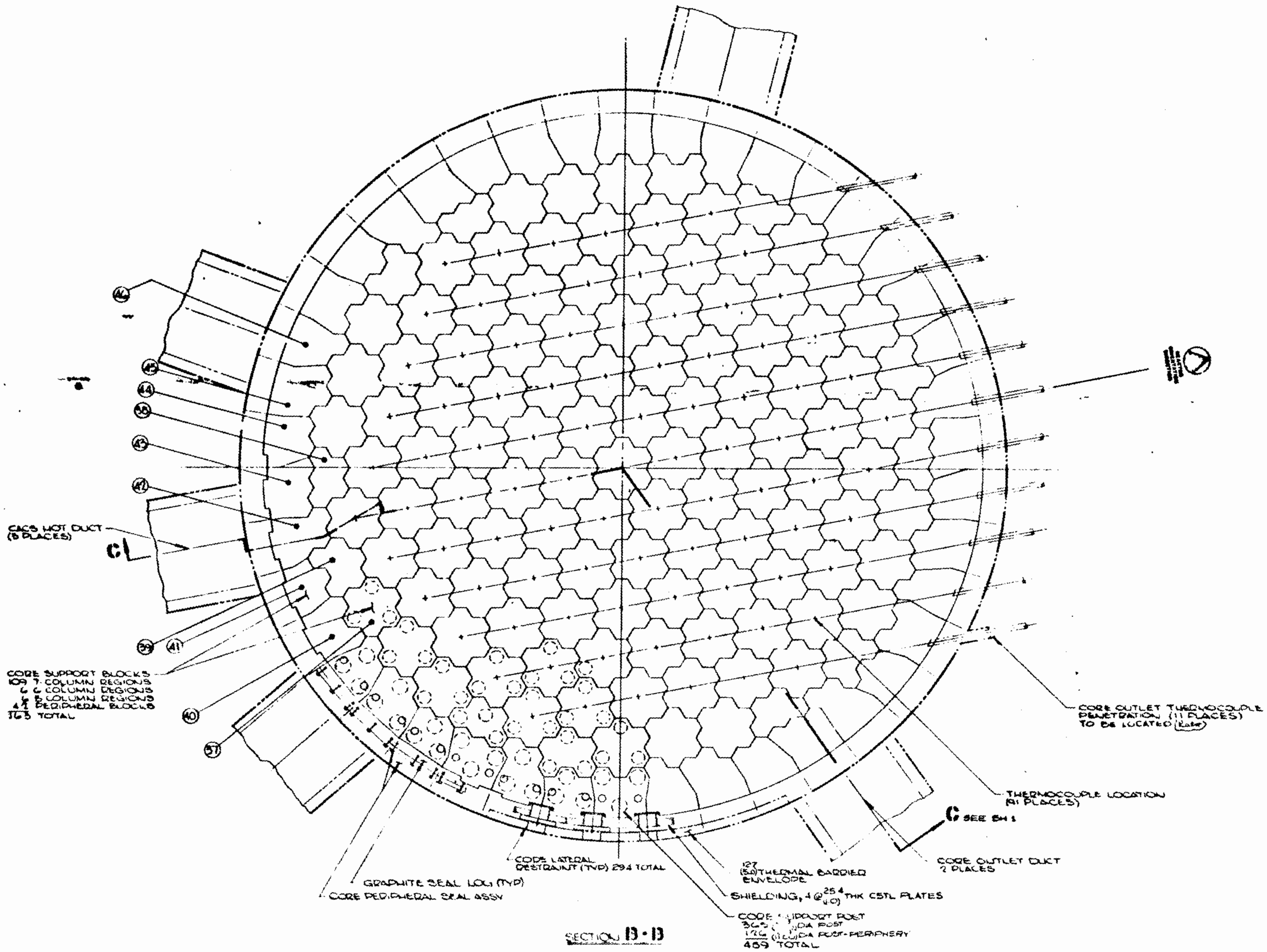


Fig. 17-4. Reactor internals arrangement for two-loop GT-HTGR commercial plant (sheet 3 of 3)

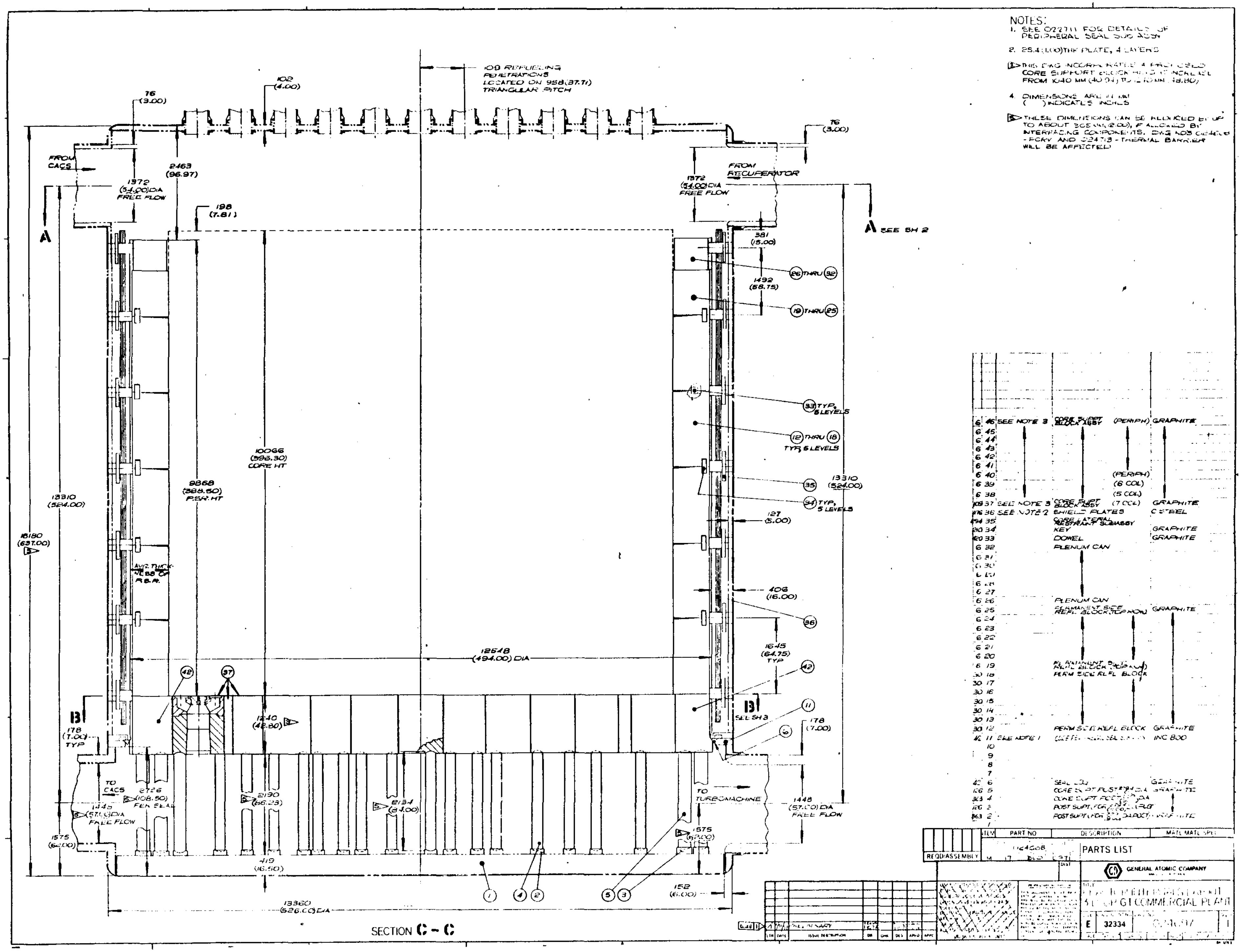


Fig. 17-5. Reactor internals arrangement for three-loop GT-HTGR commercial plant (sheet 1 of 3).

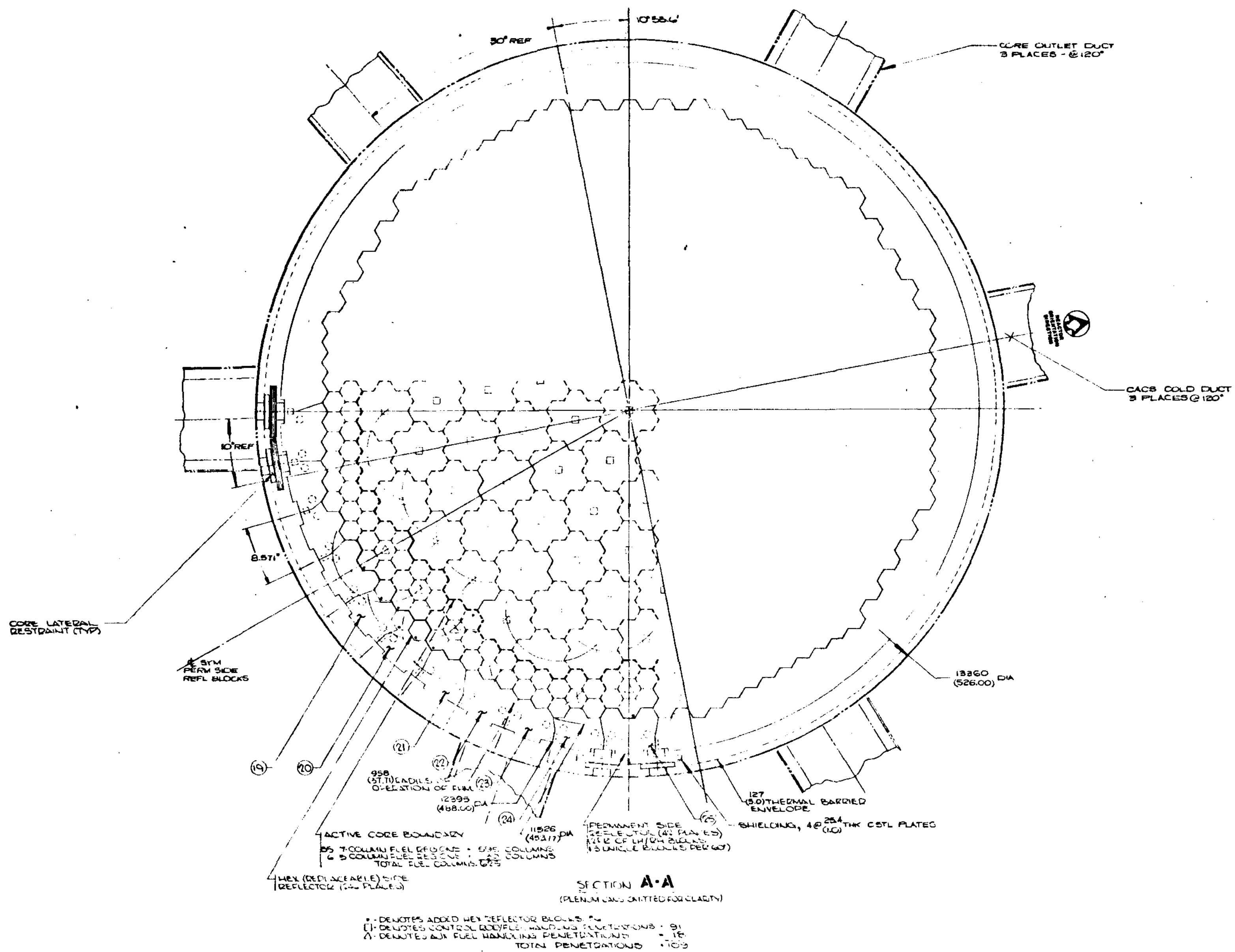


Fig. 17-5. Reactor internals arrangement for three-loop GT-HTGR commercial plant (sheet 2 of 3)

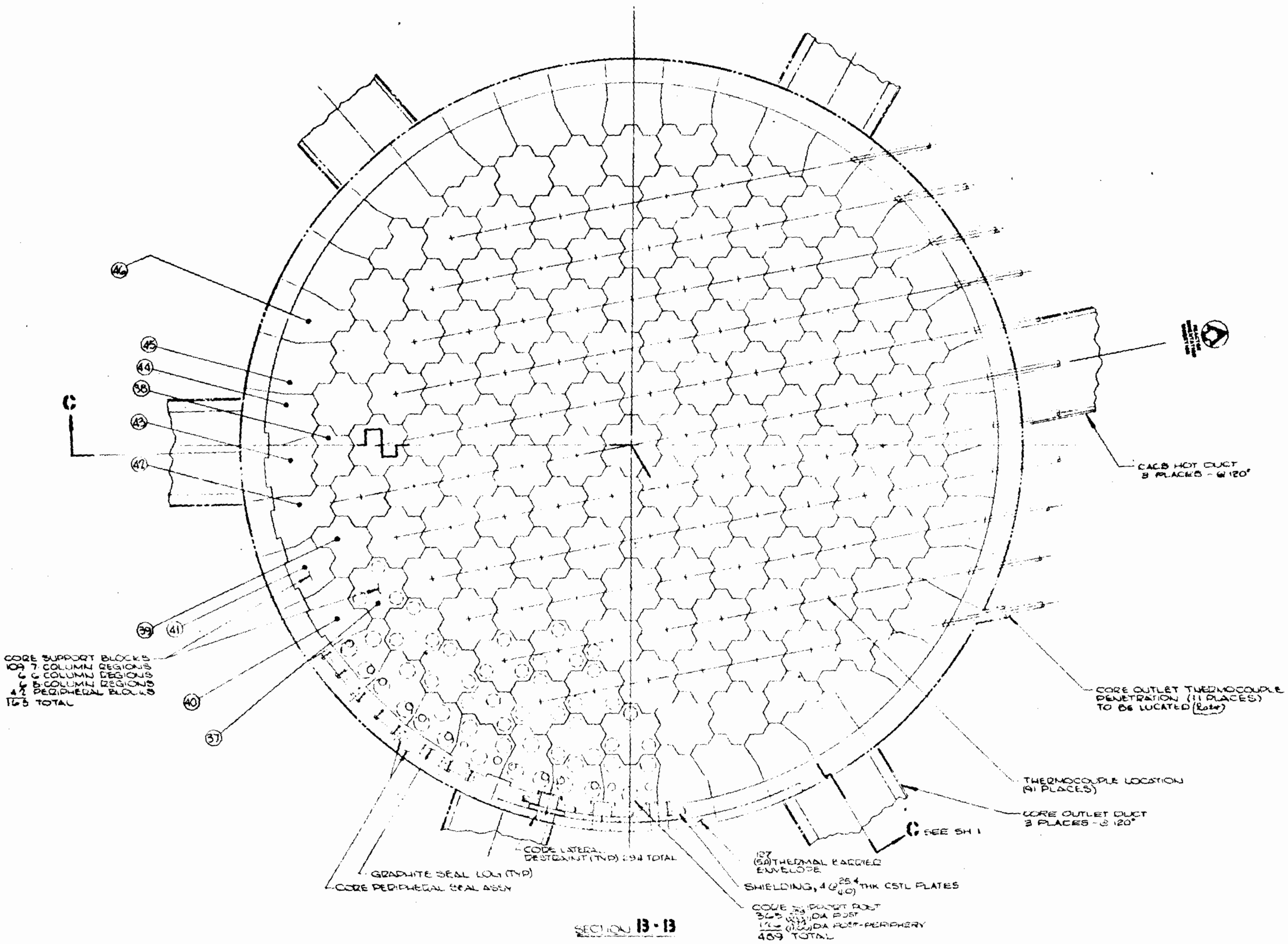


Fig. 17-5. Reactor internals arrangement for three-loop GT-HTGR commercial plant (sheet 3 of 3)

18. TURBOMACHINE (632001)

18.1 SCOPE

UTC will provide conceptual designs for the GT-HTGR turbomachines and generators. Currently, 400-, 500-, and 620-MW(e) designs are under study. Licensing, acoustic, and remote handling requirements will be considered in addition to conceptual machine configurations. The approach proposed to meet the extended operating life goal will include a conservative design and comprehensive quality assurance program. Potential turbomachine testing techniques will be reviewed, and a development plan will be outlined.

18.2 SUMMARY

Conceptual layouts for two-bearing 400-, 500-, and 620-MW(e) turbomachines have been prepared. Double labyrinth buffer seals have been included to ensure there will be no ingress of the lubricating oil into the helium flow path. An alternate scheme based on centrifugal separation was reviewed and rejected. Methods for reducing pressure drops by increasing the turbomachine outer case diameter were studied. Optimized configurations were incorporated into the designs. Sound power levels for the 400-MW(e) machine were estimated. The maximum estimated level of 174 dB occurs at the turbine inlet. Techniques for attenuating this sound level are being reviewed. One method, increased spacing between the rotor and stator, has been included in the 500-MW(e) layout. Critical speed analyses have been performed on both the 400- and 620-MW(e) configurations to ensure resonance-free conditions within the normal operating range.

An approach to remote handling maintenance has been identified. For purposes of this study it was assumed that essentially all disassembly and decontamination would be handled remotely. Major maintenance would be performed at 6-yr intervals. To assess its maintenance advantages, a

full-length split case alternate design was reviewed. However, it was concluded that the necessity for single-piece containment rings around the compressor and turbine precludes this configuration.

Increasing the turbine inlet temperature to reduce system cost requires turbine blade and vane cooling to retain life. The amount of cooling flow for an 82°C (180°F) increase in temperature would be small and of simple design, with slight impact on turbomachine efficiency. Estimated sizes for 400- and 620-MW(e) generators were provided.

18.3. DISCUSSION

18.3.1. 620-MW(e) Turbomachine

A 620-MW(e), intercooled, two-bearing turbomachine conceptual design was prepared in FY-78. Warm and cold liner concepts were prepared. The warm liner configuration is illustrated in Fig. 18-1. An engineering layout is presented in Fig. 18-2.

Operating parameters are presented in Table 18-1. The two-bearing system eliminates the third bearing, which was located between the compressor and turbine on previous 600-MW(e) designs. The third bearing was removed because it was inaccessible for maintenance in the installation. The two outer bearings are accessible through cavities in the PCRV.

A double labyrinth helium buffer seal has also been incorporated to preclude egress of the lubricating oil to the helium flow path.

Critical speed analyses on a number of variations of the two-bearing system were performed. These included:

1. Stiffened generator drive shaft.

2. Stiffened generator shaft with a shaft radial support at the thrust bearing location.

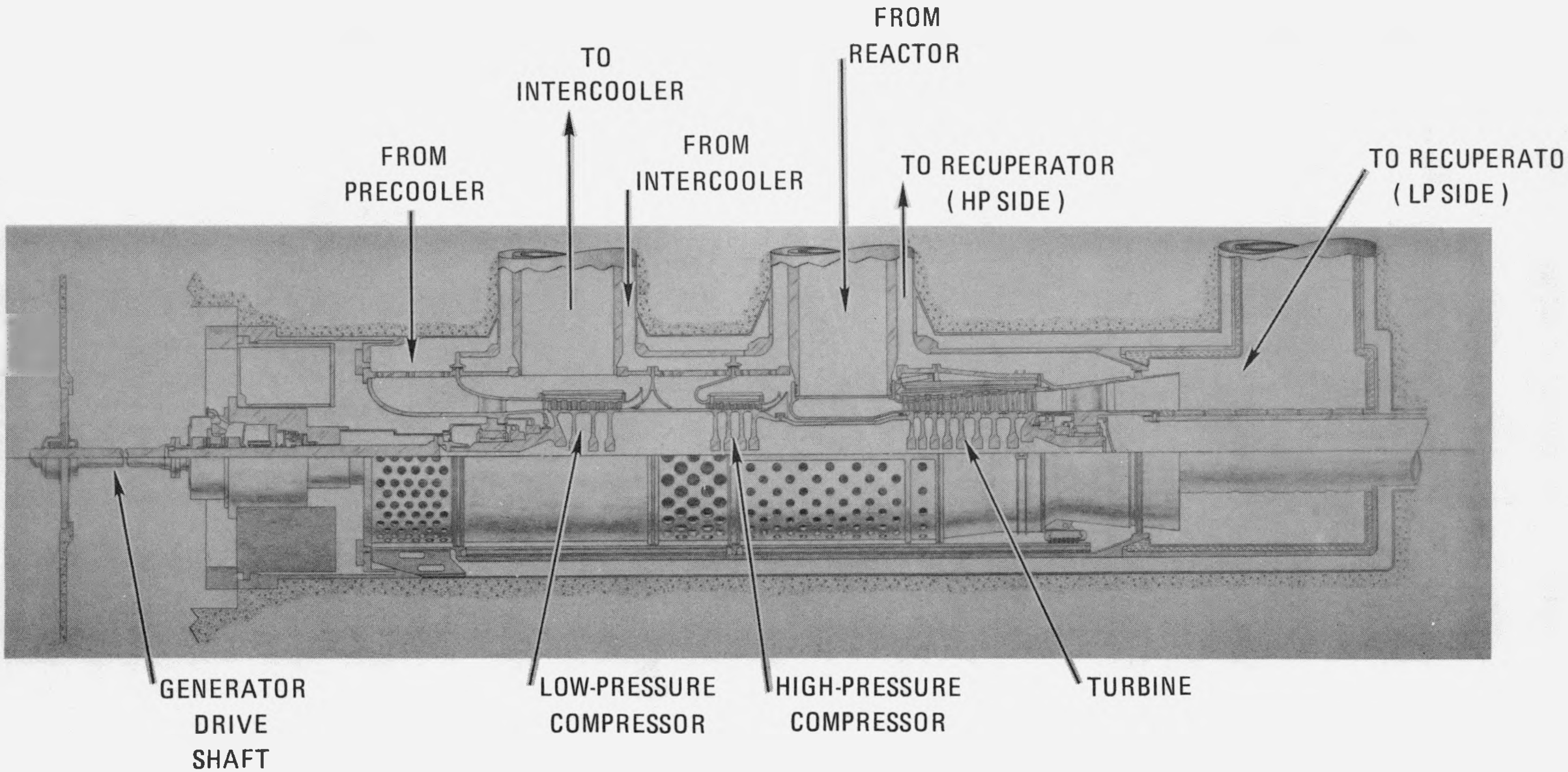


Fig. 18-1. Warm liner configuration for 620-MW(e) turbomachine

TABLE 18-1
OPERATING PARAMETERS FOR 620-MW(e) TURBOMACHINE WITH WARM LINER

	Design Point
Total flow (compressor inlet)	746 kg/s (1650 lbm/sec)
Actual rotor speed	3600 rpm
Overall system pressure loss	9.67%
Low compressor	
Inlet corr. flow	62.52 WAT $\sqrt{\theta T_2}/\delta T_2$
Number of stages	8
Pressure ratio	1.732
Efficiency	0.9080
Inlet temperature	26.8°C (80.3°F)
Intercooler	
Helium side temperature change	63.4°C (146.2°F)
High compressor	
Inlet corr. flow	37.00 WAT $\sqrt{\theta T_2}/\delta T_2$
Number of stages	8
Pressure ratio	1.732
Efficiency	0.9020
Inlet temperature	27.1°C (80.7°F)
Reactor core	
Heat generated in core	1490 MW
Plant heat loss	9.450 MW
Heat supplied to cycle	1483 MW
Turbine	
Efficiency	0.9220
Expansion ratio	2.72
Inlet temperature	848.9°C (1560°F)
Number of stages	9
Helium cooling flow	3.802%
Recuperator	
Effectiveness	0.8975
Precooler	
Helium side temperature change	118.8°C (245.9°F)
Rotor loss	3.95 MW (5295 hp)
Delivered shaft power	637.34 MW (0.85469 x 10^6 hp)
Generator efficiency	0.9880 hp
Gross electric power generated	629.9 MW
Plant auxiliary power requirement	5.500 MW
Net electric power generated	625 MW
Net power plant thermal efficiency	0.4184

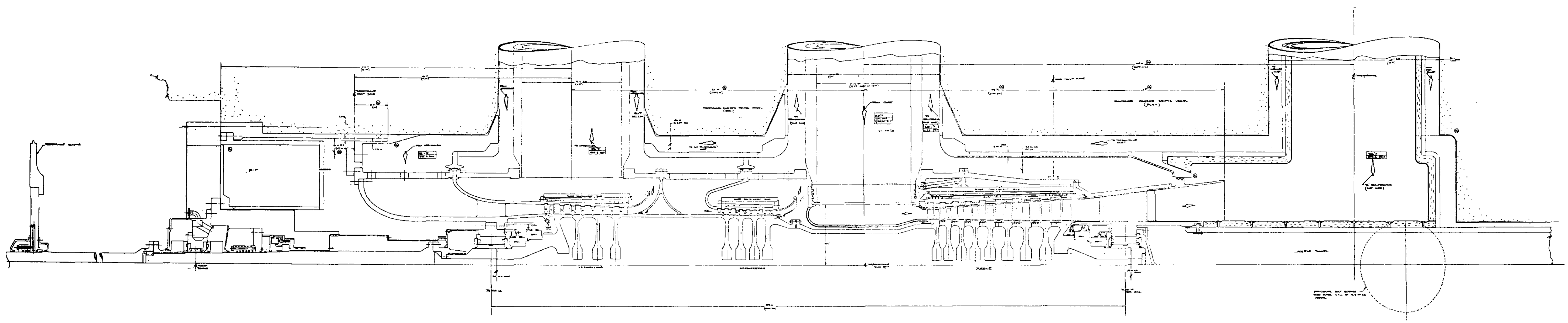


Fig. 18-2. Layout for 620-MW(e) turbomachine warm liner concept

3. Reduced generator shaft length.

4. Turbomachine rotor shortened by 203 cm (80 in.).

The intent was to identify a configuration which eliminates or minimizes critical rotor speeds within the normal operating range. Although none of the above configurations was completely successful, the combination indicated in item 2 lowered the rotor strain energy to 12% in the running range.

The critical speed analysis procedure was reviewed with Professor Stephen Crandall of the Massachusetts Institute of Technology Engineering Department, who is a recognized authority on the dynamics of rotating equipment. He agreed with UTC analytical procedures and the design criteria established for the GT-HTGR turbomachine. His suggestions included:

1. Include speeds below 500 rpm (not analyzed to date).

2. Introduce damping methods into the case and other non-rotating parts of the system, since most of the strain energy appears to reside in the static structure.

3. Avoid damping in the rotor since this may cause hysteresis instability.

4. Avoid energy transfer paths for subcritical resonances.

5. Provide for multi-plane balancing in the rotor design.

To accommodate shipment of the contaminated turbomachine for maintenance, previous designs were restricted to a maximum diameter of 3.5 m (11.5 ft). Relaxation of this constraint would allow a reduction in pressure drop, thereby increasing efficiency. Studies were performed for 4- and 4.6-m (13- and 15-ft) diameters. Results showed that increasing the turbomachine diameter to 4 m (13 ft) would offer a 1/2% improvement in overall efficiency. Further increase from 4 to 4.6 m (13 to 15 ft) offers little in pressure drop reduction. The predicted pressure drop changes for increased diameter are shown in Table 18-2. The associated performance change is shown in Fig. 18-3.

18.3.2. 400-MW(e) Conceptual Design

The conceptual design layout for the 400-MW(e) turbomachine was updated to incorporate the redundant buffering system. The bearing span has been increased from 8.67 m (28.75 ft) to 9.30 m (30.5 ft). To accommodate the revised GA hot duct sealing scheme, the turbomachine outer case diameter has been increased from 3.51 m (11.5 ft) to 3.96 m (13 ft). The flow path is unchanged from the optimized configuration previously determined. The previous configuration is shown in Fig. 18-4 and the revised layout in Fig. 18-5. The operating parameters for this revised configuration are shown in Table 18-3.

18.3.3. 400-MW(e) Turbomachine Pressure Loss and Distortion Reduction

The 400-MW(e) turbomachine flow path was reviewed, and areas have been identified where geometric changes could provide a reduction in pressure loss and flow distortion. Results of this study indicate that optimization in areas such as the compressor inlet duct to plenum intersection and the compressor bellmouth could provide a reduction in pressure loss and in distortion. Reduction in fluid velocities in sections with abrupt area changes and regions where flows are turned, such as the compressor exit and turbine inlet, could provide improved pressure loss and flow distortion. Optimizing the turbine hot duct/turbine volute intersection could reduce the pressure loss in this region. To provide optimized geometries for areas such as the compressor inlet and exit and the turbine inlet, scale model testing is recommended. Table 18-4 presents the estimated pressure losses at the critical locations for the present layout configuration.

Basically, the suggestions identified in this study are geometric changes that provide lower fluid velocities and/or improved interaction

TABLE 18-2
EFFECT OF TURBOMACHINE DIAMETER ON PRESSURE DROP

	Cycle Pressure Losses, ΔP/P (%)		
	Baseline [3.5-m (11.5-ft) Diameter]	4-m (13-ft) Diameter	4.6-m (15-ft) Diameter
LPC[a] inlet interface and shell holes	0.13	0.13	0.10
LPC inlet volute	0.20	0.20	0.20
LPC exit diffuser and dump	1.07	0.58	0.58
LPC exit contraction	0.05	0.05	0.05
HPC[b] inlet interface and shell holes	0.07	0.07	0.07
HPC inlet volute	0.20	0.20	0.20
HPC exit diffuser and dump	1.18	0.71	0.71
HPC exit shell holes	0.03	0.03	0.02
HPC exit contraction	0.02	0.02	0.02
Turbine inlet volute	0.48	0.58	0.26
Turbine exit diffuser and struts	0.37	0.37	0.37
Turbine exit contraction	0.02	0.02	0.02

(a) Low-pressure compressor.

(b) High-pressure compressor.

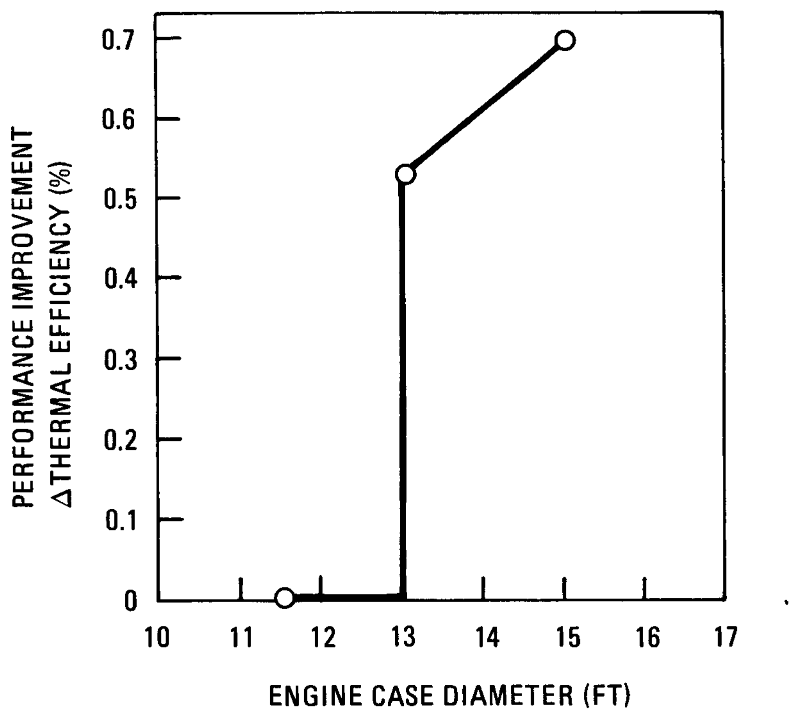


Fig. 18-3. Potential performance improvement with 620-MW(e) turbomachine case diameter increase

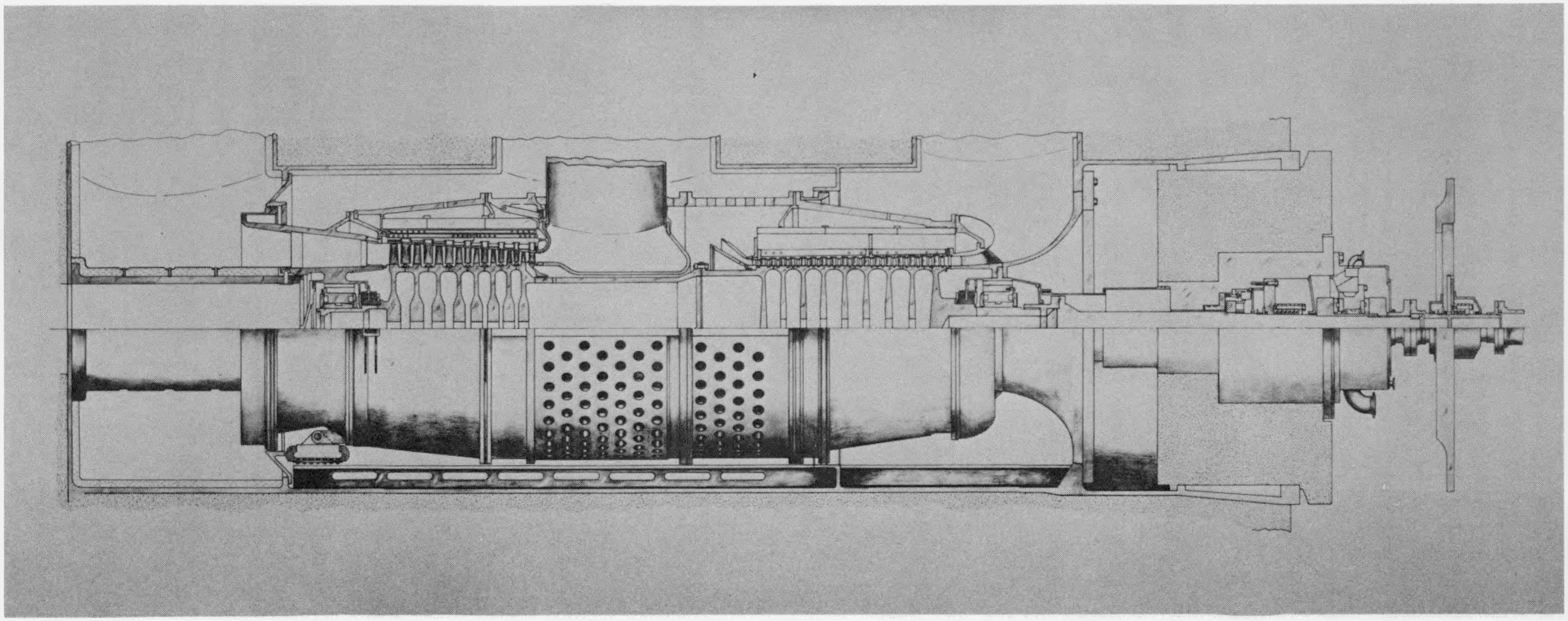


Fig. 18-4. Previous 400-MW(e) turbomachine configuration

TABLE 18-3
OPERATING PARAMETERS FOR REVISED 400-MW(e) TURBOMACHINE

	Design Point
Total flow (compressor inlet)	571 kg/s (1260 lbm/sec)
Actual rotor speed	3600 rpm
Overall system pressure loss	7.10%
Compressor	
Inlet corr. flow	41.08 WAT $\sqrt{\theta T_2}/\delta T_2$
Number of stages	18
Pressure ratio	2.5
Efficiency	89.8%
Inlet temperature	26.7°C (80°F)
Reactor core	
Heat generated in core	970.65 MW
Plant heat loss	6.25 MW
Heat supplied to cycle	964.4 MW
Turbine	
Efficiency	91.8%
Expansion ratio	2.322
Inlet temperature	850°C (1562°F)
Number of stages	8
Helium cooling flow	3.6%
Recuperator	
Effectiveness	89.8%
Precooler	
Helium side temperature change	180.7°C (357.3°F)
Rotor loss	2.32 MW (3110 hp)
Delivered shaft power	405.45 MW (0.5437 x 10^6 hp)
Generator efficiency	98.7%
Gross electric power generated	400.18 MW
Plant auxiliary power requirement	1.25 MW
Net electric power generated	398.93 MW
Net power plant thermal efficiency	41.10%

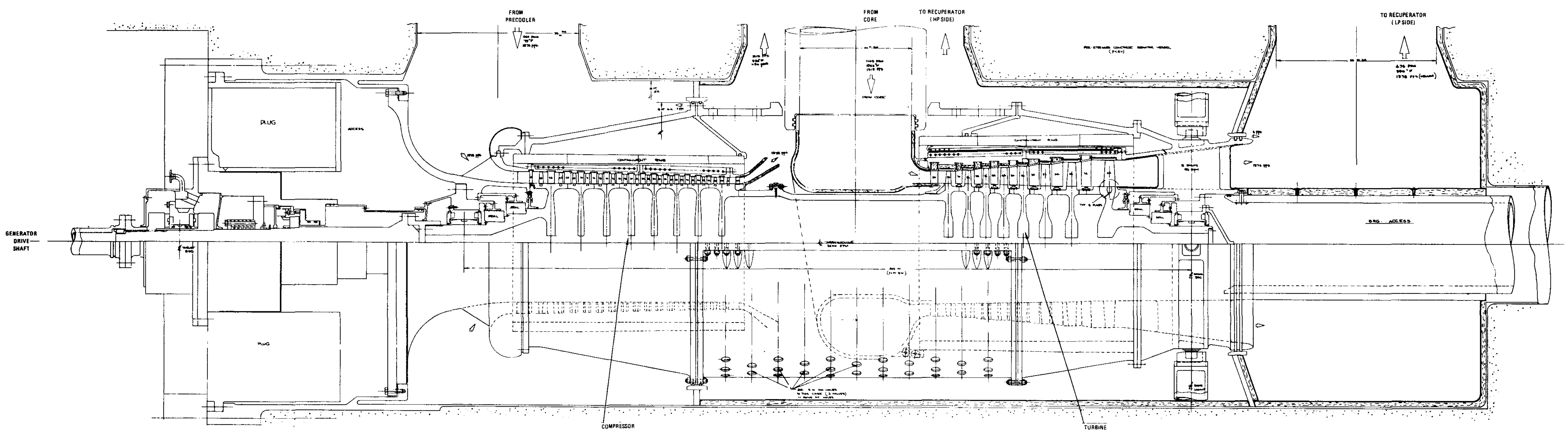


Fig. 18-5. Revised 400-MW(e) turbomachine configuration

TABLE 18-4
400-MW(e) TURBOMACHINE PRESSURE LOSSES [$\Delta P/Pt(\%)$]

Section		
Compressor inlet		0.22
Compressor exit		0.70
Diffuser	-0.46	
Diffuser dump	-0.23	
Shell holes	-0.015	
	-0.705	
Turbine inlet		0.41
Turbine exit		0.15
Diffuser	-0.092	
Struts	-0.053	
	-0.145	
Total		1.48

between components (e.g., ducts and volutes, volutes and bellmouth, etc.). For the most part, those features that result in lower pressure losses also tend to result in lower distortion. Although the potential advantages of these recommendations is easily recognizable, the exact geometric configurations must be optimized through model testing and trade-off studies.

The items recommended for consideration for follow-up configuration updating are described below.

18.3.3.1. Compressor Inlet.

Pressure Loss

Suggested compressor inlet pressure loss improvement features are as follows:

1. Increase the inlet plenum size by incorporating part of the plenum in the PCRV wall or by increasing the engine case diameter.

2. Optimize the inlet duct/plenum intersection.

3. Optimize the bellmouth geometry. Model tests would be required.

4. Add a second inlet duct.

Distortion

In the absence of inlet duct wakes, as was the case with the 620-MW(e) turbomachine warm liner concept, it is reasonable to assume that those features which reduce pressure loss also result in a reduction in distortion. It is difficult to quantify the distortion reduction at present, and model testing would be required to provide an assessment. Model test studies showed that optimizing the compressor inlet duct/plenum intersection and bellmouth geometry could result in a significant improvement as measured by circumferential pressure variation (i.e., $P_{max} - P_{min}/P_{avg}$.).

18.3.3.2. Compressor Exit.

Pressure Loss

The present curved annular diffuser geometry is probably optimum for this type of diffuser regardless of the space available, since the diffuser area ratio and diffuser length to diffuser inlet height are the real constraint. Suggested features to be considered for loss improvements are:

1. Use a radial diffuser, since it has potential for a larger area ratio and accompanying lower dump velocity. Model tests would be required to provide an optimum design.

2. Increase the turbomachine case diameter to provide increased shell hole flow area.

Distortion

As for the compressor inlet, those features which reduce pressure loss also reduce distortion. In general, inlet distortion is far more significant in terms of surge margin reduction and vibration than exit distortion.

18.3.3.3. Turbine Inlet.

Pressure Loss

Suggested turbine inlet pressure loss improvement features are:

1. Increase the turbine hot duct size. This would reduce the sudden expansion loss from the hot duct to the turbine inlet volute.

2. Optimize the hot duct/turbine volute intersection. Model tests would be required.

3. Add a second hot duct.

Distortion

Reduction in velocities in regions where flow is turned (i.e., hot duct to turbine volute and turbine volute to turbine inlet) will result in reduced distortion. Again, the features suggested for improved pressure loss will also provide improved distortion. Model testing would be required to quantify the baseline distortion level and any benefits from an improved flow path, since no other evaluation techniques are currently available.

18.3.3.4. Turbine Exit.

Pressure Loss

Since the turbine exit does not have any provisions for the removal of a swirl, a trade-off study needs to be conducted which compares losses associated with an exit vane with potential diffuser loss reduction if swirl is optimized. In addition, the strut contour and location could be optimized. In both of these areas, model testing would be important.

Distortion

The impact of distortion on the turbine exit is probably not significant.

18.3.4. 400-MW(e) Turbomachine Noise Estimates

Estimates were obtained for the acoustic power emissions from the inlet and discharge of the GT-HTGR turbomachine compressor and turbine. Tests on a two-stage model fan operating in air provided the baseline acoustic data. Procedures were developed and used to transform the baseline data to the

GT-HTGR component overall acoustic powers. These scaling procedures involve use of blade tip Mach number as the primary variable together with modifications to account for changes between baseline and GT-HTGR components in diameter, hub-tip ratio, blade loading, blade-vane geometry, and media densities and sound speeds. Estimates of the spectral distribution of acoustic power were also obtained.

The results are in close agreement with those presented in previous GA studies which assumed that only the outermost stage is the noise emitter emitter. The two-stage baseline tests used here indicate that at least the two outermost stages contribute. Therefore, to make comparison as close as possible, 3 dB have been added to the GA values and the resulting emissions are presented together with the current independent estimates in Table 18-5.

Two main parts of the procedure for estimating inlet and discharge acoustic power emissions for the GT-HTGR turbomachine compressor and turbine are (1) obtaining relevant data from geometrically representative machinery and (2) determining how to scale this information. These are discussed below.

Baseline Data

After examination of several possible test program results, it was decided to use the results of extensive tests on an 83.8-cm (32.9-in.) diameter two-stage fan Q2S (Ref. 18-1). It was believed that a two-stage machine would be more representative than a single-stage machine for the purpose of this study. Apart from size and number of stages, the two-stage fan differed in two significant ways from GT-HTGR geometry: (1) blade/vane ratios and (2) blade/vane spacing.

Recognition of these differences established overall sound power levels for inlet and discharge emission that would provide baselines for scaling the modified Q2S fan data to each of the four emitters in the GT-HTGR

TABLE 18-5
400-MW(e) GT-HTGR TURBOMACHINE COMPONENT ACOUSTIC POWER EMISSIONS
(OVERALL POWER LEVELS: dB RE 10^{-12} W)

	Compressor Inlet	Compressor Discharge	Turbine Inlet	Turbine Discharge
Present estimates	151.7	150.3	161.5	162.2
GA Est. + 3 dB	152.7	150.6	161.8	163.8
Difference	1	0.3	0.3	1.6

turbomachinery. This information (without the hub-tip change effect) is represented in Fig. 18-6 in the form of overall sound power levels as a function of blade tip circumferential Mach number.

Scaling Parameters

Sealing from the Quiet Fan to the GT-HTGR was accomplished by considering:

1. Blade loading.
2. Hub-tip ratio.
3. Relative rotor size.
4. Shaft power in helium versus air.

The results of this study are shown in Table 18-6.

In addition to the overall power levels, estimates were prepared for the spectral distribution of power shown in Fig. 18-7. The process involved selecting representative one-third octave band levels to typify results for the inlet and for the discharge of the Q2S baseline fan at low operating Mach numbers and modifying these to account for cut-off and spacing changes expected in the GT-HTGR components.

Previous GA analysis was made of the sound pressure levels to be expected in several parts of the reactor system as a consequence of the turbomachine acoustic power emissions. The GA analysis indicated that the resulting pressures would be only marginally acceptable in some regions. The following procedures are suggested for consideration for reducing the acoustic power generation:

1. As a preliminary step, baseline data supplementary to that of the Q2S fan used here should be examined to allow more reliable estimates to be made. These data may be processed by the methods described here or by another, independent procedure. Further,

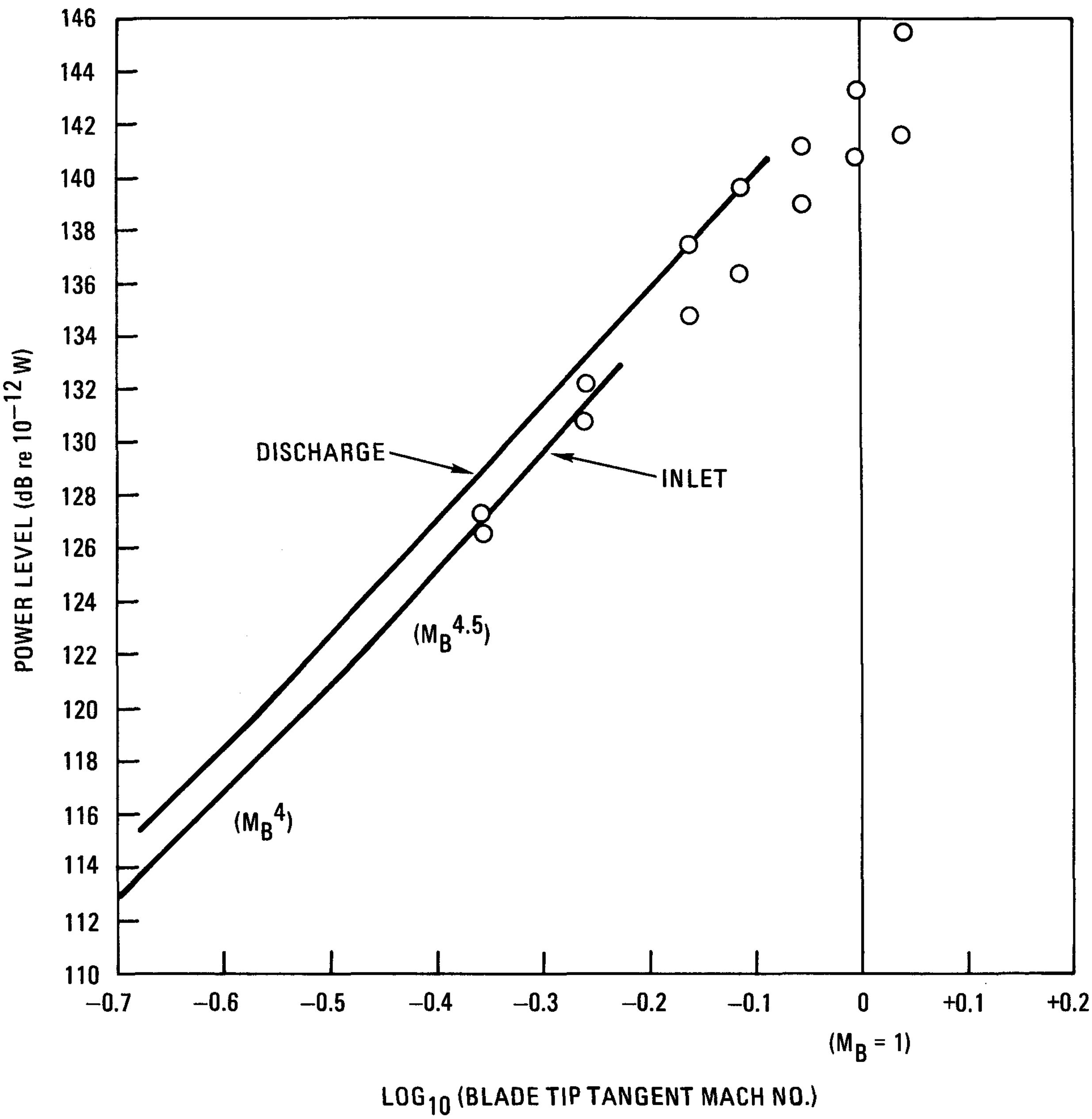


Fig. 18-6. Quiet two-stage fan baseline overall power levels (1/3 octave bands containing blade tones inceased 6 dB to estimate effects of close spacing and cut-on blade-vane combinations)

TABLE 18-6
PARAMETERS AND RESULTS OF GT-HTGR TURBOMACHINERY NOISE ESTIMATE STUDY

	Compressor		Turbine	
	Inlet	Discharge	Inlet	Discharge
GT-HTGR temperature [°C (°F)]	26.3 (79.4)	177 (350)	850 (1562)	532 (990)
GT-HTGR pressure [MPa (psia)]	3.17 (460)	7.93 (1150)	7.65 (1109)	3.28 (476)
Density ratio referred to SLS air	4.4	7.3	2.8	1.7
Sound speed ratio referred to SLS air	2.93	3.59	5.68	4.78
Diameter [cm (in.)]	182.9 (72)	176.8 (69.6)	198.6 (78.2)	218.4 (86)
Hub-tip ratio	0.87	0.90	0.86	0.76
Power per two stages [MW (hp)]	56 (75,000)	56 (75,000)	225 (302,000)	225 (302,000)
Blade-tip mach number	0.34	0.27	0.19	0.25
Q2S power [MW (hp)]	0.15 (200)	0.08 (100)	0.03 (35)	0.06 (79)
Q2S power including scaling and hub tip [MW (hp)]	0.20 (274)	0.10 (136)	0.04 (56.5)	0.20 (270)
Q2S sound pressure level (numerically = PWL) (dB)	122	119.4	112	118.6
Q2S sound pressure level scaled to size and hub tip (dB)	123.4	120.8	114.1	123.9
GT-HTGR sound pressure level = above plus HP scaling (dB)	162.8	164.5	173.6	171.3
GT-HTGR corresponding acoustic power (dB re 10^{12} W)	151.7	150.3	161.5	162.2
Acoustic power (W)	1,480	1,070	14,100	16,600

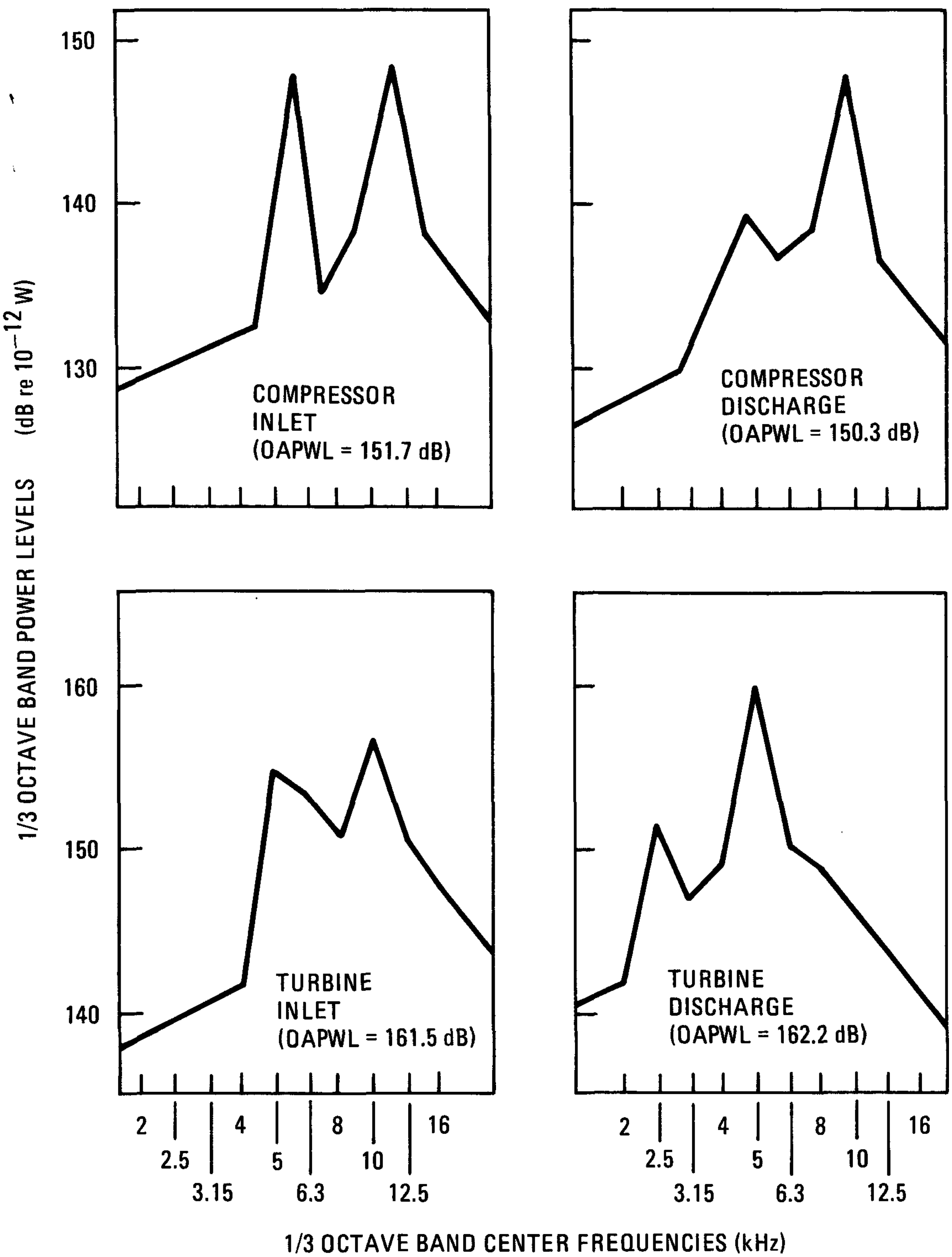


Fig. 18-7. GT-HTGR component estimated power distribution spectra

the spectral distribution should be examined more closely since this factor will influence the severity of the resultant pressure problem. At the very least, estimates of the errors in the prediction process should be sought. Presently, it seems that these estimates could be in error by something like ±6 dB.

2. With the results of step 1 in hand, the required nature and amount of source noise reduction can be established to use in evaluating possible design modifications.

3. The following is a list of some noise-reduction methods that might be applicable. All of these concepts have worked in certain applications, but none of them will always work. The efficacy of any measure depends greatly on details of the operating environment, and the practicability of a method also depends on related factors. Some possibilities for exploration are as follows:

   a. Change the acoustic interactions between blades and vanes in the outermost two stages of the compressor and turbine from generating propagating modes to decaying modes. This is usually effected in aircraft turbine components by increasing the number of stationary vanes relative to the number of rotor blades. In the GT-HTGR, because of the comparatively low blade Mach numbers, it may be feasible to effect this change by actually decreasing the vane number. A preliminary examination suggests that this proven noise reduction concept may be particularly attractive in the GT-HTGR application.

   b. Increase spacing between blades and vanes in the outermost two stages of the GT-HTGR components. Whether this step is required depends on certain details of step (a) above. If vane numbers cannot be changed sufficiently to produce cutoff of twice blade passage frequency, increasing the spacing can be used to reduce sound pressure levels by at least 10 log (spacing ratio).

c. Examine inlet struts and plenum chambers to determine their effects on inflow non-uniformity and unsteadiness. The benefits of such noise reduction measures as cutoff stator design (item a) have been demonstrated (Ref. 18-2) to be worth at least 20 dB over cut-on designs, but only when inflow air is smooth, as in aircraft flight conditions. When inflow is irregular and/or unsteady, as in aircraft ground operations or in test cells, the interaction of the rotor with this type of flow generates significant noise, thus short-circuiting the provisions of cutoff stator or spaced stator designs.

   It is therefore fruitless to change component stage designs without ensuring that the inflow is satisfactorily uniform and steady. Great success has been achieved in the aircraft industry through the use of "turbulence control structures" for use around the power plant inlet during static noise tests. These results suggest that a basically similar approach may be profitable with the GT-HTGR components.

d. The discharge regions of the GT-HTGR compressor and turbine should be examined in an analogous manner. Relatively little experience in this area is currently available.

e. Finally, the use of sound-absorbing lining or structures to dissipate acoustic power near the sources should be examined.

18.3.5. Graphite Dust Effects on Turbomachine

The effects of graphite dust included in the helium flow stream on the turbomachine were reviewed. It was indicated that about 177 g (100 lb) per year of graphite dust will be generated in the reactor core and presumably carried into the turbine. GA estimates the concentration of this dust at about 0.3 mg/ft^3. This dust is in the form of a very fine powder composed of ellipsoidal grains with a mean diameter of about 0.3 microns and a mean length of about 0.7 microns. Trajectory approximation calculations were made to determine the character of the flow of this particulate material through the helium turbomachine. Conclusions from this study are summarized below:

1. Approximations of the trajectories of the particles as they pass through the helium turbine were made using a theoretical approach developed by UTS (Refs. 18-3, 18-4). This analysis indicates that the particles will precisely follow the streamlines through the machine. The particles are very small and the density of graphite is quite low compared, for example, with the size and density of sand particles sometimes ingested into aviation gas turbines operating in dusty conditions. In the case of the graphite particles in the turbine, the transverse acceleration forces acting on the particles in the curved flow field are very small compared with the aerodynamic drag associated with particle motion perpendicular to the streamlines. Also, the transverse displacements of the particles are very small during transit time past a blade row. As a result, the particles follow the streamlines quite precisely and no impaction occurs on the air foils except at the stagnation point. Therefore, little or no erosion is expected to occur in the turbine. In the case of the compressor, a much lower concentration of particulate material is expected in the gas stream than in the case of the turbine, since the gas leaving the turbine passes through the recuperator and the precooler before entering the compressor. These heat

exchangers should function as fairly efficient separators to remove a large fraction of the particulate material carried by the gas. In any event, the situation in the compressor is similar to that in the turbine with regard to the particle trajectories. Here again, the very fine particles are expected to follow the streamlines with little erosive effect on the air foils.

2. The graphite dust is not expected to be very erosive since graphite itself is very soft and non-abrasive, as demonstrated by its wide use as a lubricant. Graphite has a hardness of 0.5 to 2 on the Moh scale compared with a hardness of 7 for silica and 4.5 to 6.5 for the glass-like substances, such as those found in fly ash. However, further information on the erosive characteristics of graphite particles is needed to confirm their benign character, since graphite is a highly anisotropic substance and may have impact characteristics which are quite different from its characteristics on sliding surfaces.

3. The graphite dust is not expected to build up thick layers on metallic surfaces in the dry, clean environment of the helium gas stream even at high temperatures, since graphite is a highly refractory material and the gas stream and metal temperatures are far below its melting point.

4. Attention must be paid to the potential for mechanical accumulation of the graphite dust in critical locations such as cooling passages. While the total volume of dust generated by the reactor core in a year is only on the order of 0.03 m^3 (1 ft^3) and is miniscule compared with the total volume of the system, experience with gas turbines has shown that some parts of these machines can function as very efficient centrifugal separators and can collect dust in critical locations. Care must be taken in the design of the turbomachine to avoid dust collection in this type of location.

18.3.6. Increased Turbine Inlet Temperature

The present GT-HTGR turbomachine operates with a turbine inlet temperature of 850°C (1562°F). Turbine blades or vanes are not cooled, although some cooling flow is supplied to the turbine disc rims and outer case. System advantages may be gained by allowing the turbine temperature to increase.

The trade-off between turbine inlet temperature and operating life was reviewed with reduced system cost associated with the higher temperature as the incentive. The specific exchange addressed was a 100°C (212°F) increase in temperature to the non-cooled turbine with a corresponding reduction from 280,000 to 100,000 hr of operating life. Results showed this trade to be impractical. The solution may be the addition of cooling helium, providing a significant impact from a simple design. The flow requirement would be extremely low with negligible impact on efficiency.

Reduction of turbine airfoil metal temperature by cooling the airfoils with helium taken from the compressor is an extremely effective approach to increasing the power output of the GT-HTGR turbomachine, either by permitting increased gas flow or increased turbine inlet temperature, or both. The allowable stress for a given total creep over a specified time interval is extremely sensitive to metal temperature. The allowable blade and vane stresses at a given metal temperature determine the maximum turbomachine size and power output. At a fixed machine size, the maximum turbine inlet temperature is set by the allowable blade and vane stresses at the corresponding airfoil metal temperatures.

The effectiveness of airfoil cooling in the GT-HTGR helium turbomachine is greatly enhanced by the excellent heat transfer properties of helium and by the relatively low temperature of the cooling gas available from the compressor. These characteristics permit a very high cooling effectiveness to be obtained with very simple airfoil coolant passage geometry. This results in substantial metal temperature reduction with very small coolant flows and, consequently, a very small efficiency penalty associated with turbine airfoil cooling.

The possibility of increasing the allowable turbine inlet temperature of the uncooled turbomachine by relaxing the creep life requirement for the turbine airfoils has been suggested. This is a very ineffective approach because creep rate at constant stress is extremely sensitive to temperature, and therefore even a very large reduction in required creep life will result in only a very small increase in allowable turbine inlet temperature. This is illustrated in Example 2 below.

The following examples are offered to quantify the foregoing statements:

1. To increase the maximum allowable turbine inlet temperature from 850° to 950°C (1562° to 1742°F) at constant metal temperature with no change in the turbine flow path, the first five or six rows of airfoils would have to be cooled. The theoretical cooling flow required for the first row of vanes and the first row of blades to maintain the metal temperature at 850°C (1562°F) is about 0.07% of engine air flow for each row. This is probably below the practical lower limit for cooling flow, and in any case, the total cooling flow for the first five or six rows would be less than 1% of engine gas flow.

2. If the 1% creep life requirement is reduced from 280,000 to 100,000 hr for the uncooled turbomachine designed to operate at 850°C (1562°F) turbine inlet temperature, the maximum allowable turbine inlet temperature could be increased only about 17°C (63°F). Thus, this is not a practical approach to increasing turbine inlet temperature.

Although the calculations for the above examples are very rough, they serve to illustrate the effectiveness of turbine cooling in the GT-HTGR helium turbomachinery.

18.3.7. Alternate Helium Buffer Seal

A dynamic seal concept as an alternative to the redundant labyrinth seal was reviewed. The operating principle is to balance a centrifugally loaded oil bath against a helium-buffering system. Although this concept has been previously incorporated in other types of systems, its use in the GT-HTGR turbomachine is discouraged by the anticipated oil motion on the impeller tips and the associated power loss.

18.3.8. 400-MW(e) Turbomachine Remote Disassembly Techniques

A preliminary review of the techniques for remote turbomachine disassembly was made to identify design and handling equipment requirements. The proposed disassembly sequence is as follows:

1. Rotate the turbomachine into the vertical position.

   a. Attach the pivotal fixture to the compressor end.

   b. Attach the lifting fixture to the turbine case mounting pins.

   c. Lift the engine into the vertical position.

   d. Lock the pivotal fixture in this position.

2. Support the power plant at the compressor end flange of the split case.

3. Remove the split outer case.

   a. Attach the removal fixtures to both halves of the case.

   b. Remove the bolts from the circumferential flange at the turbine end.

c. Replace the turbine rotor locking fixture with a similar one about 1/3 cm (1/8 in.) shorter to permit raising the stator assembly.

d. Raise the stator assembly.

e. Remove the bolts from the longitudinal flange and compressor end circumferential flange.

f. Pull the case halves radially away from the engine.

4. Remove the compressor discharge ducts.

5. Provide the support fixture from the compressor stator flange to the rear of the compressor shaft.

6. Remove the split turbine inlet duct assembly.

7. Provide the support fixture from the turbine stator flange to the exposed turbine shaft.

8. This is a "go/no-go" decision point. The entrance and exit stages of both the compressor and turbine are now accessible for inspection, presenting the following options:

   a. If there were no pre-shutdown anomalies and no visible signs of hardware distress, the above disassembly steps may be reversed, and the turbomachine returned to the PCRV.

   b. If pre-shutdown performance was suspect, or if hardware distress is evident, turbomachine disassembly should continue.

9. Remove the bolts from the shaft flange. Lift the turbine assembly off the compressor and mount it in the vertical position on the outermost turbine case flange.

10. Unbolt the aft turbine case flange, and using the rotor/stator locking fixture, lift the turbine assembly out of the containment ring assembly. Mount the rotor/stator assembly in the vertical position on the end of the turbine shaft.

11. Stator assembly removal.

    a. Provide support to the outermost exhaust case flange.

    b. Attach the removal fixtures to both pieces of the split case assembly.

    c. Remove the shaft support fixtures as supplied in step 7.

    d. Unbolt the circumferential flange and lower stator assembly to clear the flange snap.

    e. Unbolt the longitudinal flange.

    f. Remove the stator halves radially away from the rotor.

    g. Place both pieces in the horizontal fixture to remove the vanes.

12. The turbine vanes are now accessible for visual inspection. If necessary, the vanes can be removed from the case. If vane removal is unnecessary, hold the stator assemblies for rebuilding of the engine.

13. Remove the turbine exhaust case.

    a. Assembly is in the vertical position supported on the turbine shaft flange and exhaust case major flange.

    b. Remove the rotor/stator locking fixture.

    c. Unbolt the bearing assembly from the bearing support ring, and remove the bearing.

    d. Unbolt the bearing support ring and primary seal rub-strips. Remove the support ring and primary seal as a unit. Further disassembly can be done on the bench.

    e. Unbolt the secondary seal rub-strips and lift the turbine exhaust case with the secondary seal system from the rotor. Further seal disassembly can be done on the bench.

14. Turbine blades are now accessible for visual inspection. If necessary, blades can now be removed from the drum for repair or replacement. If blade removal is unnecessary, hold the rotor assembly for engine rebuilding.

15. To perform compressor disassembly, follow a procedure similar to steps 10 through 14.

18.3.9. 400-MW(e) Split Case Design Concept

To assess its advantages, a full-length split case design was laid out around the present 400-MW(e) flow path as shown in Fig. 18-8. This configuration provides significant benefits with regard to remote handling, decontamination, and maintenance. The cases can be separated and the rotor removed as a single unit. Use of this design would require incorporation of split containment rings. These rings are included for safety purposes to contain particles in the unlikely event of a blade or rotor failure.

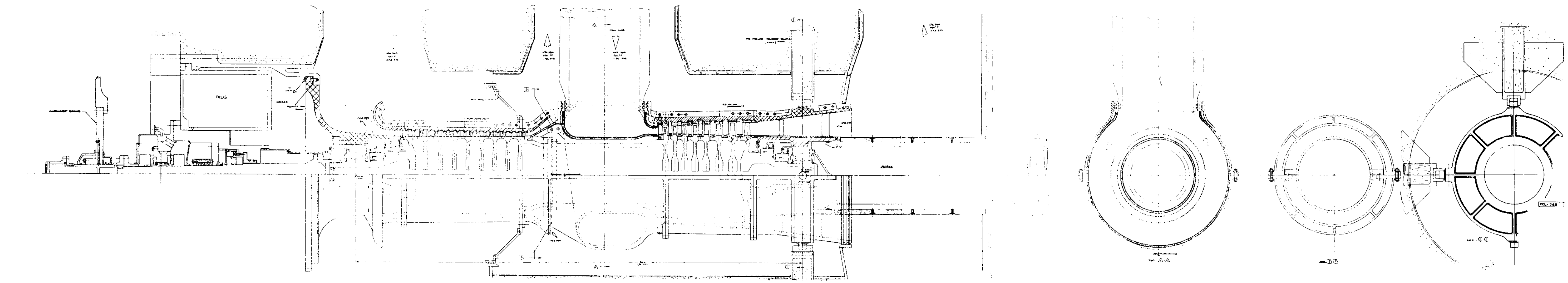


Fig. 18-8. 400-MW(e) split case design concept

It was necessary to review the containment problem to see if split containment rings with the necessary longitudinal mechanical joint are practical. Westinghouse Research Laboratories (Ref. 18-5) has developed an energy absorption theory of disc fragment containment and correlated the containment prediction system with the results of large-scale disc burst tests in which the disc fragments impinged on steel containment rings. In that analysis, disc failure is treated as a two-stage process. In stage 1 the disc fragments impact the containment ring, transferring momentum and kinetic energy to the ring. The energy associated with the momentum transferred to the ring on impact must be absorbed by plastic shear and compression strain in the area of impact or the fragments will perforate the ring and escape. If the fragments do not punch through the ring, the remainder of the kinetic energy of the fragments must be absorbed by plastic strain associated with stretching deformation of the ring in stage 2 of the process. The containment ring must be capable of absorbing this kinetic energy in plastic strain or it will break and release the disc fragments as missiles.

The Westinghouse procedure was used to analyze the mechanics of containment of the eighth-stage turbine disc in the 400-MW(e) helium turbomachine at the minimum burst speed condition, which for this machine is 150% of the 3600-rpm operating speed. As a basis for the containment calculation, it was assumed that the eighth-stage (heaviest) disc burst at 150% overspeed into four approximately equal fragments. This assumption leads to about the maximum fragment translational kinetic energy which can be attained.

The results from this study led to the following conclusions:

1. A containment ring of approximately 17.8 cm (7 in.) thickness should be adequate to contain an eighth-stage disc failure if the ring is extended axially downstream by 38 to 51 (15 to 20 in.) beyond the area of impact.

2. Varying the thickness of the containment ring along its length would probably save cost and weight since the disc energy increases stage by stage through the turbine. Detailed calculations taking into account the energy of fragments from each of the turbine discs and the trade between ring thickness and extension of the ring at the ends of the turbine would be required for optimum design of the containment ring.

3. The nature of the energy absorption process, which depends upon large plastic strain in the containment ring, almost certainly precludes the use of any sort of axial mechanical joints in the ring.

Based on the results of this study, the split case configuration has been removed from consideration.

18.3.10. 500-MW(e) Conceptual Design

To accommodate studies of various plant sizes, a 500-MW(e) conceptual design layout was prepared. This configuration is shown in Fig. 18-9. Additional spacing has been incorporated between the compressor and turbine inlet and exit stator and rotors to provide for sound power level attenuation. Table 18-7 presents associated performance characteristics.

18.3.11. Generator Size

Initial system layouts were configured with hydrogen-cooled generators located outside the secondary containment building (SCB). This precluded the possibility of hydrogen leakage into the SCB. However, it resulted in the requirement to penetrate the SCB with the shaft connecting the turbomachine and generator. To eliminate this penetration, water-cooled generators have been investigated, which can be located inside the SCB. Brown, Boveri and Cie (BBC) has provided information on water-cooled generator configurations and physical sizes.

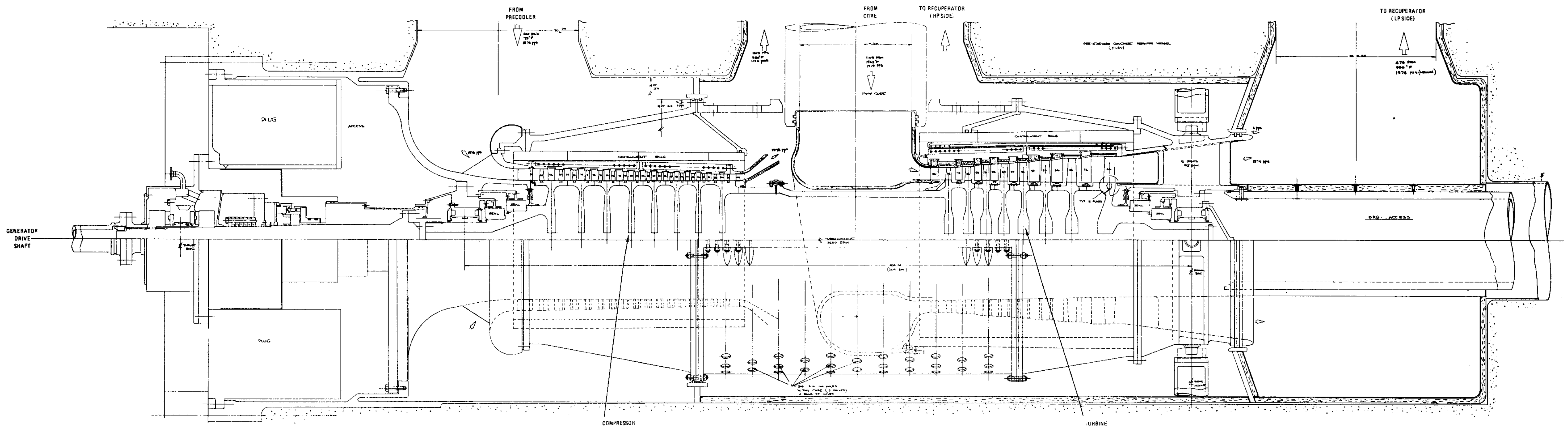


Fig. 18-9. 500-MW(e) conceptual design

TABLE 18-7
PERFORMANCE CHARACTERISTICS OF 500-MW(e) TURBOMACHINE

	Design
Total flow (compressor inlet)	1576 lbm/sec
Actual rotor speed	3600 rpm
Overall system pressure loss	7.10%
Compressor	
Inlet corr. flow	51.19 WAT $\sqrt{\theta T_2}/\delta T_2$
Number of stages	16
Pressure ratio	2.5
Efficiency	89.8%
Inlet temperature	26.7°C (80°F)
Reactor core	
Heat generated in core	1212.55 MW
Plant heat loss	6.25 MW
Heat supplied to cycle	1206.3 MW
Turbine	
Efficiency	91.8%
Expansion ratio	2.322
Inlet temperature	850°C (1562°F)
Number of stages	8
Helium cooling flow	3.6%
Recuperator	
Effectiveness	89.8%
Precooler	
Helium side temperature change	181°C (357.3°F)
Rotor loss	2.32 MW (3110 hp)
Delivered shaft power	507.72 MW (0.6809 x 10^6 hp)
Generator efficiency	98.7%
Gross electric power generated	501.12 MW
Plant auxiliary power requirement	1.25 MW
Net electric power generated	499.87 MW
Net power plant thermal efficiency	41.22%

REFERENCES

18-1. Sofrin, T. G., and N. Rilcoff, Jr., "Two-Stage, Low Noise, Advanced Technology Fan, V. Acoustic Final Report," National Aeronautics and Space Administration Report NASA CR 234831, September 1975.

18-2. Sofrin, T. G., and C. C. Mathews, "Asymmetric Stator Interaction Noise," Paper No. 79-0638, AIAA 5th Aeroacoustics Conference, Seattle, March 12-14, 1979.

18-3. Dring, R. P., and M. Suo, "Particle Trajectories in Swirling Flow," United Technologies Research Center Report UTRC 76-167, November 30, 1976.

18-4. Dring, R. P., and J. R. Cucpar, "Particle Trajectories in Turbine Cascades," United Technologies Research Center Report UTRC 77-29, March 1977.

18-5. Hagg, A. C., and G. O. Sankey, "The Containment of Disk Burst Fragments by Cylindrical Shells," Trans. ASME 96, 114-123 (April 1974).

19. CONTROL VALVE (632003)

19.1. SCOPE

The purpose of this task is to determine the trim, attemperation, main bypass, and safety trip functional requirements and prepare conceptual design layouts.

19.2. SUMMARY

Conceptual designs for the GT-HTGR bypass valves have been developed, and this work is reported in Refs. 19-1 and 19-2. Figures 19-1 through 19-4 show the four basic valve configurations: control, safety, trim, and attemperation. These valves are normally closed when the plant is at 100% power. They are used in the event of a sudden or slow change in plant load to control the turbine speed. A summary of the valve requirements is given in Table 19-1.

19.3. DISCUSSION

Based on the work to date, the following areas of concern have been identified:

1. <u>High Stem Forces</u>. The existing design of the safety and control valves will require a stem force of over 22,679 kg (500,000 lb) just to overcome the fluid forces. This much force will require a very special, high-powered hydraulic actuator. In addition, the valves have a very fast stroke time, which will mean very large amounts of energy will be required to both move the valve and stop it. This problem is not as severe with trim and attemperation valves.

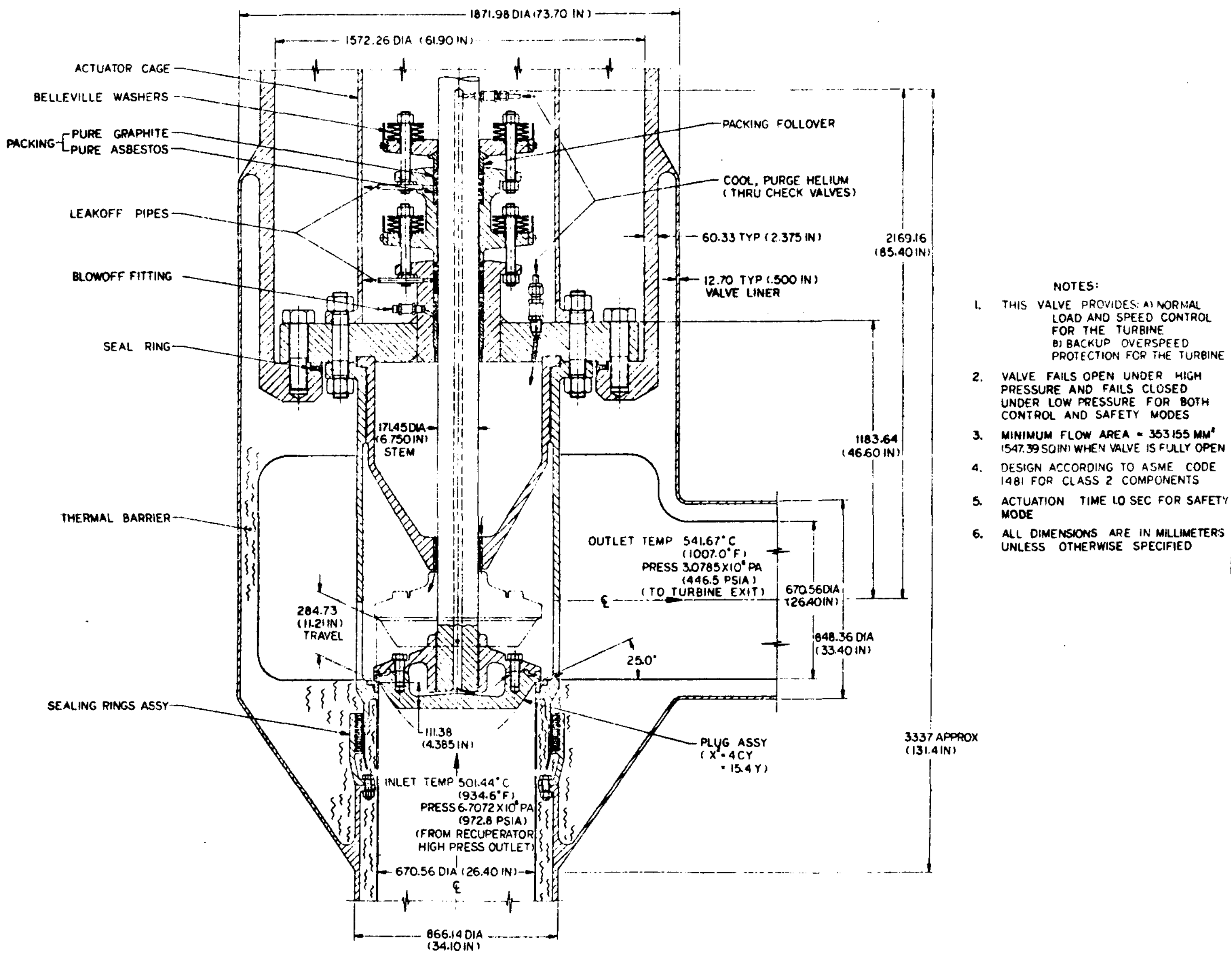


Fig. 19-1. Primary bypass valve

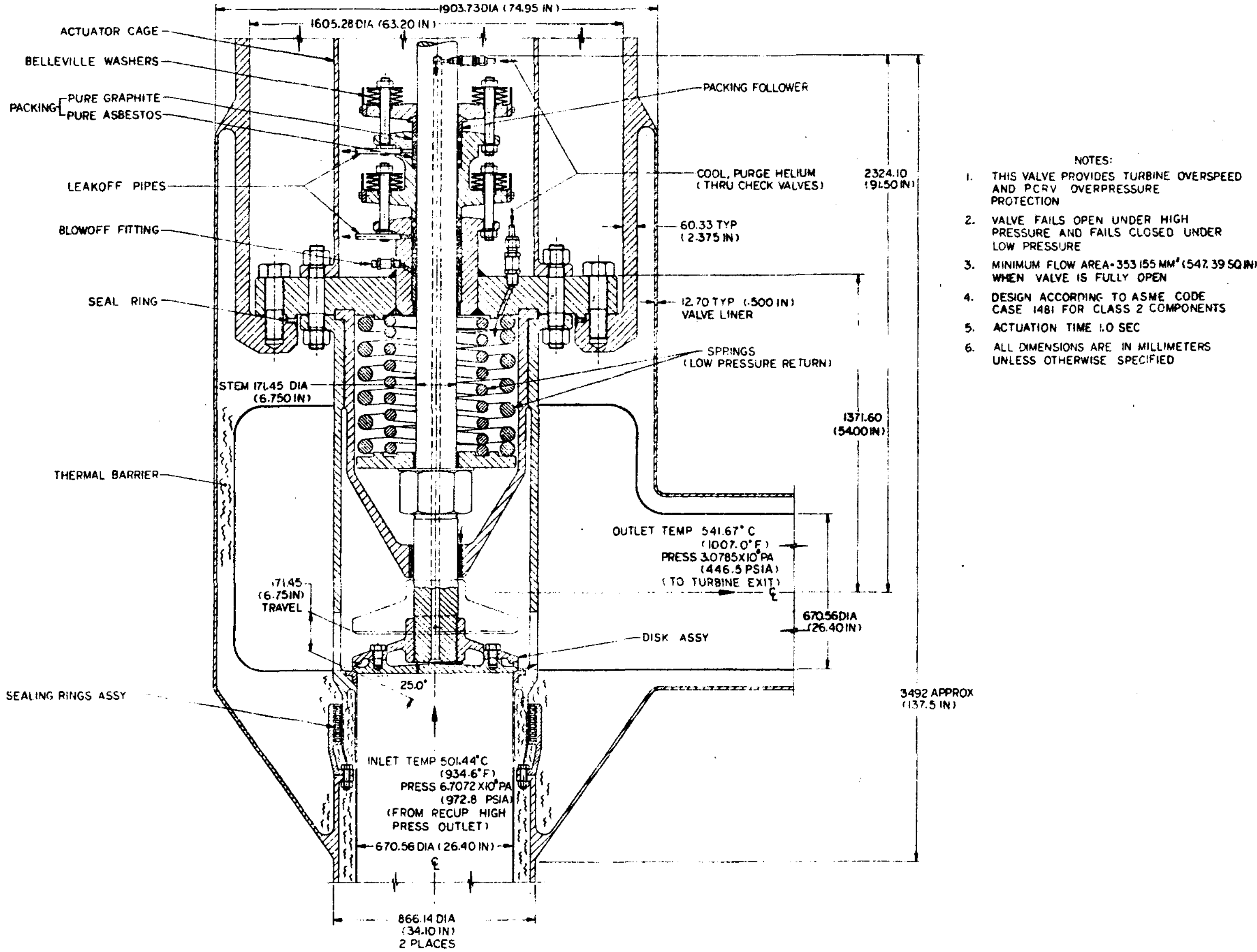


Fig. 9-2. Safety valve

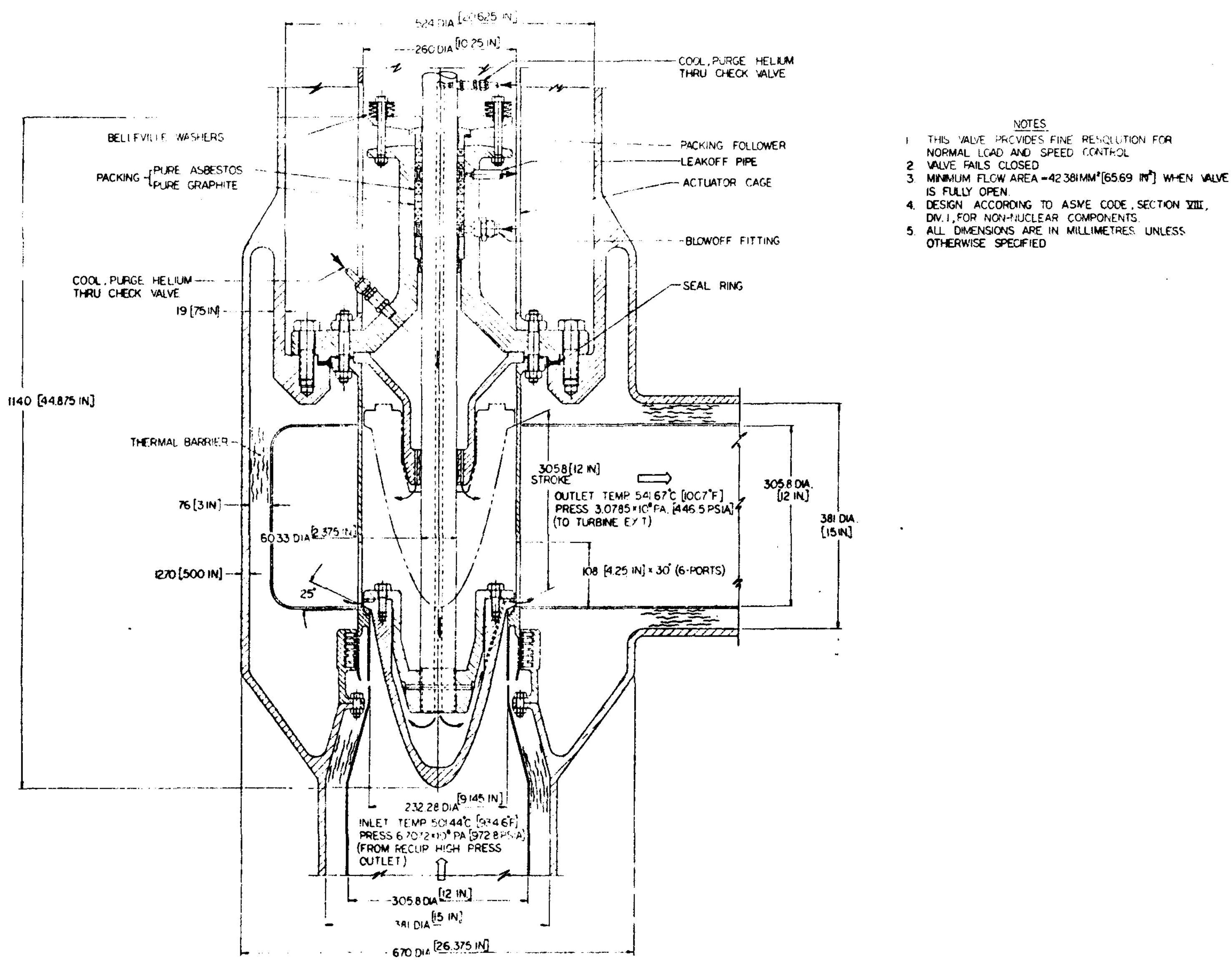


Fig. 9-3. Trim valve

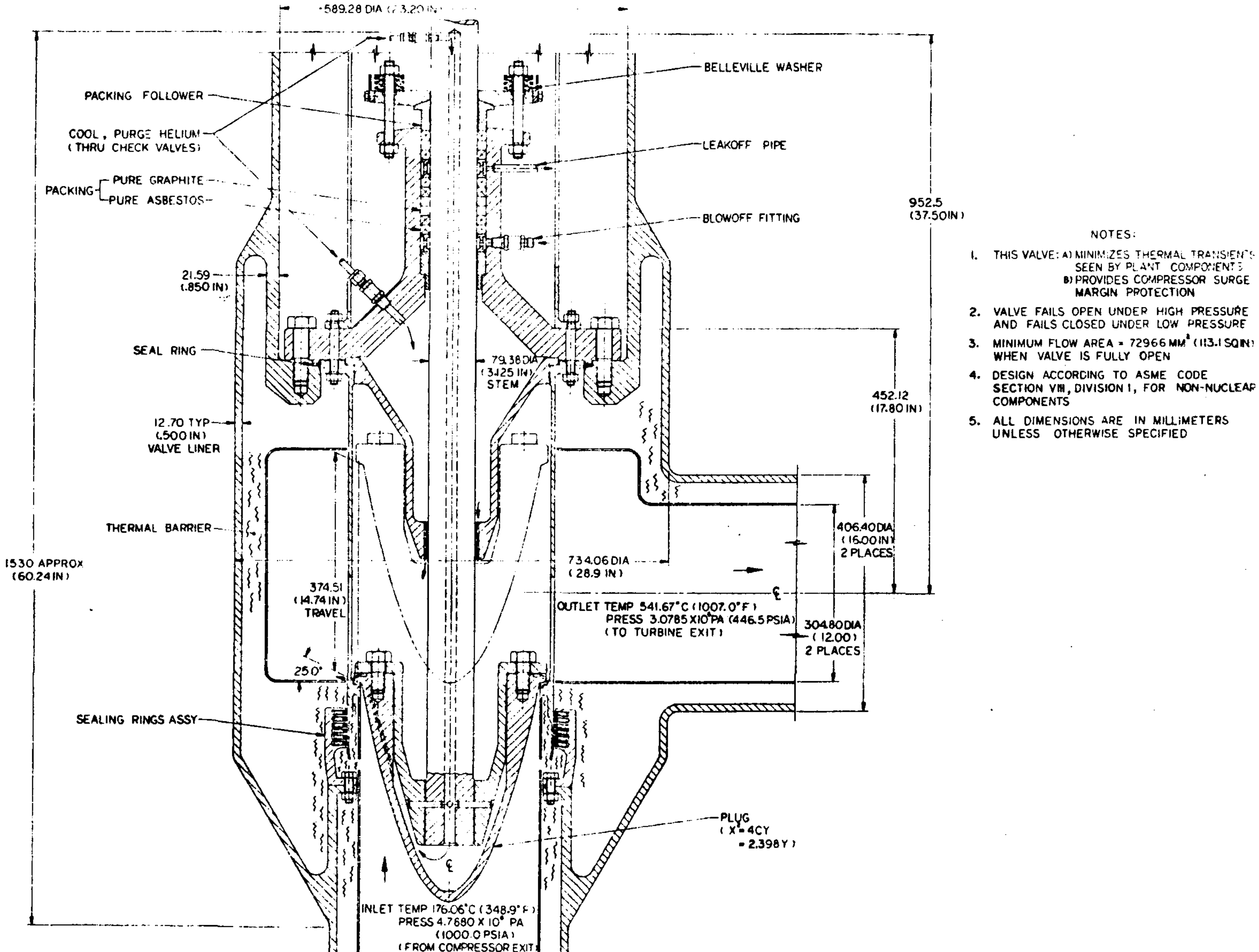


Fig. 19-4. Attemperation valve

TABLE 19-1
THREE-LOOP, 400-MW(e) TURBOMACHINE - NON-INTERCOOLED, REFERENCE PLANT

	Control Valve	Trim Valve	Safety Valve	Attemperation Valve
Design pressure [MPa (psig)]	8.17 (1185)	8.17 (1185)	8.17 (1185)	8.17 (1185)
Design temp.	Per analysis	Per analysis	Per analysis	Per analysis
Normal inlet pressure [MPa (psia)]	7.93 (1150)	7.93 (1150)	7.93 (1150)	7.93 (1150)
Normal temp. [°C (°F)]	498 (928)	498 (928)	498 (928)	174 (346)
Flow area, 100% open [mm^2 ($in.^2$)]	493×10^3 (765)	41×10^3 (65)	493×10^3 (765)	73×10^3 (113)
0%-100% travel time open (s)	1	1	1	1
Control range (%)	0-100	0-100	Not req'd	0-100
Position resolution	0.2% of stroke	0.2% of stroke	N/A	0.2% of stroke
Loss of power position	Open: $\Delta P > 20$ Closed: $\Delta P < 20$	Open: $\Delta P > 20$ Closed: $\Delta P < 20$	Open: $\Delta P > 20$ Closed: $\Delta P < 20$	Open: $\Delta P > 20$ Closed: $\Delta P < 20$
Maximum press. drop [MPa (psi)]	4.65 (675)	4.65 (675)	4.65 (675)	4.65 (675)
Type of control contour	Linear	Equal percent.	Quick open	Linear
Normal ΔP [MPa (psi)]	4.65 (675)	4.65 (675)	4.65 (675)	4.65 (675)
Seat leakage normal ΔP [kg/s (lb/hr)]	0.19 (1500)	0.19 (1500)	0.19 (1500)	0.19 (1500)
10% step change, time constant (s)	0.25	0.25	N/A	0.4

2. Friction and Self-Welding. The valves remain in the closed position when the plant is base-loaded during normal operation. Also, leakage has a very negative effect on plant efficiency. Thus, the designs will tend to have high seat forces to give good shutoff. The combination of high temperature, high surface forces, and long hold time in an inert atmosphere could easily lead to self-welding. Careful material selection and possibly some testing will be required.

3. Seat Erosion. The normal differential pressure across the valve seat is sufficient to cause sonic flow and attendant high velocities, i.e., 1524 m/s (5000 ft/sec). This high velocity combined with entrained particulate material could be very erosive.

4. Stem Leakage. The actuator must be housed within the primary pressure boundary, or a bellows sealed stem must be incorporated.

5. Removal and Maintenance. Attention needs to be directed to the removal and maintenance requirements. Special provisions must be made so that the valves may be installed and removed with remote handling equipment. In addition, those parts which may become contaminated need to be designed to allow for disassembly with remote maintenance tools.

6. Flow Noise. The high pressure drop will induce noise which may lead to vibration problems with the thermal barrier or other adjacent equipment.

The very high stem forces required for the control and safety valves could be reduced by adopting a balanced or semibalanced valve disc design. Examples of such designs would be a double-seated valve, a butterfly valve, or a valve with an auxiliary balancing chamber.

A review of the Fort St. Vrain main steam turbine stop and control valves was made to provide a point of reference. The stop valve must close in less than 1 s. The valve utilizes a combination of spring force and fluid force to provide quick closing; the actuator has very limited power to open the valve. In contrast, the control valve uses a balance chamber to minimize the stem forces. The seat diameter is 20.3 cm (8 in.), and the balance chamber diameter is about 17.8 cm (7 in.). This will balance out most of the fluid forces and allow the use of a modest size actuator.

Future valve investigation will be directed toward designs which will lower the required stem forces.

REFERENCES

19-1. "Gas Turbine HTGR Program, Semiannual Progress Report for the Period July 1, 1975 through December 31, 1975," ERDA Report GA-A13740, General Atomic Company, January 29, 1976.

19-2. "Nuclear Gas Turbine Power Plant Preliminary Development Plan," ERDA Report GA-A12709, General Atomic Company, October 1976.

20. HEAT EXCHANGERS (6321)

The heat exchanger design effort is divided between GA and its subcontractor Combustion Engineering (CE).

20.1. CE EFFORT

20.1.1. CE Scope

The scope of CE's part of the heat exchanger design effort is as follows:

1. Complete the conceptual mechanical designs, estimated costs, and schedules for the power conversion loop (PCL) heat exchangers for three different GT-HTGR plant configurations:

    a. 600-MW(e), one-loop intercooled plant.
    b. 1200-MW(e), two-loop intercooled plant.
    c. 1200-MW(e), three-loop non-intercooled plant.

    The mechanical designs and estimates were to be consistent with performance and interface requirements specified and provided by GA. Information provided to CE by GA prior to the start of the effort included:

    a. Preliminary designs.
    b. Thermal-hydraulic data.
    c. Operating requirements and envelope.
    d. GA concept selection report.
    e. Statement of technical approach.

All work was to be documented in a final summary report highlighting:

a. Conceptual mechanical design (drawings).
b. In-service inspection (ISI) concepts.
c. Maintenance and repair concepts.
d. Installation and removal concepts.

2. Review the HHT (Sulzer) heat exchanger drawings and criteria and, in conjunction with GA, establish the basis for the heat exchanger designs which will meet the requirements of U.S. and European GT-HTGR programs.

20.1.2. CE Summary

The GT-HTGR heat exchanger design work being performed by CE is aimed principally at establishing the manufacturability of the components. The workscope did not permit detailed structural/thermal analysis of the design, nor did it include any verification of thermal-hydraulic performance of the heat exchangers. Numerous items have been identified by CE as warranting further investigation to resolve some of the uncertainties that exist in the designs. GA's Heat Exchanger Department has reviewed these items and has identified certain design issues that must be resolved before complete justification of design feasibility can be established. For the remainder of FY-79, CE and the GA Heat Exchanger Department will address these problem areas, and it is anticipated that most can be resolved by relatively minor design modification.

20.1.3. CE Effort Discussion

Figures 20-1 through 20-8 illustrate the mechanical design developed by CE for the PCL heat exchangers for three different plant configurations:

600-MW(e) One-Loop Intercooled Plant

Figure 20-1	Recuperator
Figure 20-2	Precooler
Figure 20-3	Intercooler

1200-MW(e) Two-Loop Intercooled Plant

Figure 20-4	Recuperator
Figure 20-5	Precooler
Figure 20-6	Intercooler

1200-MW(e) Three-Loop Non-Intercooled Plant

Figure 20-7	Recuperator
Figure 20-8	Precooler

As stated earlier, the basic conceptual designs for the heat exchangers under consideration were provided to CE by GA as a starting point for the mechanical design and the manufacturing and shipping studies that followed. For the most part, the designs offered by GA were retained during the course of work and are judged by CE to be designs that can be developed to fulfill the requirements for GT-HTGR heat exchangers. During the course of the study, certain of the design features were modified by CE to make them more amenable to manufacture and to reduce costs wherever possible, consistent with the requirements specified by GA.

Figures 20-1, 20-2, and 20-3 illustrate the recuperator, precooler, and intercooler for the one-loop plant. From these figures it can be seen that the basic conceptual design of the heat exchangers is identical to its counterpart in the two-loop and three-loop plants. The minor variations

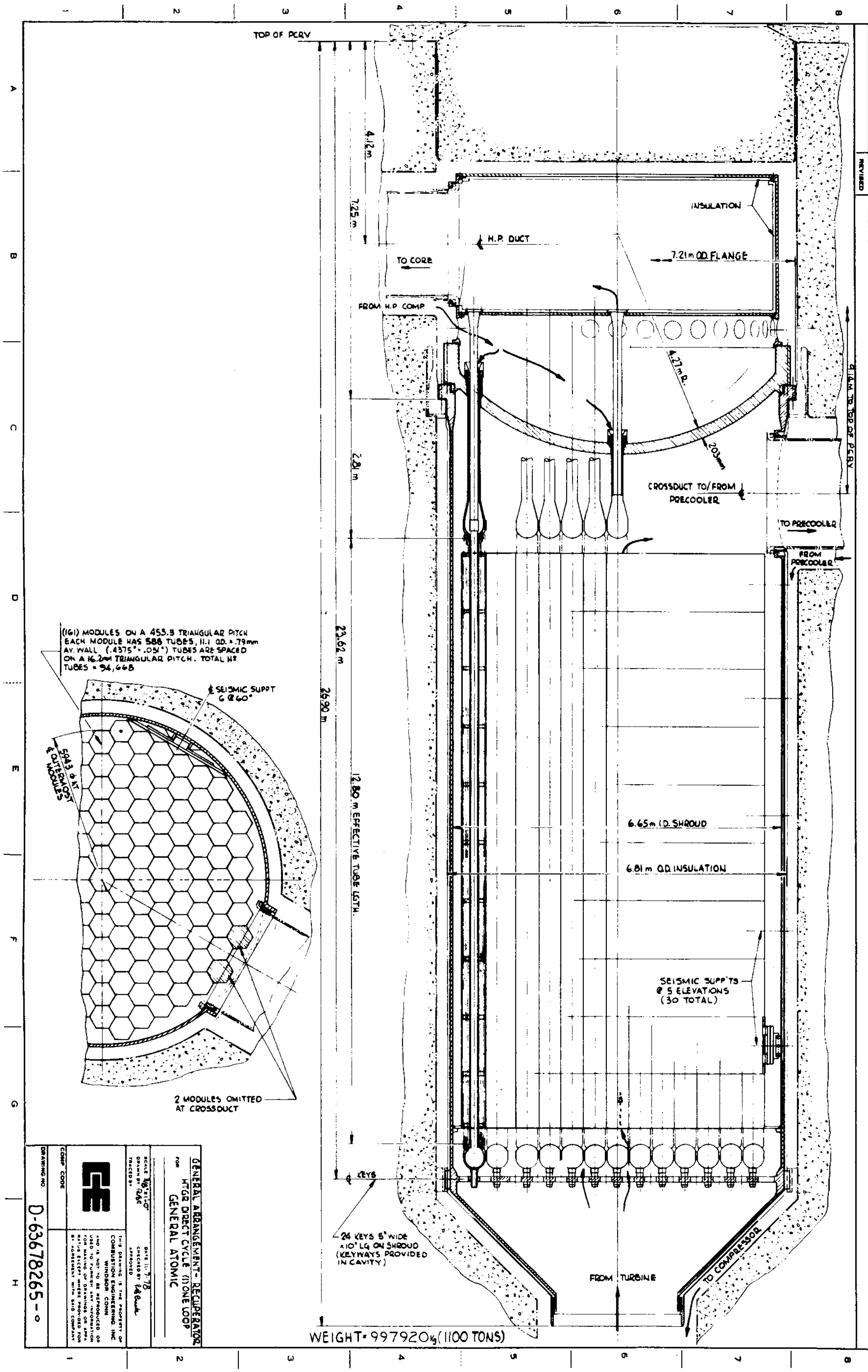


Fig. 20-1. Recuperator for 600-MW(e), one-loop intercooled plant

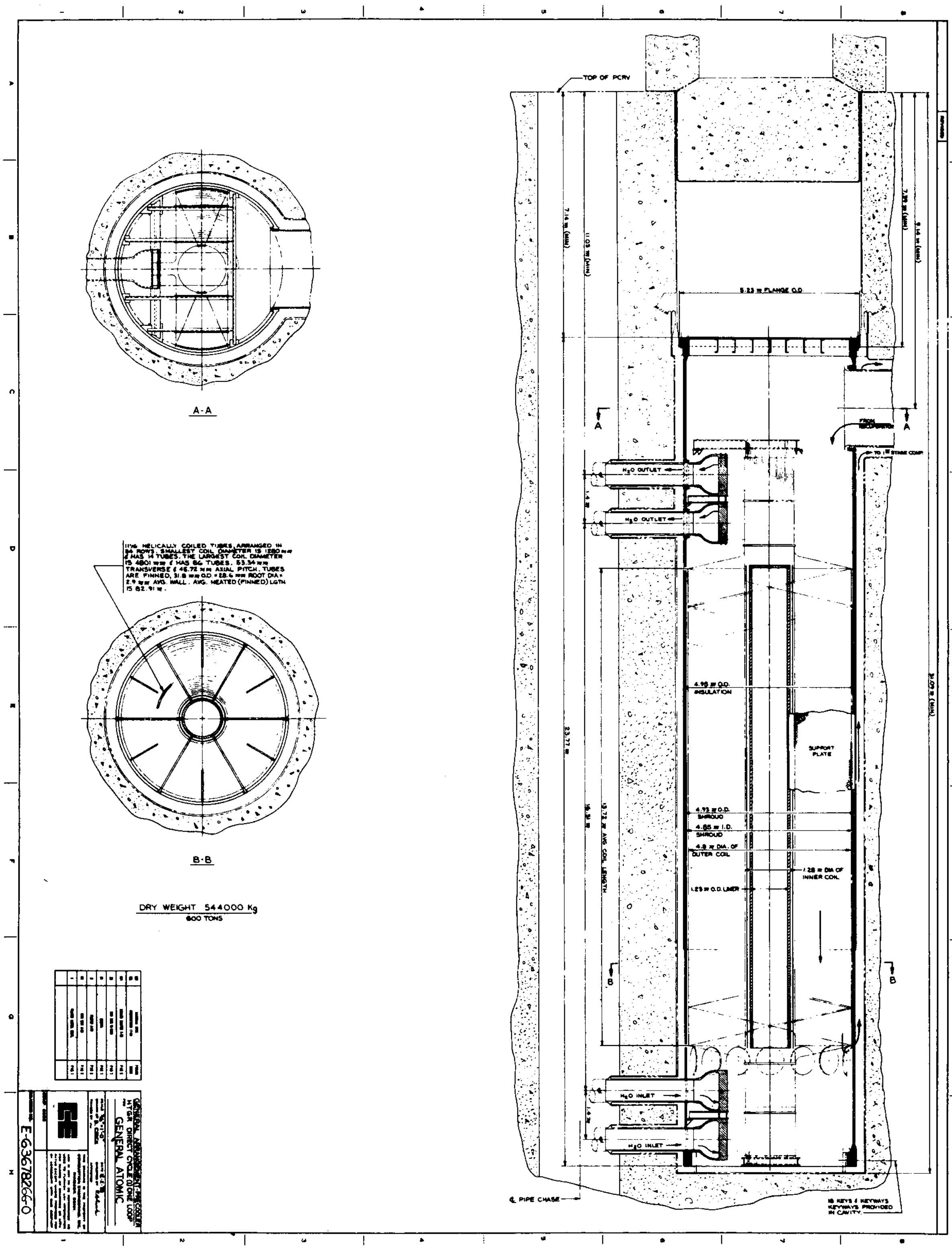


Fig. 20-2. Precooler for 600-Mw(e), one-loop intercooled plant

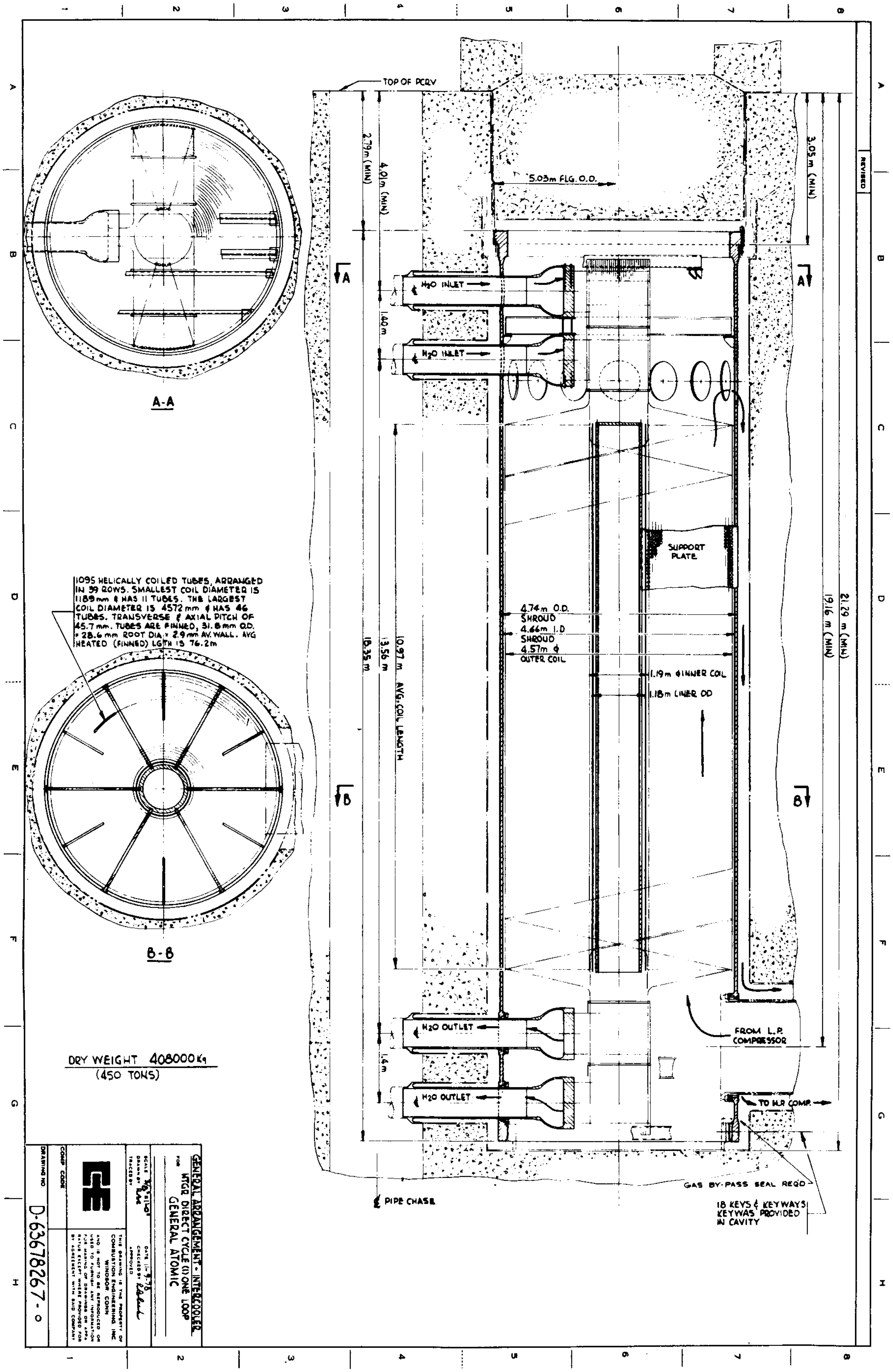


Fig. 20-3. Intercooler for 600-MW(e), one-loop intercooled plant

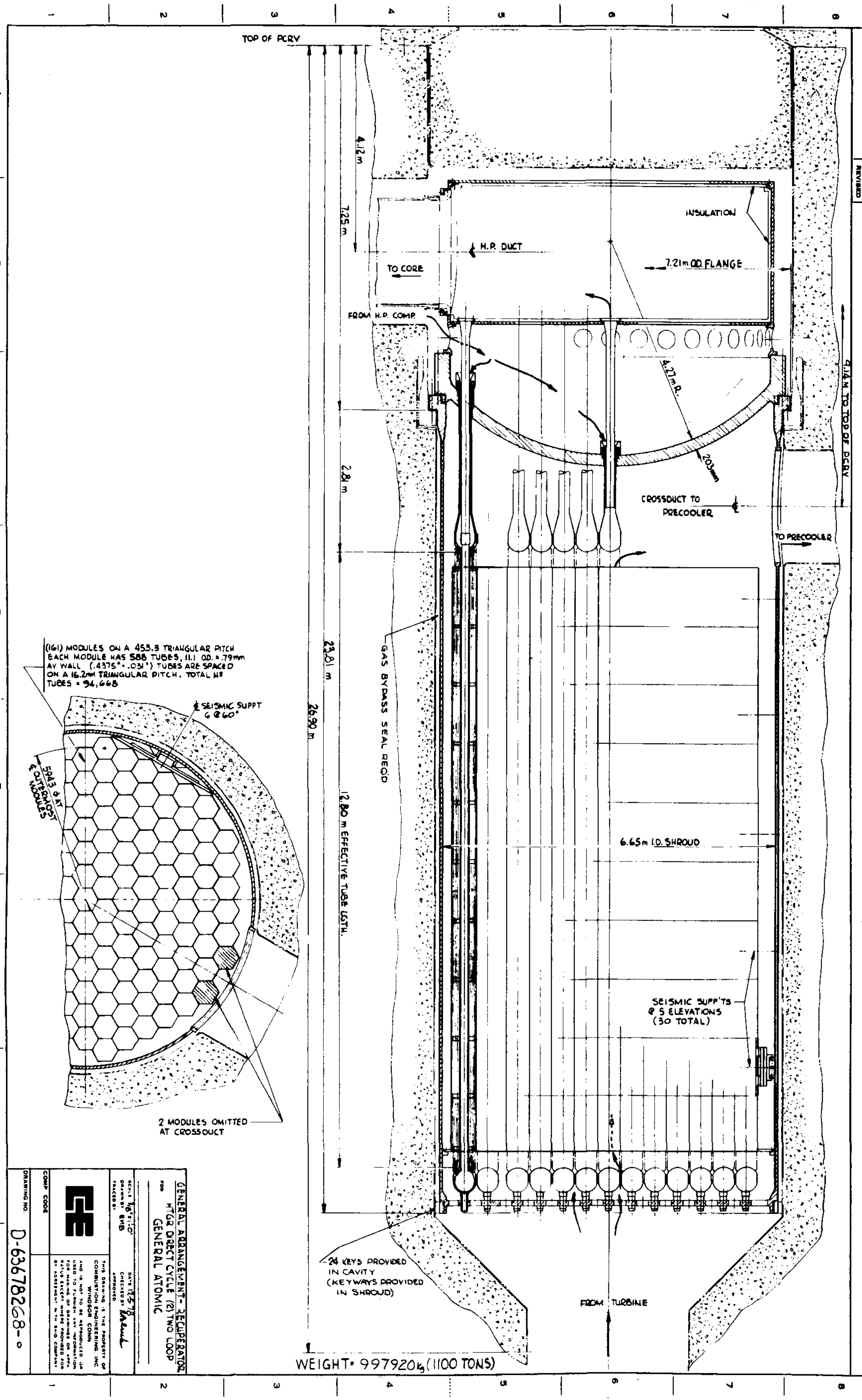


Fig. 20-4. Recuperator for 1200-MW(e), two-loop intercooled plant

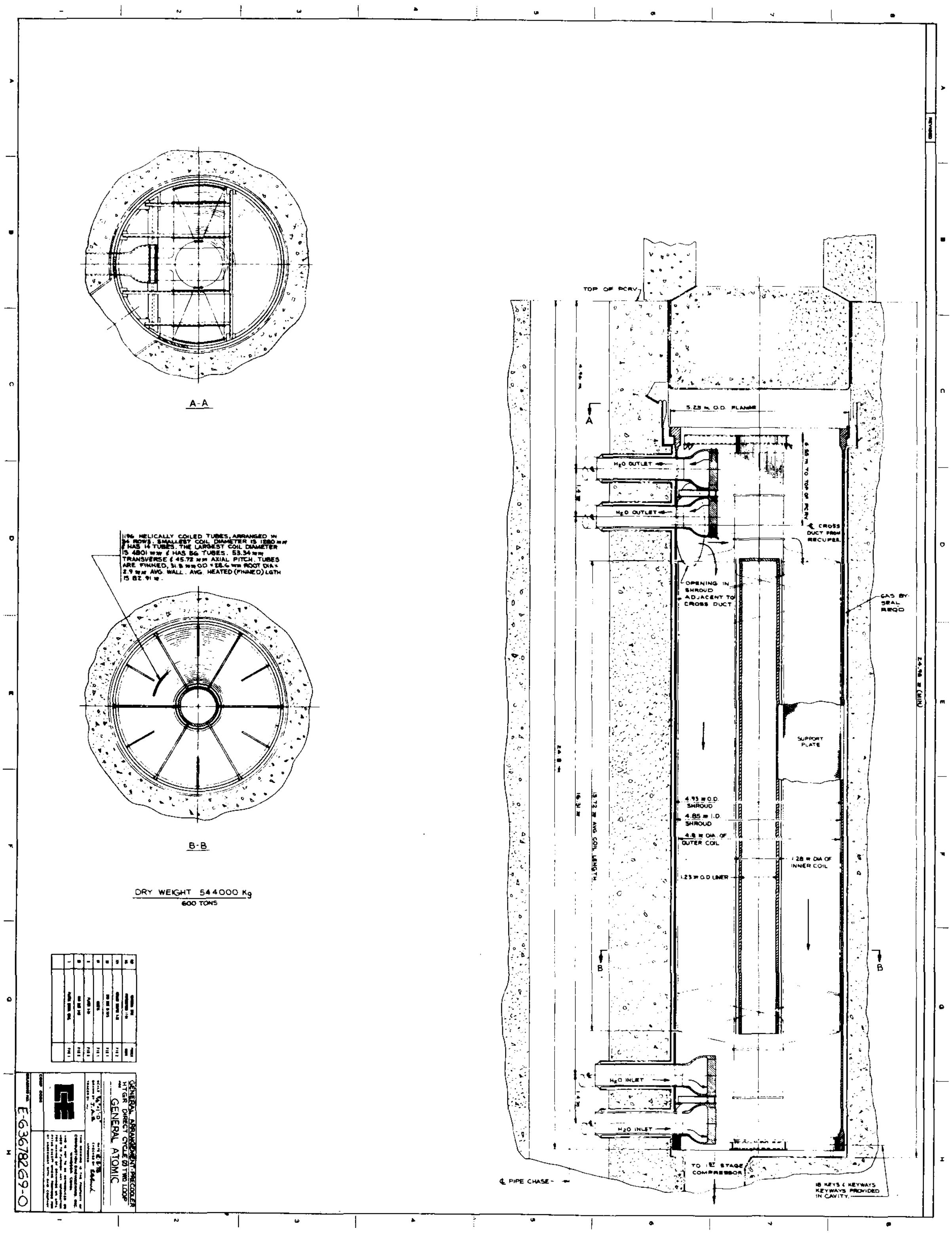


Fig. 20-5. Precooler for 1200-MW(e), two-loop intercooled plant

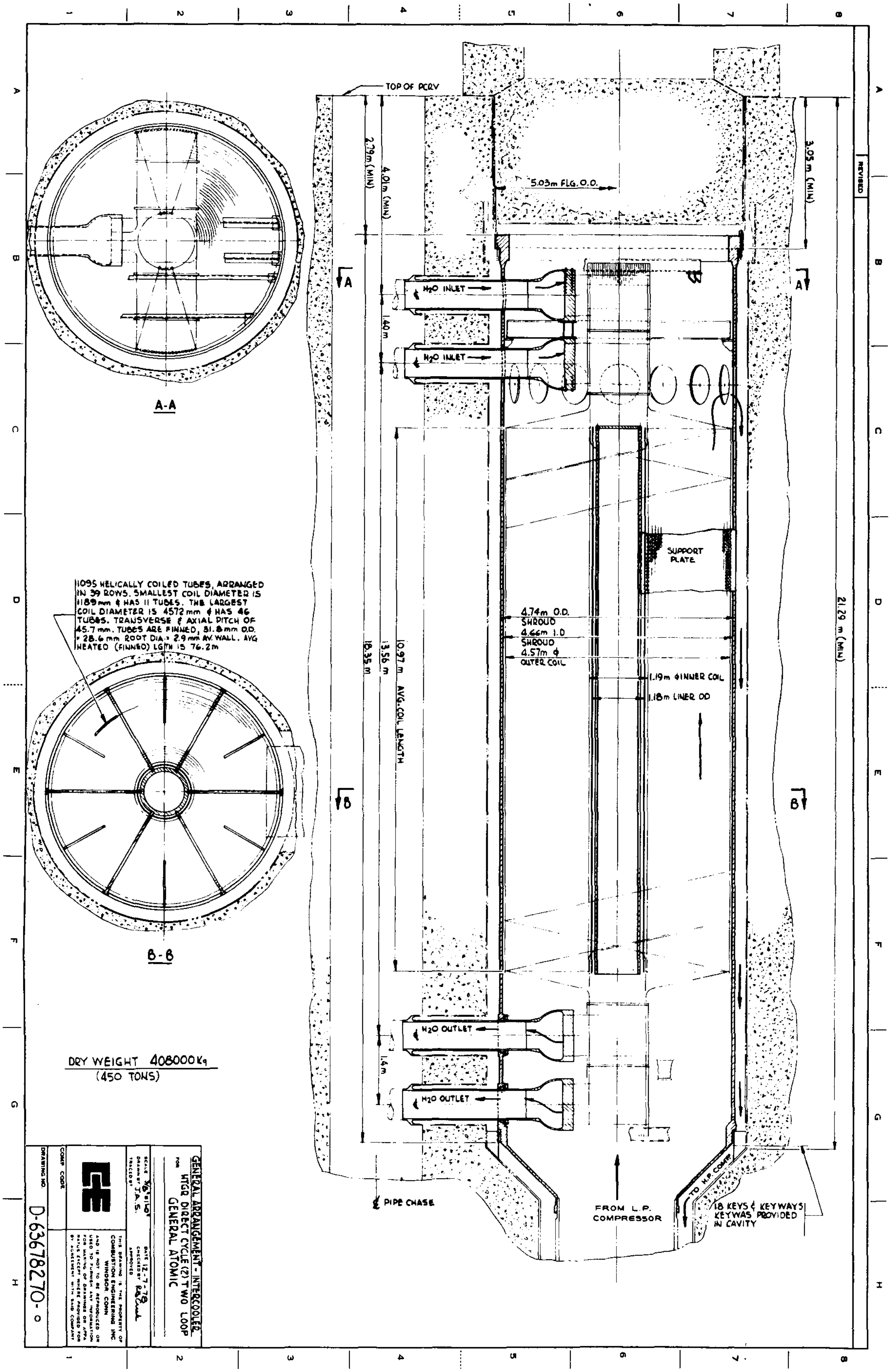


Fig. 20-6. Intercooler for 1200-MW(e), two-loop intercooled plant

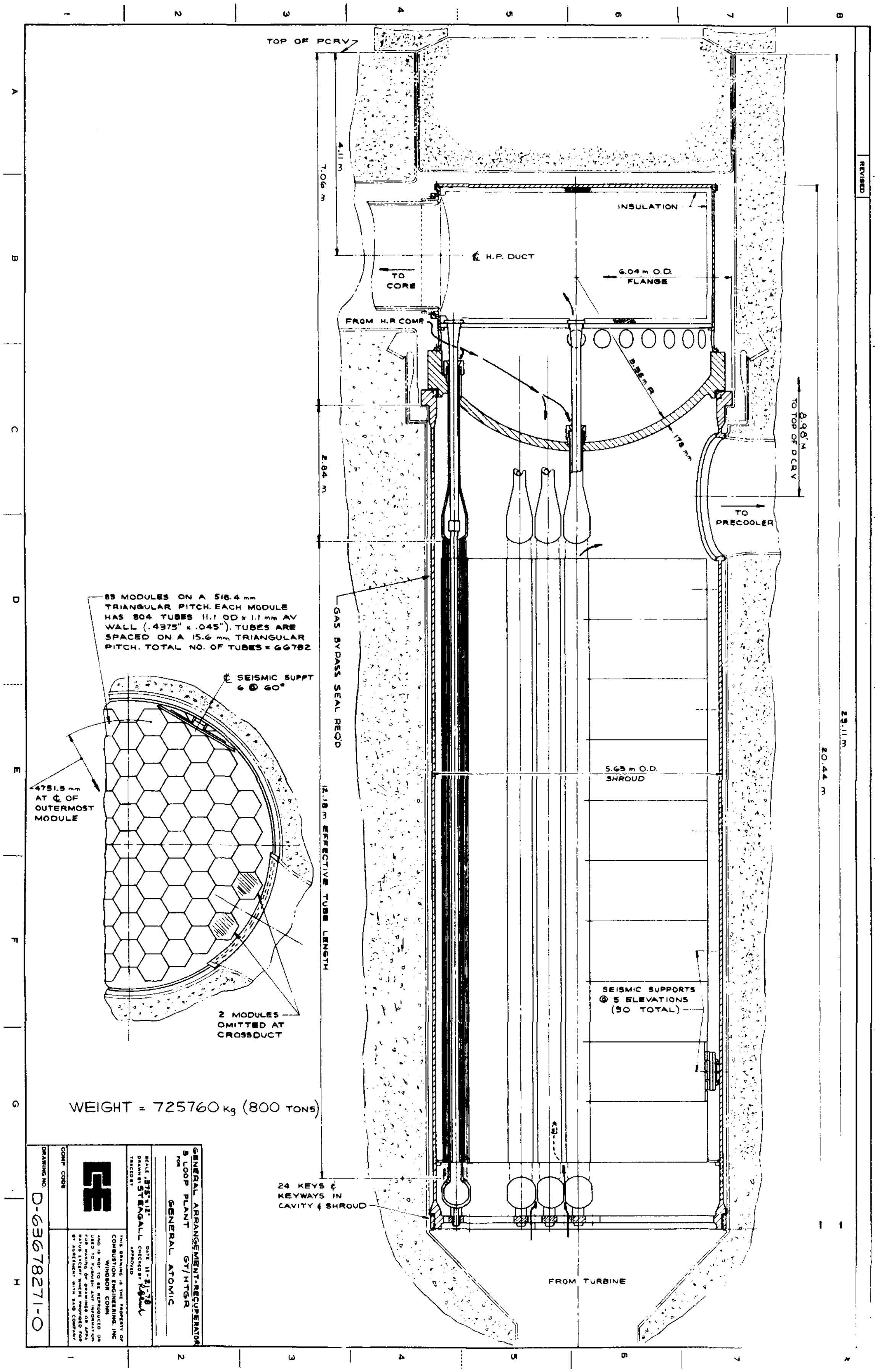


Fig. 20-7. Recuperator for 1200-MW(e), three-loop non-intercooled plant

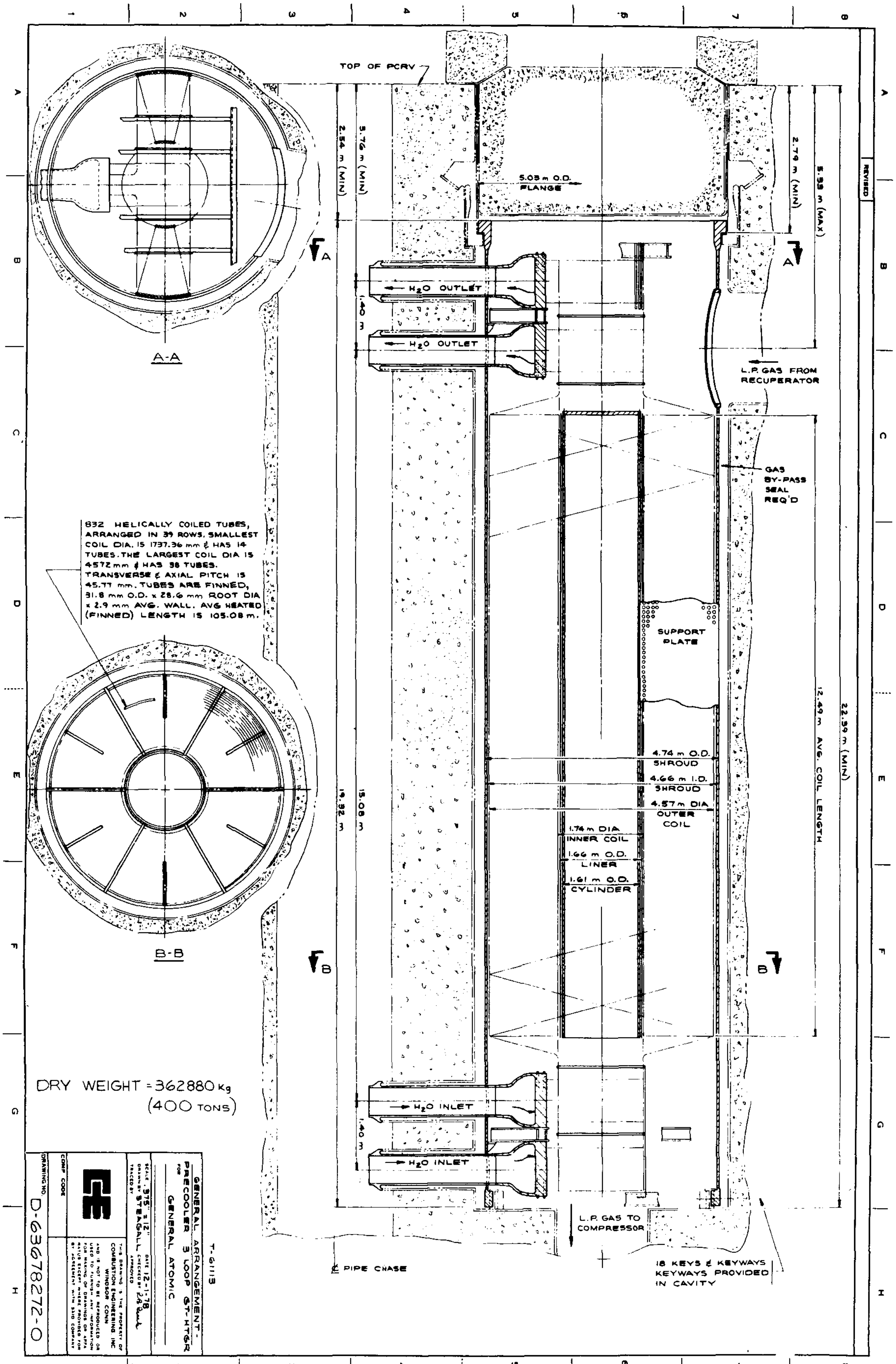


Fig. 20-8. Precooler for 1200-MW(e), three-loop non-intercooled plant

that exist in the one-loop and two-loop designs result from the differences in the type of PCRV liner cooling incorporated into the plant design. The one-loop plant design utilizes the "warm liner" concept wherein the cool gas from the precooler and intercooler is circulated within the heat exchanger cavities to limit the PCRV concrete temperature. This liner cooling concept is being investigated in Europe for use in the HHT demonstration plant.

The two- and three-loop plant designs utilize "conventional" PCRV cavity liner cooling whereby the PCRV concrete temperature is limited by a thermal barrier and cooling water that is circulated through cooling tubes on the outside surface of the liner. Irrespective of the other advantages and disadvantages associated with these liner cooling concepts, the "warm liner" imposes more difficulties in interfacing between the heat exchanger and its surrounding cavity.

The heat exchangers for the three-loop non-intercooled plant are of a different thermal size from their counterparts in the other plants, but the basic conceptual design is the same.

Tables 20-1 and 20-2 reflect some of the differences in the three plant configurations. Table 20-1 compares the recuperators for the three plants and Table 20-2 compares the precoolers and intercoolers. Note that in the absence of intercoolers in the three-loop plant, added thermal duty must be borne by the precoolers. Included in these tables are comparative cost factors for the heat exchangers in the different plant configurations. The three-loop recuperator design reflects a cost improvement achieved by increasing the size of the module. Experience has shown that, normally, total costs will decrease with a decrease in the number of modules. This leads to larger modules and, since the three-loop recuperator was designed later in the study, less time permitted for optimization. These factors have not been incorporated into the one-loop and two-loop designs and would, to some extent, result in reduced costs in those heat exchangers. Limited time and workscope did not permit in-depth attempts at cost optimization of any of the heat exchangers, but it is recognized that performance and implementation of parametric studies would result in cost savings.

TABLE 20-1
RECUPERATOR COMPARISON FOR ONE-, TWO-, AND THREE-LOOP PLANTS

	One-Loop Plant	Two-Loop Plant	Three-Loop Plant
Loop rating [MW(t)]	1530	1500	1000
Recuperator rating [MW(t)/HX]	1253	1253	918
Number of HXs/loop	1	1	1
HX surface [m^2 (ft^2)/HX]	42,312 (455,406)	42,312 (455,406)	28,400 (305,732)
Number of tubes/HX	94,668	94,668	66,732
Number of modules/HX	Hex. 161	Hex. 161	Hex. 83
Number of tubes/module	588	588	804
Tube size [cm (in.)]	1.1113 x 0.0813 (0.4375 x 0.032)	1.1113 x 0.0813 (0.4375 x 0.032)	1.1113 x 0.114 (0.4375 x 0.045)
ISI/repair	Module	Module	Module
Shipping mode	Barge	Barge	Barge
Shop/site assembly	Shop	Shop	Shop
Relative cost/HX	1	0.86 1.72 for 2	0.58 1.73 for 3
Diameter [m (ft)]	6.80 (22.3)	6.70 (22)	5.64 (18.5)
Height [m (ft)]	26.1 (85.6)	21.2 (69.4)	20.4 (67)
Weight [tonnes (tons)]	1043 (1027)	1041 (1025)	813 (800)

TABLE 20-2
PRECOOLER/INTERCOOLER (P/I) COMPARISON FOR ONE-, TWO, AND THREE-LOOP PLANTS

	One-Loop P/I	Two-Loop P/I	Three-Loop P/I
Loop thermal rating [MW(t)]	1530	1500	1000
HX thermal rating [MW(t)]	533/337	533/337	581/-
Number of HXs/loop	1/1	1/1	1/-
Diameter [m (ft)]	4.9/4.7 (16/15/5)	4.9/4.7 (16/15.5)	4.6 (15)
Height [m (ft)]	23.8/18.3 (78/60)	23.8/18.3 (78/60)	19.9 (65)
Weight [tonnes (tons)]	559/412 (550/405)	559/412 (540/400)	488 (480)
Number of tubes	1196/1118	1196/1118	832
Heat transfer surface relative to one-loop precooler	1/0.86	1/0.86 (1.86 x 2 = 3.72)	0.88 (0.88 x 3 = 2.65)
Log mean temperature difference [°C (°F)]	9.6/1.6 (49.2/34.8)	9.6/1.6 (49.2/34.8)	12.7 (54.9)
Relative cost/HX (one-loop recuperator as base)	0.41/0.36 x 1 = 0.77	0.34/0.30 x 2 = 1.28	0.26 x 3 = 0.78
	R = 1.0	R = 1.72	R = 1.73
	P/I = 0.77	P/I = 1.28	P = 0.78
Total	1.77	3.00	2.51

There are several key constraints within which the cost reduction studies must be made, and these constraints have not yet been clearly defined.

20.2. GA EFFORT

20.2.1. GA Scope

The scope of GA's portion of the heat exchanger design effort is as follows:

1. Coordinate the work by the heat exchanger subcontractor (CE).
2. Evaluate the heat exchanger designs developed by CE and identify any areas which require more detailed study. These areas will be analyzed in more detail by GA and CE.
3. Evaluate alternate heat exchanger design approaches to increase safety, increase reliability, reduce cost, and reduce maintenance downtime.
4. Together with CE, review the HHT (Sulzer) heat exchanger drawings and criteria in order to establish a common design basis for the U.S. and European GT-HTGR programs.

20.2.2. GA Heat Exchanger Summary

Following completion of CE's work, described in Section 20.1, the GA Heat Exchanger Department initiated work in the second quarter of FY-79 to become familiar with the conceptual designs prepared by CE and to become involved in the development of the design configurations, alternatives, and feasibility of the heat exchanger concepts proposed. The GA Heat Exchanger

Department became involved in the design of the heat exchangers at this time for two principal reasons. Since GA is the GT-HTGR systems designer, it is necessary that technical heat exchanger information in support of the system design development be available in-house and that integration of the heat exchangers into the system and with related components and the balance of plant (BOP) be conducted to satisfy all interfacing requirements. Therefore, the two main tasks performed by the GA Heat Exchanger Department were:

1. Review CE's heat exchanger conceptual designs and backup information to enable analytical models of the heat exchangers to be formulated and incorporated in the system optimization/analysis computer program, CODER. This work relied heavily on the results of the activities in item 2, below.

2. Conduct a review, and analysis as required, of the CE heat exchanger designs in order to (1) first gain a full understanding of the conceptual designs of the components and (2) subsequently verify feasibility and envelope, determine component performance characteristics, evaluate the capability of component internal and external interfaces to satisfy criteria and requirements imposed, among them being seismic adequacy, removability/replaceability, and ISI, and define design issues requiring further resolution in order to achieve viable component designs. This work, including the interfacing with CE, is also in support of the interfacing with the HHT heat exchanger designs by Sulzer to be conducted later in the year, aimed at achieving commonality of European and U.S. heat exchanger designs.

To accomplish the above tasks, GA began review and familiarization work on principally the CE recuperator and precooler conceptual designs in the second quarter of FY-79. No effort was expended directly on the intercooler designs, since they are essentially the same as the precoolers. Preliminary algorithms of the thermal-hydraulic characteristics of the recuperator and precooler were prepared and submitted for incorporation into the CODER code. These algorithms were provided on a preliminary basis to serve as a starting point for analysis and optimization work. They will be updated and refined to more accurately represent the heat exchanger characteristics later in the year and as more detailed information unfolds on the conceptual design of these two components.

Review of the mechanical, thermal-hydraulic, and structural aspects of the recuperator and precooler designs was initiated to gain the necessary familiarization and understanding of the work done by CE. Analyses were started on the structural and seismic characteristics of principally the recuperator. The precooler design is a relatively low-temperature, helical bundle design, and in many ways it parallels the helical bundle design for the SC-HTGR steam generators. Experience on steam generator design was applied in the precooler review, resulting in the conclusion that concerns in this area were not significant. Therefore, work was focused principally on the recuperator. The recuperator, which is a higher-temperature component of a straight tube, integral return tube configuration, was quickly identified as the component which should receive the most attention.

The structural and seismic review of the recuperator will continue, leading toward formulation of design issues to be jointly reviewed with CE.

The mechanical design review focused on the mechanical design aspects of the two heat exchangers, emphasizing the recuperator, and included manufacturing concerns. Efforts concentrated on mechanical and manufacturing issues worthy of further attention, many of which have a direct bearing on mechanical design, structural adequacy, and performance.

The thermal-hydraulic review and assessment of the precooler and, primarily, the recuperator yielded a number of issues related to pressure loss and gas bypasses. These issues, again, were closely interrelated with the mechanical and structural adequacy of the conceptual design configurations.

These review and analysis tasks will continue through the third and fourth quarters of FY-79, during which time they will be reviewed with CE. Certain issues will be selected for further work and resolution, leading to joint GA/CE/Sulzer reviews and selection of the recommended heat exchanger design concepts to be adopted for the GT-HTGR commercial plant.

21. PLANT PROTECTION (6332) AND PLANT CONTROL (6333) SYSTEMS

21.1. SCOPE

The purpose of these tasks is to establish conceptual design requirements and interfaces for the PPS and PCS.

This work includes a review of data which had previously been generated, identification of major technical problems, and the preparation of conceptual system schematics leading to the initiation of block diagram development.

21.2. SUMMARY

The review of previous work in the PCS and PPS areas was begun in the second quarter of FY-79 and has led to the identification of three critical items that require immediate attention:

1. Determine the controllability of proposed hydraulic valves.

2. Determine the technical and licensing feasibility of assigning combined control and backup safety functions to one valve.

3. Develop the necessary electrical technology to assess the impact of bringing main generator power cables into the reactor building, which affects separation, isolation, and noise rejection.

The first problem will be addressed when requested mid-year funding becomes available. The second problem is being addressed to the extent allowed by the current budget. It is proposed to address the third problem with the generic funding requested for FY-80 and FY-81.