# GULF GENERAL ATOMIC

Gulf-GA-A12651

# HTGR GAS TURBINE POWER PLANT CONFIGURATION STUDIES

by

R. G. Adams, F. R. Bell,

C. F. McDonald, and D. C. Morse



This is a preprint of a paper to be presented at the ASME Winter Annual Meeting, November 11-15, 1973, Detroit, Michigan, and to be printed in the Proceedings.

Work supported in part by the U. S. Atomic Energy Commission, Contract AT(04-3)-167, Project Agreement No. 46.

Gulf General Atomic Projects 0435 and 0352 July 26, 1973

GULF GENERAL ATOMIC COMPANY
P.O. BOX 81608, SAN DIEGO, CALIFORNIA 92138

# HTGR GAS TURBINE POWER PLANT CONFIGURATION STUDIES

## ABSTRACT

The High Temperature Gas Cooled Reactor (HTGR) gas turbine power plant concept combines the existing design HTGR core with a closed-cycle helium gas turbine power conversion system which is driven by the reactor coolant helium. The use of a gaseous working fluid enables the heat rejection to be well matched to the capabilities of dry cooling towers which can minimize environmental impact while conserving water resources. The objective of the studies reported was to establish the main features of the reactor and gas turbine power conversion system layout for subsequent preliminary design studies.

Initial decisions were made to adopt an integrated arrangement (gas turbine loops located within the prestressed concrete reactor vessel) embodying multiple turbomachinery loops, based on a non-intercooled cycle with a high degree of heat recuperation.

The plant configuration studies consisted of the preparation and evaluation of alternative designs incorporating various orientations of the turbomachinery and heat exchangers with respect to the reactor core and primary and secondary containment systems. Design studies were made of single-shaft and split-shaft turbomachinery in conjuction with both horizontal and vertical generators and were evaluated on the basis of control, safety, and maintenance.

HTGR Gas Turbine Power Plant
Configuration Studies.

## INTRODUCTION

The HTGR gas turbine power plant is the combination of the helium-cooled High-Temperature Gas-Cooled Reactor (HTGR) with a closed-cycle, helium gas turbine power conversion system. The HTGR is developed and commercially accepted for use with modern steam turbine plants. The required technology is currently available for design and development of the helium gas turbine, heat exchangers, and control valves, and for the reactor modifications. The HTGR gas turbine power plant is exceptionally well suited to economical rejection of waste heat directly to the air using dry cooling cooling towers, although other forms of cooling can of course be used.

Although gas-cooled reactors have, with a few exceptions, been developed for use with steam turbine power conversion systems, the potential benefits for a closed-cycle gas turbine driven by the reactor coolant gas have long been recognized. The principal potential advantages foreseen for such a direct-cycle nuclear power plant are (1) more efficient use of the high-temperature capability of the reactor without the temperature degradation that necessarily occurs in the steam generator of an indirect-cycle plant, (2) simplification through a reduction in the number of systems and components, (3) a more compact power conversion system because of the high-density working fluid achievable in the closed-cycle gas turbine, and (4) economical adaptability to dry cooling.

The closed-cycle gas turbine is by no means new and as reported by Keller(1)[1] has been used for the last 30 years (mainly in Europe) in fossil fuel fired plants for electrical power generation combined with district heating. Nine fossil-fired closed cycle gas turbines with outputs ranging up to 30 MW(e) are in service.

The fundamental difficulty that has hindered development of the closed-cycle gas turbine for use with fossil fuel resides in the means for getting chemically produced heat into the cycle, calling for a heat exchange surface necessarily operating even hotter than the peak working fluid temperature, and facing at the same time the problems associated with corrosion by combustion products. Nuclear power eliminates this problem altogether by permitting direct heating of the clean, non-corrosive working fluid.

Current investigations at Gulf General Atomic are aimed at producing the preliminary design, performance, engineering, and economic data needed for assessment of commercial application of HTGR closed-cycle gas turbine power

[1] Underlined numbers in parentheses designate References at end of paper.

plants in the 1100 MW(e) class. Environmental aspects of the HTGR gas turbine power plant have been reported by Bell and Koutz (2).

This paper outlines the initial phase of the preliminary design program in which different plant configurations were studied. The basic thermodynamic cycle is described and preliminary design aspects of the sizing of the helium gas turbine and heat exchangers, together with layout studies of the reactor and the power conversion system are discussed. A summary of the evaluation of the various plant configurations from the standpoints of performance, control, safety, and maintenance is included.

The HTGR gas turbine power plant configuration studies outlined in this paper were accomplished by Gulf General Atomic with support by a group of utility companies and by the United States Atomic Energy Commission. Plant component and layout studies were carried out with utility company support, and the complimentary program on plant control, safety, and maintenance was supported by the AEC under contract AT(04-3)-167, Project Agreement No. 46.

During this initial preliminary design phase of the program three leading turbomachinery companies participated on a cooperative basis, each working independently to a common set of requirements by GGA. The participating turbomachinery companies are: General Electric Company, Gas Turbine Products Division, Pratt and Whitney Aircraft, Division of United Aircraft Corporation, and Brown Boveri-Sulzer Turbomachinery, Ltd., Switzerland. The turbomachinery designs and plant configurations outlined in this paper however, are based on work performed by Gulf General Atomic.

## THERMODYNAMIC CYCLE

The basic Temperature - Entropy diagram for the HTGR gas turbine is shown in Figure 1, and a simplified cycle schematic, with the temperatures and pressures throughout the system is shown in Figure 2. The primary fluid, is heated in the reactor core to 1500$^{\circ}$F (only slightly higher than the temperatures in todays commercial HTGRs) and is then expanded through a turbine, decreasing the pressure from 960 psia to 464 psia, and decreasing the temperature to 1042$^{\circ}$F. The turbine provides the power necessary to drive the compressor and also drives the electrical generator. At the turbine exit all useful work has been extracted from the helium but it still has a great deal of useful thermal energy. This energy from the turbine exit is transferred via a recuperator (heat exchanger) to the cooler high pressure helium that is about to enter the reactor core. From Figure 1 it can be seen that the magnitude of this internal energy transfer within the cycle is of the same order as the reactor thermal input, and as outlined in the next section, this emphasizes the importance of the recuperator in a closed-cycle gas turbine system. At 443$^{\circ}$F, all energy useful to the power cycle has been extracted. The remaining heat to be rejected is removed by a water-cooled heat exchanger (precooler), transported by a circulating water loop, and ultimately rejected to the ambient air through a dry cooling tower. The 105$^{\circ}$F

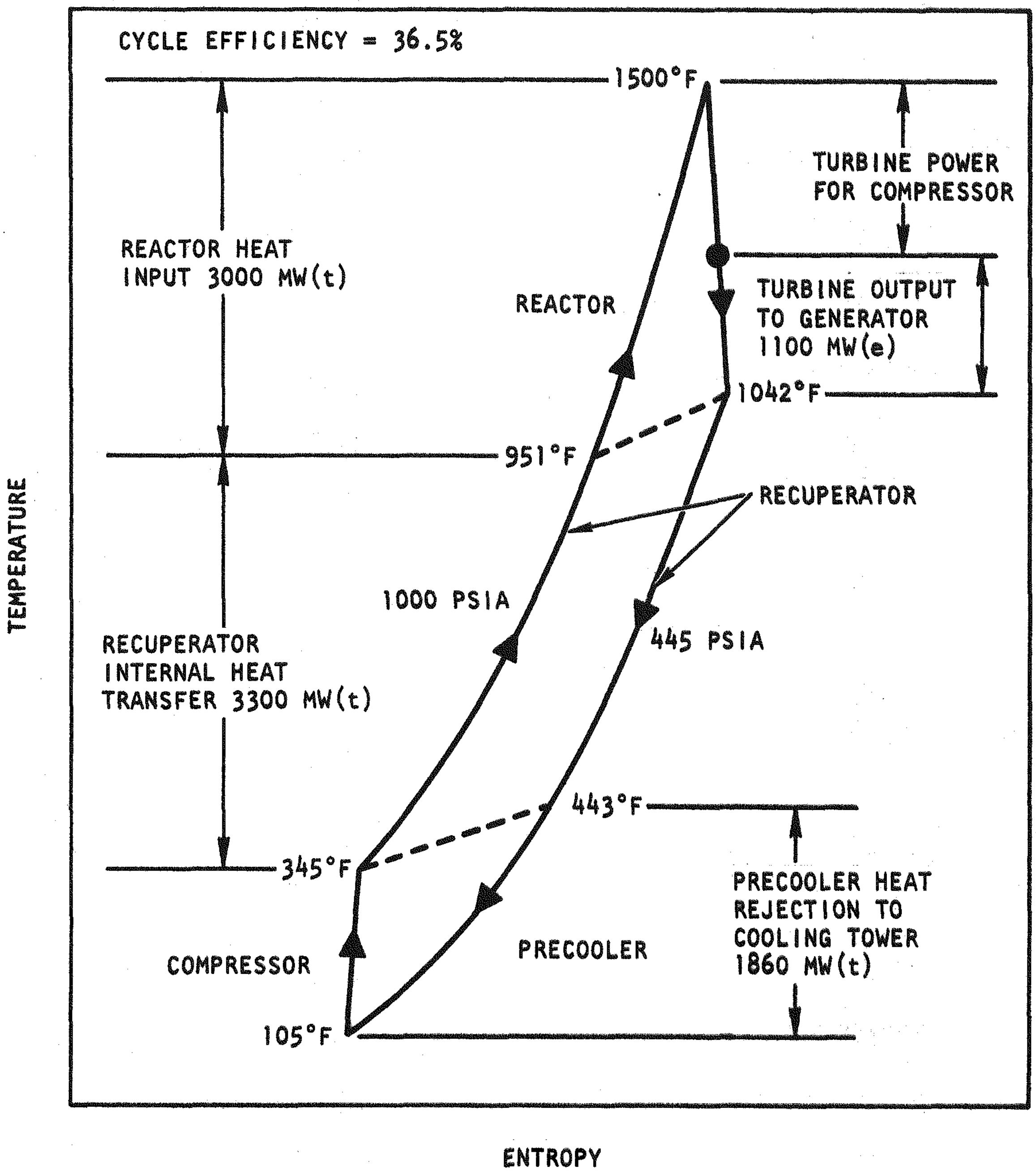



Figure 1. HTGR Gas Turbine Thermodynamic Cycle

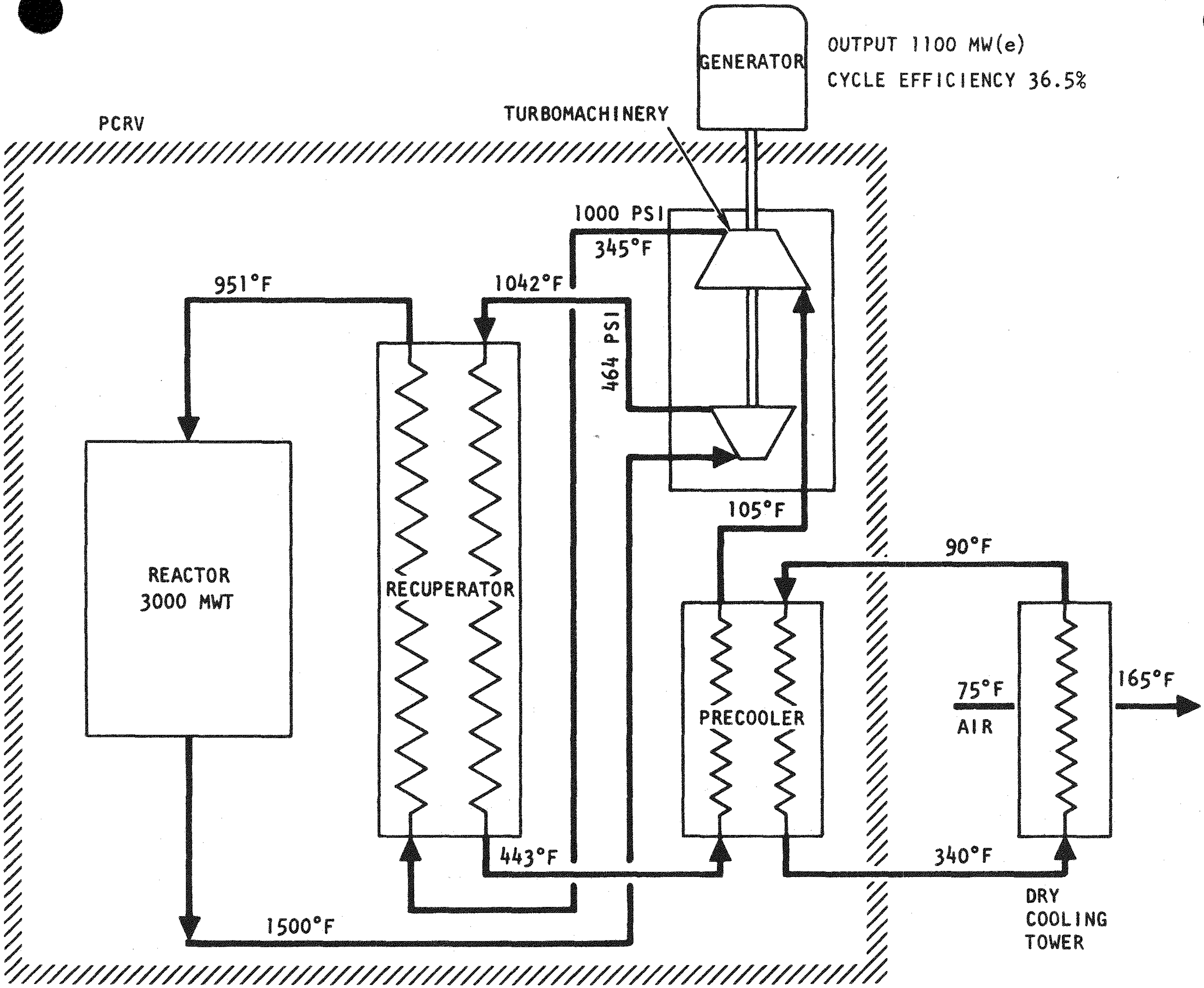



Figure 2. HTGR Gas Turbine Cycle with Dry Cooling

helium is then compressed back to 1000 psia, becomes the receiver of recuperator heat, and at 955°F enters the reactor core, completing the cycle.

The unique thermodynamic feature of this HTGR gas turbine system is that heat is rejected at a sufficiently high temperature (340°F water temperature from the precooler) to permit economical application of dry cooling towers. Aspects of the HTGR gas turbine power plant related to dry air cooling and reject heat utilization have been reported by Fortescue (3, 4), Krase (5) and Sager (6).

## MAJOR CYCLE PARAMETERS

Two of the major parameters which have a significant effect on the cycle, namely reactor outlet temperature and maximum system pressure, were established at the onset of the program. In line with the principle of fullest use of existing technology and designs for the gas turbine plant, the reactor core design for the 1160-MW(e) HTGR steam-cycle power plant has been utilized. For this 3000 - MW(t) core design a mixed mean reactor outlet gas temperature of 1500°F could be provided with the present fuel without exceeding the current HTGR maximum fuel temperature limits. This design approach not only minimized the reactor development involved, but also relieved the turbine design from the need for advanced materials or special blade cooling requirements, since the 1500°F turbine inlet temperature is modest by existing long-life industrial gas turbine standards. The selected maximum system pressure level of 1000 psia was established for consistency in the design of the prestressed concrete reactor vessel (PCRV) with the 700 psia normal working pressure of the HTGR steam plants and the 1250 psia pressure of Gas Cooled Fast Reactor design studies.

The influence of compressor pressure ratio and recuperator effectiveness on cycle efficiency is shown in Figure 3. While maximum cycle efficiency is the obvious goal, the choice of pressure ratio and recuperator effectiveness were strongly influenced by turbomachinery design considerations, and heat exchanger space availability within the PCRV side wall cavities, respectively. From Figure 3 it can be seen that a high degree of recuperation is necessary for acceptable levels of cycle efficiency. A pressure ratio of 2.25, slightly lower than that for maximum efficiency, was selected in conjuction with a recuperator effectiveness of 0.87. The incentive for chosing a slightly lower pressure ratio is that with the associated reduced number of stages the length of the turbomachinery is less, and this eases bearing requirements, and improves shaft dynamic characteristics. The selected pressure ratio also gives a turbine exhaust volume flow rate which is close to the minimum value. Since the last turbine stage is the largest element of the rotating assembly, and tends to determine the overall diameter of the turbomachinery, there is an obvious incentive to select cycle parameters to give minimum exit volumetric flow.

The major design parameters chosen for the initial phase of the preliminary design program are given on Table 1. The computed cycle efficiency of 36.5

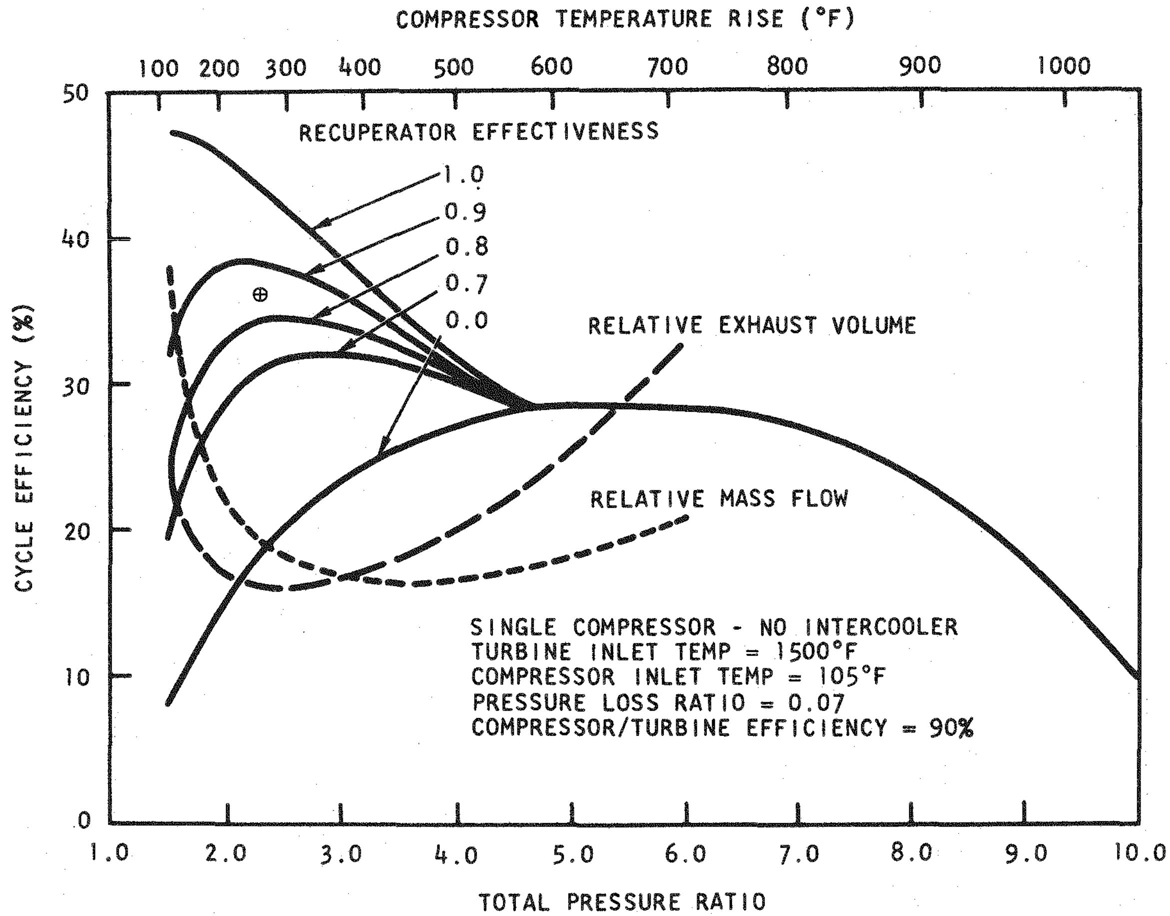



Figure 3. Cycle Efficiency For HTGR Gas Turbine As A Function of Compressor Pressure Ratio And Recuperator Effectiveness.

TABLE 1

HTGR GAS TURBINE MAJOR DESIGN PARAMETERS

| | |
|---|---|
| Compressor pressure ratio | 2.25 |
| Turbine inlet temperature | 1500°F (815°C) |
| Compressor inlet temperature | 105°F (41°C) |
| Compressor discharge pressure | 1000 psia (70 bar) |
| Plant pressure loss ( $\Delta p/p$) | 0.07 |
| Recuperator effectiveness | 0.87 |
| Compressor adiabatic efficiency | 90% |
| Turbine adiabatic efficiency | 90% |
| Compressor helium flow rate | 15,160,000 lb/hr |
| Precooler water inlet temperature | 90°F (32°C) |
| Precooler water outlet temperature | 340°F (171°C) |
| Turbine disk cooling flow | 1.0% |
| Compressor leakage flow | 0.5% |
| Power turbine bypass valve leakage | 0.1% |
| Compressor bypass valve leakage | 0.2% |
| Bearing losses (based on net output) | 1.0% |
| Generator efficiency | 98% |
| Primary system heat loss | 25 MW(t) |
| Station auxiliary power | 5 MW(e) |
| Reactor thermal power | 3000 MW(t) |
| Plant net electrical output | 1096 MW(e) |
| Cycle efficiency | 36.5% |

percent is for a dry cooled plant, and is estimated to increase to 37.1 percent, and 37.9 percent, respectively for wet tower and once-through water cooled versions of the same power plant.

## OVERALL POWER PLANT DESIGN CONSIDERATIONS

Early in the HTGR gas turbine program, the following points of design philosophy were established as guiding principles throughout the program: (1) make maximum use of existing HTGR and gas turbine technology; (2) simplify systems, even at the expense of slight efficiency penalties; (3) use conservative design parameters; and (4) give major attention to safety, reliability, and maintainability early in the design.

The studies outlined in this paper were directed at the assessment of various plant configurations and decisions in the following areas were made:

- o Integrated Vs. Non-Integrated Circuit.
- o Single Vs. Multiple Power Conversion loops.
- o Intercooled Vs. Non-intercooled Cycle.
- o Split-shaft Vs. Single Shaft Turbomachinery.
- o Horizontal Vs. Vertical Machinery.

Decisions in the first three areas were made early in the program and the remainder of the preliminary design phase was devoted to component design and arrangement studies related to the latter two areas. The configuration studies incorporating various orientations of the turbomachinery and heat exchangers with respect to the reactor core and primary and secondary containment systems are discussed in the next section. The sizing of the helium turbomachinery and heat exchangers together with aspects of the primary containment are discussed in a later section of this paper. Initial decisions in the first three areas mentioned had a strong influence on the overall power plant preliminary design as outlined below:

### Integrated Vs. Non-Integrated Circuit.

In early nuclear gas turbine plant designs, as in early reactors of all types, the reactor was in one steel vessel and the components were in others with connecting pipes. With the adoption of the prestressed concrete reactor vessel (PCRV), all major builders of gas-cooled reactors soon settled on integrated circuit configurations in which the complete primary circuit was contained in one PCRV.

The non-integrated type of design having the turbomachinery external to the reactor vessel would have eased certain mechanical problems and faciliated development and testing by providing a more ready access to the turbomachinery for initial modifications and instrumentation as well as providing a generally wider degree of machinery design freedom. However, this would necessarily have involved discarding the prime advantage of the PCRV type of containment which, as applied to the present HTGR steam plant, eliminates the failure of large external ducting as a potential cause of rapid depressurization. In direct-cycle applications, necessarily involving large ducting, the consequences of such failures would be particularly serious. With the integrated design concept the reactor, turbomachinery, heat exchangers, and the entire helium inventory are enclosed within the PCRV.

For reasons of safety and integrity the integrated design concept was selected for the HTGR gas turbine preliminary design. The integrity of the PCRV is the result of the highly redundant prestressing tendons and strands which are not subject to neutron embrittlement and thermal cycling. In the preliminary design phase of the gas turbine program, integrated circuit concepts embodying a PCRV similar to the type developed for the forthcoming 1160MW(e) HTGR steam plant, as shown on Figure 4 and reported by Waage (7), were selected so that existing PCRV technology could be utilized.

Single Vs. Multiple Power Conversion Loops.

Factors that have a major influence on the plant design are the number of gas turbine loops per reactor and, in particular, the question of one loop versus multiple loops. In the qualitative analysis of the factors involved it become apparent that the multiple loop design approach was not only desirable, but mandatory, to satisfy the basic design considerations outlined earlier. The influencing features in the adoption of the multiple loop approach are brifely outlined below.

A consideration in adopting multiple loops is that they provide extra redundancy in emergency cooling situations. For a plant with a single gas turbine, all emergency cooling function must be provided by additional emergency cooling loops, which increase the cost and complexity of the emergency cooling system for the single loop power conversion system.

With multiple loops the turbomachinery is smaller and the stresses in the rotating components are reduced thus eliminating the need for advanced turbine materials. With the multiloop gas turbine design, a multicavity PCRV design similar to that used on the large steam-cycle HTGRs can be used, with the gas turbine modules and heat exchangers being located in 14 ft. diameter wells in the PCRV sidewalls. The multiloop design approach was also in line with the retention of a shipping envelope consistent with factory assembly and relatively unrestricted shipment.

The desireability of non-nuclear testing a complete prototype gas turbine loop favors the selection of multiple loops since the fossil-fired test of the prototype

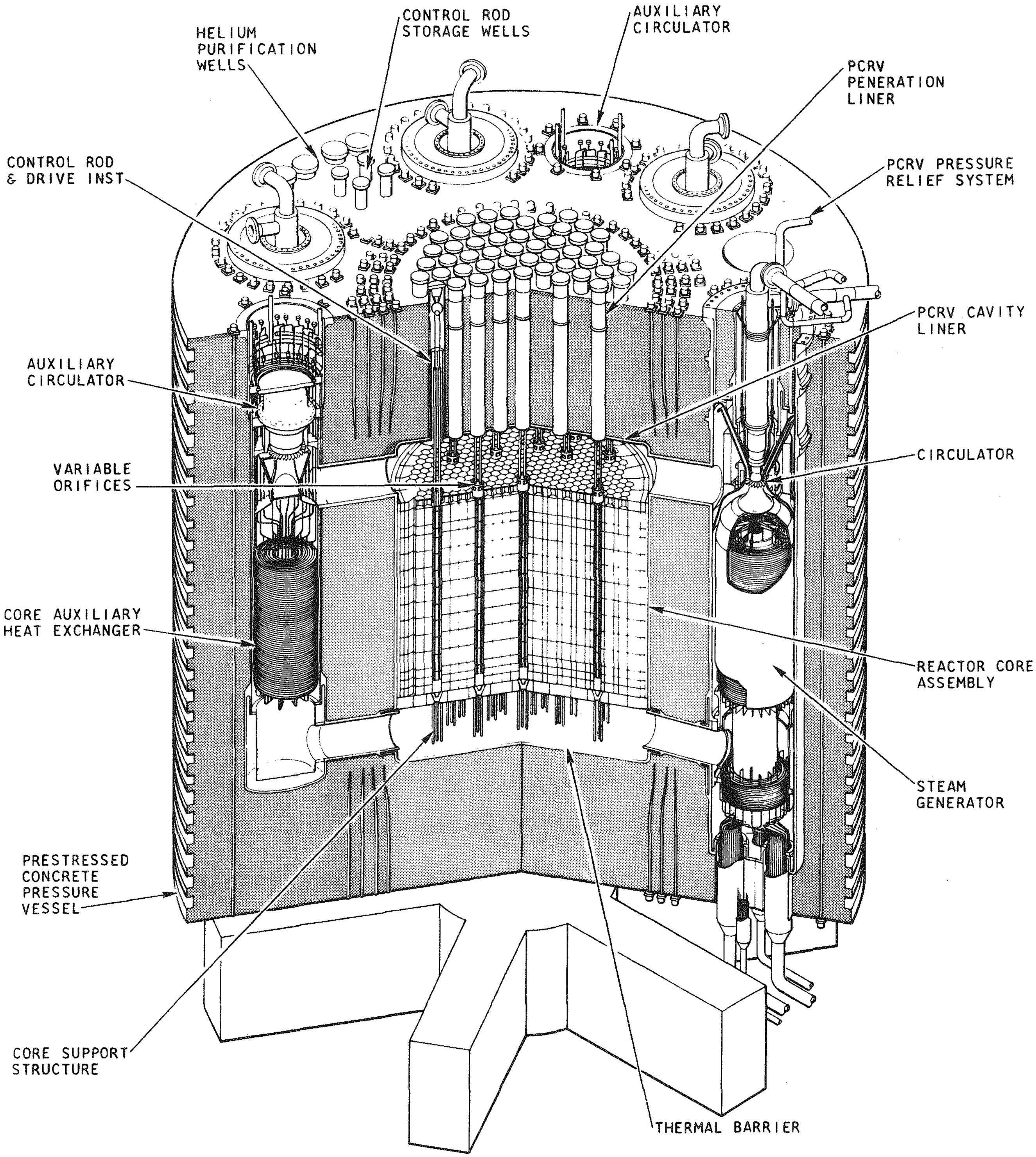



Figure 4. HTGR Nuclear Steam Plant Core Arrangement.

gas turbine loop will be more practical and less expensive. With a multiloop design a departure from strictly on-site repair, substituting instead the philosphy of part replacement by spare power conversion modules, is rendered possible by virtue of the relatively small bulk and limited cost of the individual components of the multiple loops and their possible common use in other plants.

A first sight a single gas turbine loop would appear somewhat simpler and more conventional, but when considering that use of advanced turbine materials is necessary, in conjunction with complex helium ducting between the turbomachinery, reactor, and multiplicity of heat exchangers, it was concluded that an integrated multiple loop configuration came closer to satisfying all of the plant safety, reliability, maintenance, and economic goals.

For the 1100 MW(e) plant a preliminary design embodying four turbomachinery loops results in turbomachinery sizes that are within transportation limits, can be accomodated in the PCRV sidewalls, and are dimensionally comparable with large industrial open-cycle air breathing gas turbines.

## Intercooled Vs. Non-Intercooled Cycle.

In any operation involving the compression of gases by machines having more than one stage, the efficiency of the process can be increased by cooling the gas between the stages. Non-integrated fossil-fired, closed-cycle gas turbines employ one or more intercoolers between compressor sections, however, it was felt that such units would seriously detract from the simplicity and maintainability of an integrated nuclear gas turbine. Studies showed that while the intercooler offers a cycle efficiency increase of about 3 efficiency points, this advantage appeared to be offset by the added installation complexity of the system.

It was found that the addition of even a single intercooler made a two-bearing gas turbine system no longer possible, and the need for additional bearings not only increases the capital cost of the system but tends to reduce reliability. The more complex compressor unit (including longer shaft), interconnecting ducting, more complex PCRV arrangement, and separate water cooling loop and heat rejection system for the intercooler, were some of the effects that tended to increase capital costs and thus reduce the apparent cost advantage of the higher efficiency intercooled cycle. An inherent disadvantage of the intercooled cycle, from the dry cooling standpoint, is that the associated lower extreme temperature difference across the tower heat exchanger results in a size and surface area penalty. Consistent with one of the major goals of the plant design, namely keeping the system simple, even at the expense of slight performance penalties, a non-intercooled cycle was selected.

## GAS TURBINE PLANT CONFIGURATION ALTERNATIVES

The plant configuration investigation consisted of the preparation and evaluation of layouts showing various orientations of the turbomachinery, heat exchangers, and reactor core, with primary and secondary containment systems.

the study was an iterative process whereby attractive and adverse features surfaced as the plant design evaluation progressed.

In support of the configuration studies, aerothermodynamic and dynamic analyses, and sizing of the major plant components were carried out. Details of these are given in the next section of the paper. With simplicity, safety, and ease of maintenance as major considerations, many design arrangements embodying the above features were reviewed, and three plant configurations as shown in a simplified form on Figure 5 were selected for more detailed study. Evaluations of the three selected power plant configurations were made to assist in making the two remaining major design considerations; namely to use single-shaft or split-shaft turbomachinery, and whether to use vertical or horizontal machinery. Various aspects of the three selected plant configurations are discussed below.

## Configuration 1. Split-Shaft Turbomachinery and Horizontal Generator

The general arrangement for this plant design is shown in Figure 6, and from the plan view it can be seen that two vertical cavities per loop are required. With a split-shaft turbomachinery arrangement the compressor and compressor-turbine are on one shaft, (vertical in this case) and the power turbine and generator (horizontal) are on another shaft. The cylindrical reactor vessel has an outside diameter and height of 117 ft and 97.5 ft respectively. Each of the eight 14 ft. diameter cavities surrounding the core cavity are closed with removable plugs of composite steel and reinforced concrete construction to facilitate installation, removal, and maintenance of components. The side cavities are arranged in pairs, each pair of cavities, together with interconnecting ducts, and ducts carrying helium to and from the core cavity, forming one loop of the four-loop system. One cavity houses the precooler with the vertical turbo-compressor group mounted above it. In the precooler cavity the cooling water lead pipes penetrate the bottom head of the vessel and are connected to the header drums, which in turn are manifolded with pipes to and from the tower cooling water loop. The other cavity contains the recuperator with the horizontal power turbine mounted in the top section. The power turbine drive shaft penetrates the vessel and is connected to the generator which is mounted to the PCRV wall on a heavy structural steel foundation. As shown, the generator will be a conventional horizontally mounted type. Although not shown in detail the plant has a secondary containment building which completely encloses the PCRV and the entire power conversion system.

The helium flow through the closed loop starting from the reactor is as follows. From the core cavity the hot high pressure helium flows up through the duct to the vertical compressor turbine and after expansion flows through a cross duct in the vessel to the horizontal power turbine. From the power turbine the low pressure helium discharges down through the recuperator (outside the tubes) and via another cross duct enters the precooler cavity. Flowing upwards through the precooler (helium flow outside the tubes) the gas is cooled to the compressor inlet temperature. From the compressor the high pressure helium flows through a cross duct to the recuperator cavity and passes down through a vertical duct in the center of the heat exchanger to the bottom header where the gas enters the

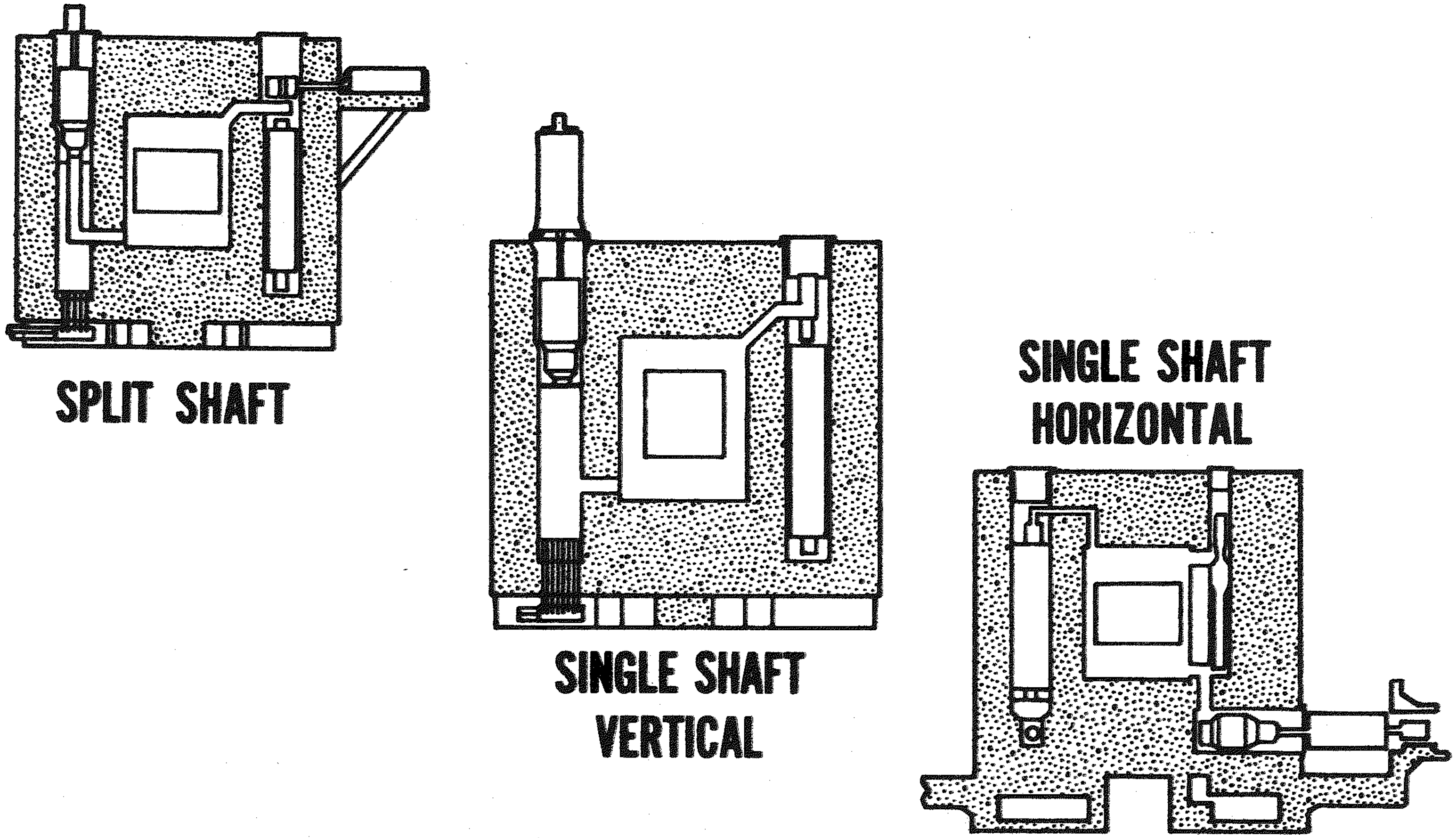



Figure 5. Simplified Views of HTGR Gas Turbine Plant Configuration Alternatives.

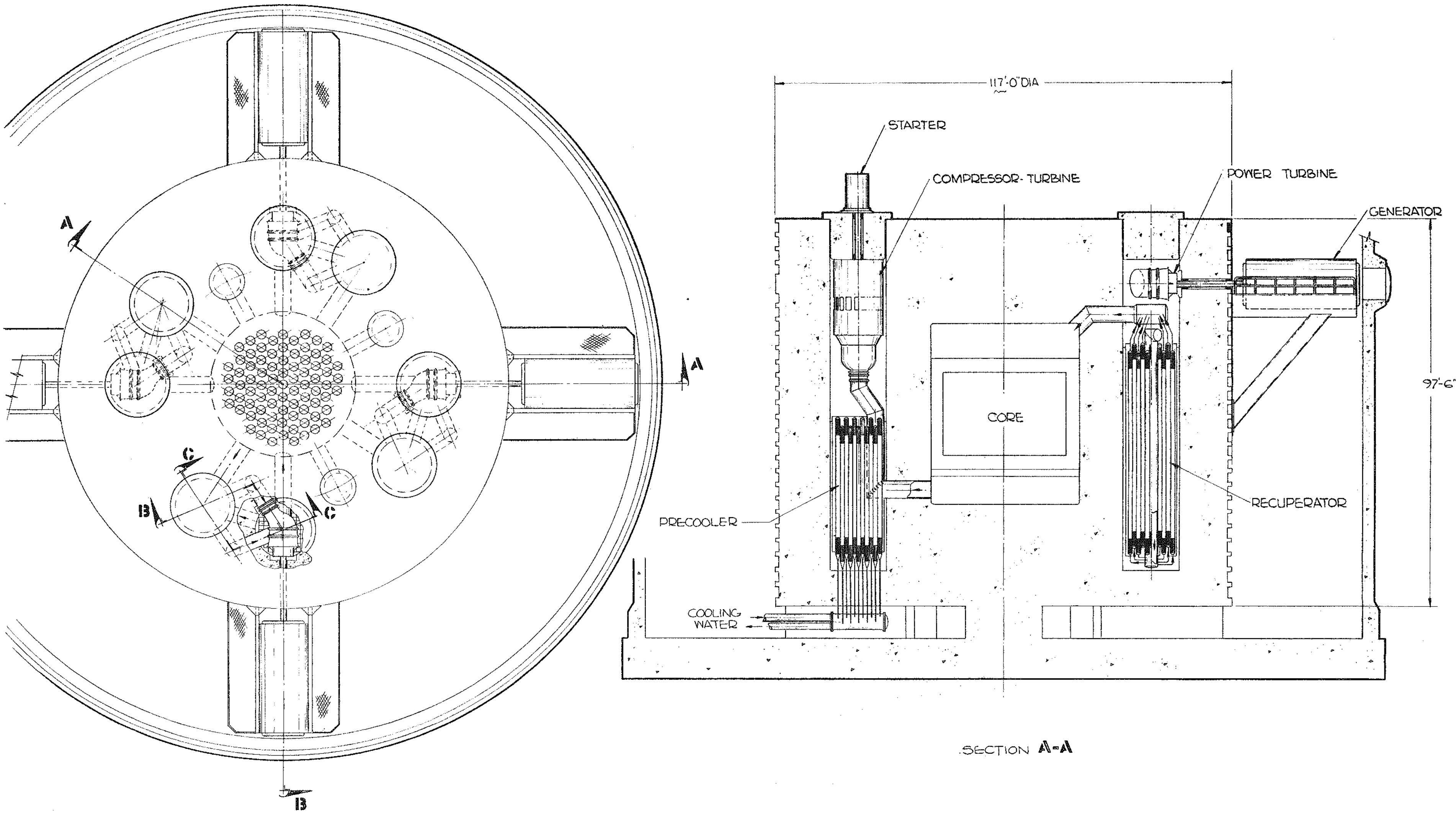



Figure 6. Plant Configuration 1. Showing Split-Shaft Turbomachinery and Horizontal Generator

tube bundle. (The vertical duct is necessary to give a counterflow arrangement in the recuperator). The high pressure gas flows up through the recuperator tubes and with regained heat flows through a cross duct and back into the reactor completing the loop.

## Configuration 2. Single-Shaft Turbomachinery and Vertical Generator

The general arrangement for this plant design is shown on Figure 7, and with two vertical cavities per loop, the outside dimensions of the cyclindrical vessel are the same as for Configuration 1. In this plant schematic it can be seen that the complete turbomachinery assembly is mounted above the precooler in one cavity, and the axial counterflow recuperator is housed in the other cavity. The single-shaft turbine unit is supported from the underside of the concrete closure plug of the precooler cavity and is directly connected to a generator, which is mounted on top of the plug. The generator mounting and all connections will be provided with suitable features to facilitate shifting the generator for plug and turbomachinery removal. The combined end thrust and weight of the generator and turbomachinery are taken on a thrust bearing located outside the PCRV for accessibility and ease of maintenance.

The helium flow is essentially the same as for the previous configuration with the exception that all of the expansion takes place in one turbine assembly. (i.e. turbine, compressor and generator all on the same shaft). From the turbine the gas flows through a cross duct in the vessel to the recuperator cavity, and enters the concentric header (necessary to give a counterflow arrangement) at the top of the heat exchanger. Completion of the loop in the closed-cycle system is the same as outlined above for Configuration 1.

## Configuration 3. Single-Shaft Turbomachinery and Horizontal Generator.

The general arrangement for this plant design is shown on Figure 8, and while the diameter of the vessel remains the same as in the previous configurations, the height of the PCRV has increased to 106 ft. due to the provision of horizontal cavities for the turbomachinery in the bottom section of the vessel. It can be seen that in addition to the eight vertical 14-ft-diameter recuperator and precooler cavities and the three auxiliary cooling system cavities, four horizontal cavities 14 ft in diameter by about 37 ft long are provided for the turbomachinery in the bottom of the PCRV. This configuration thus requires two vertical cavities and one horizontal cavity per loop. The generators are mounted on supports integral with the PCRV and are driven from the turbines by shafts which penetrate the turbomachinery cavity closures.

The horizontal turbomachinery is located at the bottom of the PCRV between the recuperator and precooler cavities which are connected by cross ducts. From the core cavity the hot high pressure helium flows down through a short duct into the turbine, where after expansion it discharges into a cross duct and flows up through the recuperator tube bundle. Leaving the recuperator at the top, the helium flows through a cross duct in the vessel and flows down through the precooler tube bundle where it is cooled down to the compressor inlet temperature.

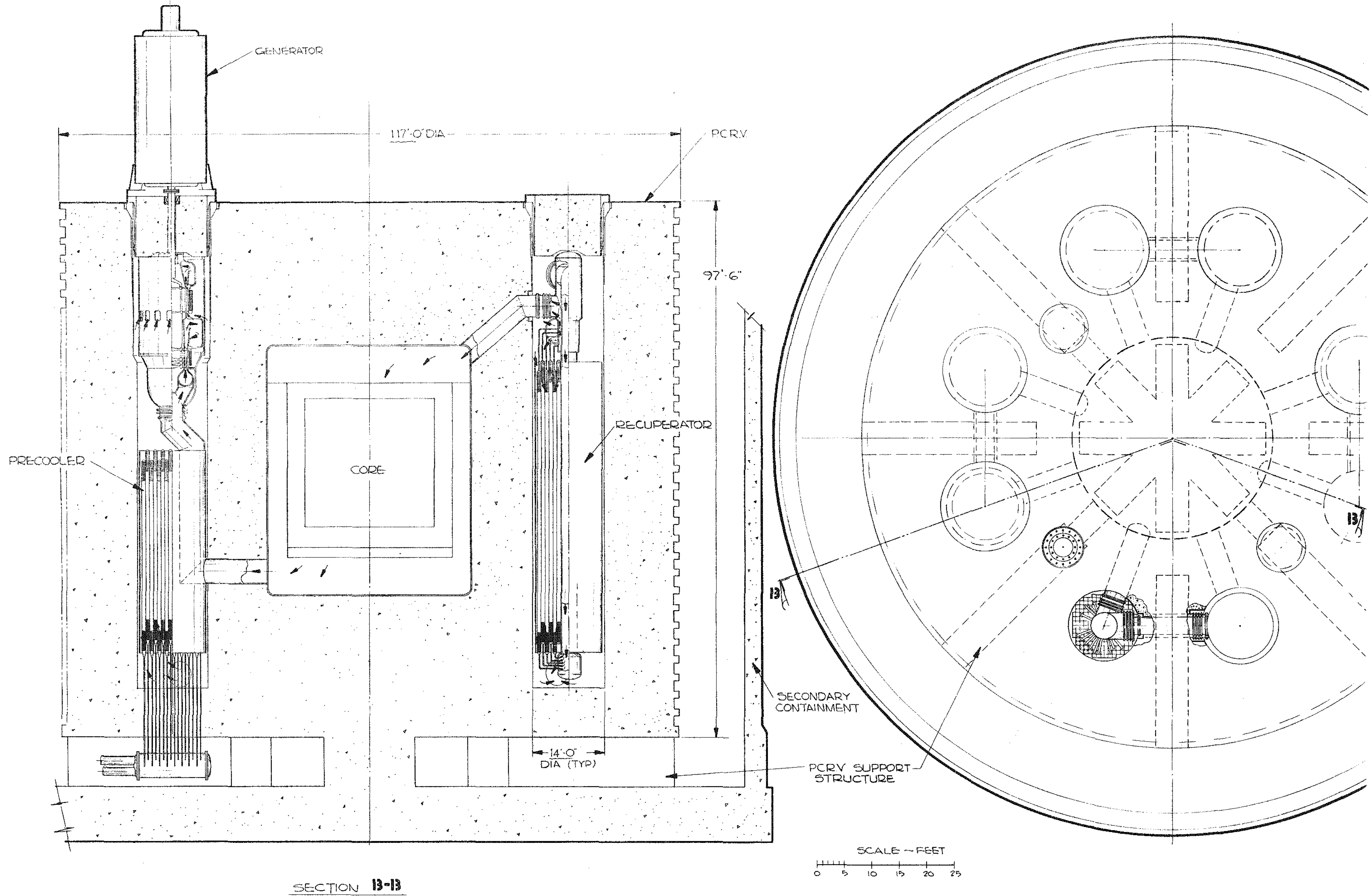



Figure 7. Plant Configuration 2. Showing Single-Shaft Turbomachinery and Vertical Generator.

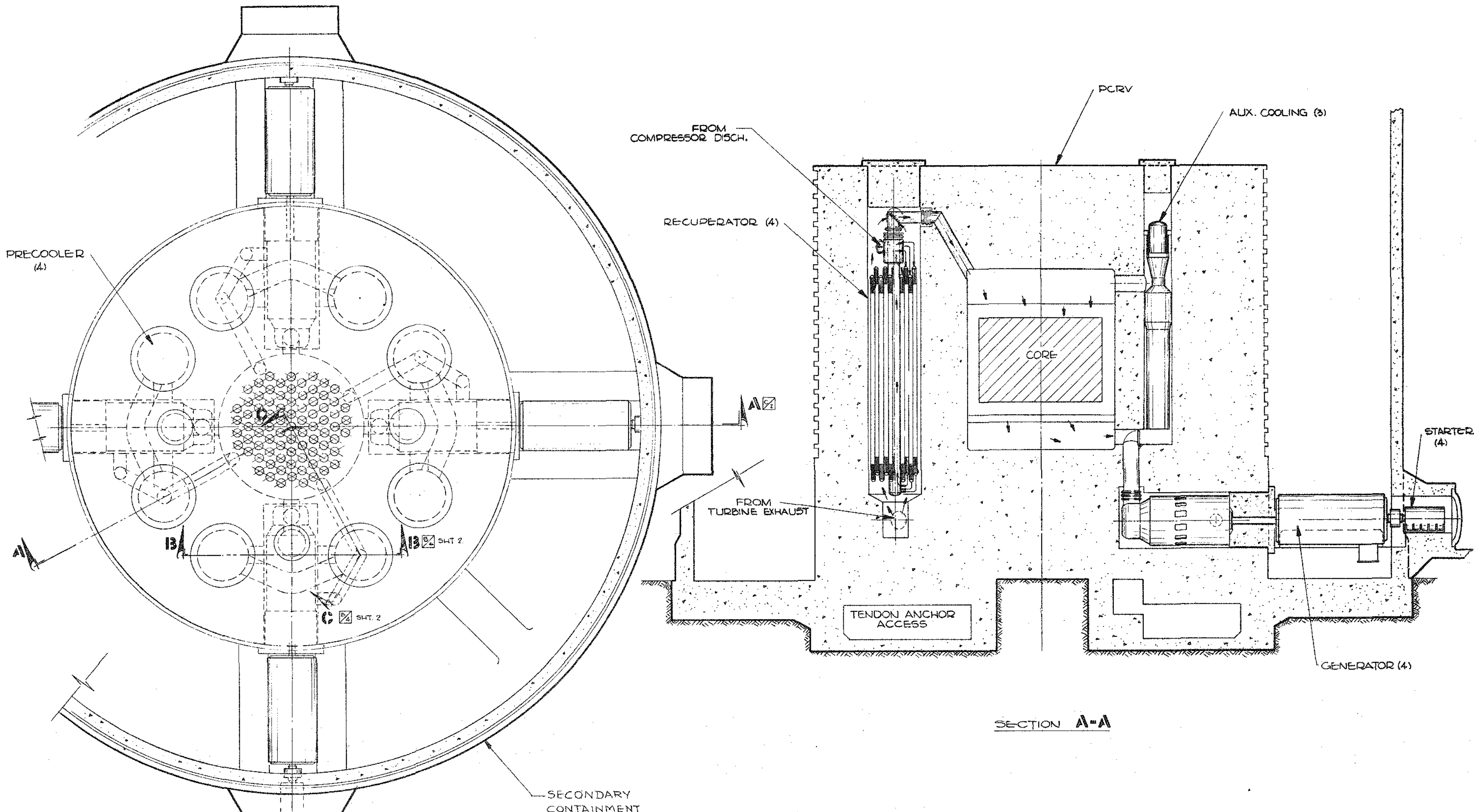



Figure 8. Plant Configuration 3 Showing Single-Shaft Turbomachinery and Horizontal Generator.

From the compressor the high pressure helium flows up through a long vertical duct in the vessel and from the concentric top header flows down inside the recuperator tubes. Leaving the tubes at the bottom the heated helium flows up through the centre duct in the recuperator, and via a cross duct at the top of the vessel it enters the core cavity completing the loop.

## PCRV CONSIDERATIONS

All three power plant configurations outlined above are based on the same reactor core and related equipment. Each PCRV provides a pressure containment for the reactor with its primary and auxiliary loops and serves as a biological shield. The 37 ft diameter and 45.5 ft high central cavity contains the reactor, reactor support, and shielding. An isometric view of an integrated HTGR gas turbine embodying single-shaft turbomachinery and vertical generator (Configuration 2) is shown on Figure 9. As outlined earlier, in the gas turbine cycle which necessarily involves large ducting, the PCRV for the integrated plant eliminates the failure of large external ducting as potential cause of rapid depressurization.

The PCRV is a vertical cylindrical vessel that has incorporated in it the steel liners for the internal cavities and their interconnecting ducts. Vertical prestressing is by linear tendons and radial prestressing is generally by circumferential wire winding. Vertical and radial reinforcement of the concrete is provided by reinforcing steel.

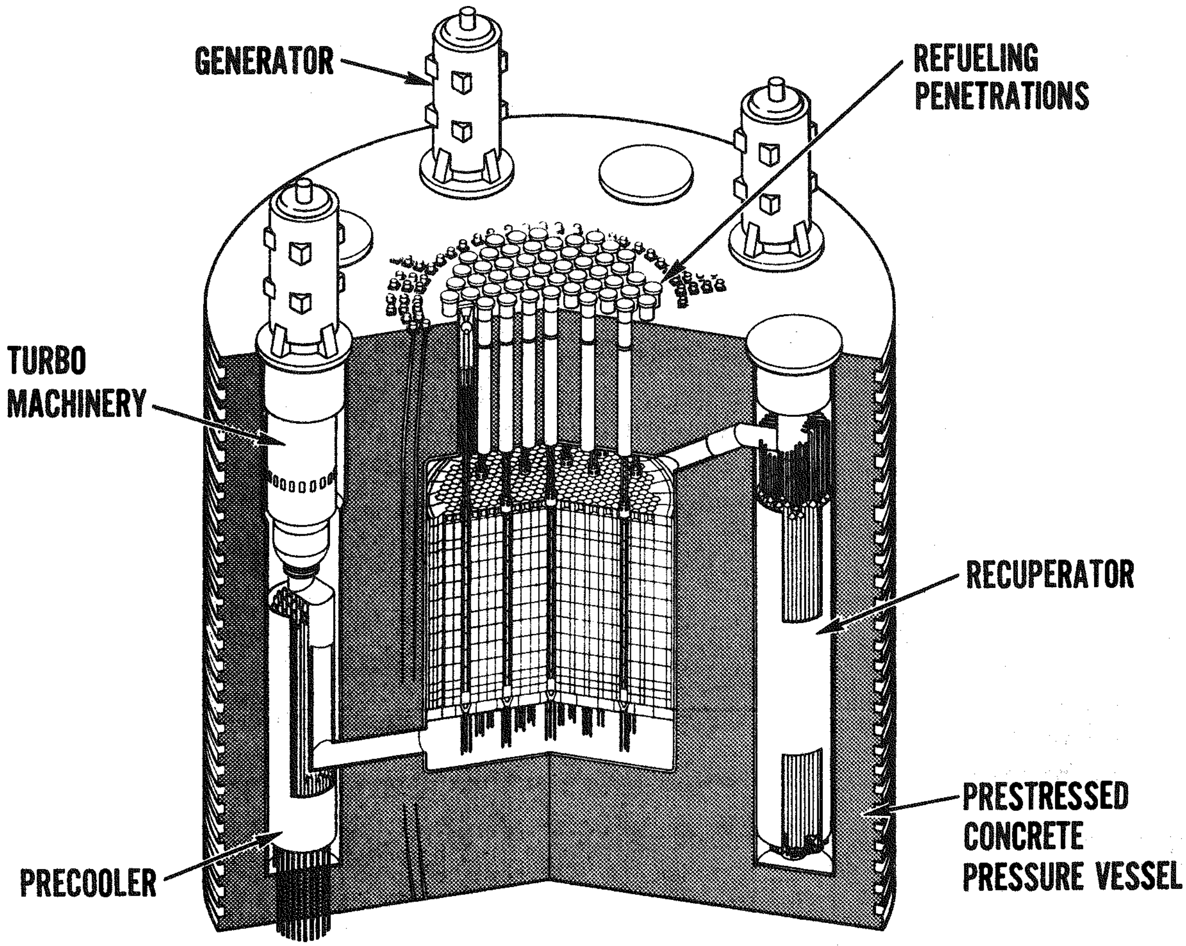
The HTGR gas turbine plant will have a secondary containment building which completely encloses the PCRV. The secondary containment together with the PCRV incorporate safety features which limit loss of primary coolant, and missile damage in the event of failures in the turbomachinery, shaft seals, generator, heat exchangers or PCRV primary closures.

## TURBOMACHINERY DESIGN CONSIDERATIONS

From the turbomachinery standpoint the use of a low molecular weight fluid such as helium implies a large number of stages, but is it fortunate that the optimization of a highly recuperated closed-cycle system gives a relatively low pressure ratio, (as shown in Figure 3) and hence the number of compressor and turbine stages are comparable with existing open-cycle industrial gas turbines. Many of the aerothermodynamic and dynamic analytical and design techniques developed extensively over the years for open cycle gas turbines are directly applicable to the closed-cycle helium turbomachinery in the HTGR gas turbine, and some aspects of the compressor and turbine preliminary designs are briefly outlined below.

### Compressor Aerodynamic Design.

A series of aerothermodynamic analyses were carried out for the compressor to establish size, geometries; and performance in support of the overall plant




LC98802

Figure 9. Integrated HTGR Gas Turbine Power Plant Embodying Single-Shaft Turbomachinery and Vertical Generator

study. For the specified design conditions the required number of stages was determined and a complete blading design performed, including annulus dimensions, number of blades, amount of twist in the blades, blade chords and stagger angles (which enable a compressor bladed length to be determined) the blade profile, and blade weight. Analytical techniques developed for helium axial compressor design for current HTGR steam plant were adopted.

In the case of the single-shaft design, the compressor rotational speed is, of course, fixed at the generator synchronous speed of 3600 rpm. Turbine stress criteria established the rotational speed of the split-shaft design at 5058 rpm. Because of the high gas bending loads associated with operation in very dense helium, the blade chords tend to be large; and with relatively small blade heights, the resultant aspect ratios (blade height/chord) are small. The high sonic velocity of helium is an advantage since it removes all Mach number limitations from the design and allows blade sections to be thickened.

From Table 2 it can be seen that for both single and split-shaft designs, 16 axial stages were selected. While a small gain in efficiency could be realized by increasing the number of stages a penalty would have to be paid in compressor length, and in the early phase of the program efforts were directed towards minimizing the length of the turbomachinery. With 16 stages and an axial velocity of 664 ft/sec the maximum diffusion factor (aerodynamic loading factor) is on the order of 0.40, which is acceptable for conservatively designed industrial gas turbines and should give a satisfactory surge margin.

Helium compressors for closed-cycle gas turbines are characterized by small blade heights, high hub-to-tip ratios, and low aspect ratios. In this type of machine end wall losses become significant and careful mechanical design is necessary to minimize tip clearances. While the end wall effects have an adverse influence on efficiency two factors that will partically offset the tendency for lower efficiencies are the very high operating Reynolds number and very low Mach number.

## Turbine Aerodynamic Design

A common design philosophy was used for the single-shaft turbine and the two turbines in the split-shaft machine. The pitch-line diameters were chosen to give as nearly optimum blade speed as possible. The axial gas velocity which determines the turbine annulus area and hence the blade height, was chosen to be 850 ft/sec for the three turbine designs. The turbine blade centrifugal stress (for a given geometry) is proportional to the $rpm^2$x annulus area, and as mentioned in the previous section, in the case of the split-shaft machine, the compressor rotational speed is limited by the turbine blade stress. The preliminary turbine designs outlined in this paper have been based on the use of existing nickel-base alloys that have been used extensively in open cycle gas turbines. The use of advanced materials (molybdenum-base alloys for example) for the turbine may be desirable at some time in the future, when reactor outlet temperatures are increased above the present day values.

TABLE 2

DETAILS OF 16-STAGE COMPRESSOR DESIGNS FOR HTGR GAS TURBINE PLANT

| Compressor | Single-Shaft Design(3600rpm) | | | | Split-Shaft Design (5058 rpm) | | | |
|---|---|---|---|---|---|---|---|---|
| Stage | First | | Last | | First | | Last | |
| Radial Position | Root | Tip | Root | Tip | Root | Tip | Root | Tip |
| Diameter, in | 68.70 | 76.00 | 68.7 | 73.50 | 48.90 | 58.70 | 48.90 | 55.00 |
| Hub/Tip Ratio | 0.904 | | 0.935 | | 0.833 | | 0.882 | |
| Axial Velocity, ft/sec | 664.0 | | 664.0 | | 664.0 | | 664.0 | |
| Blade Speed, ft/sec | 1080.0 | 1194.0 | 1080.0 | 1155.0 | 1080.0 | 1296.0 | 1080.0 | 1225.0 |
| Rotor Blade Chord, in | 3.038 | 2.613 | 2.480 | 2.241 | 4.219 | 3.209 | 3.467 | 2.869 |
| Rotor Solidity | 1.333 | 1.037 | 1.333 | 1.125 | 1.333 | 0.844 | 1.333 | 0.973 |
| Blade Height, in. | 3.65 | | 2.40 | | 4.90 | | 3.30 | |
| Rotor Aspect Ratio | 1.30 | | 1.02 | | 1.34 | | 1.05 | |
| Rotor Mach Number | 0.363 | 0.391 | 0.307 | 0.322 | 0.363 | 0.417 | 0.307 | 0.337 |
| Rotor Diffusion Factor | 0.417 | 0.396 | 0.416 | 0.401 | 0.416 | 0.377 | 0.415 | 0.388 |
| Reynolds Number $\times 10^6$ | 5.53 | 5.26 | 5.62 | 5.44 | 7.68 | 7.00 | 7.86 | 7.38 |
| Number Rotor Blades | 93 | | 116 | | 47 | | 59 | |
| Number Stators | 116 | | 145 | | 61 | | 76 | |
| Rotor Bending Stress, psi | 10,000 | | 10,000 | | 10,000 | | 10,000 | |
| Stator Bending Stress, psi | 10,000(30,000)* | | 10,000(30,000)* | | 10,000(30,000)* | | 10,000(30,000)* | |
| Rotor Centrifugal Stress, psi | 8,095 | | 5,495 | | 15,356 | | 10,730 | |
| Compressor Bladed Length, ins | 71.60 (57.20)* | | | | 95.20 (75.70)* | | | |
| Adiabatic Efficiency | 0.884 | | | | 0.888 | | | |

*Numbers in parentheses show possible reduction in length with higher allowable stator bending stress.

Details of the turbine preliminary designs for the HTGR gas turbine plant are given in Table 3. It can be seen from this table, that for reasons similar to those affecting the compressor design the turbine for closed-cycle helium turbomachinery is characterized by short blade heights, high hub-to-tip ratios, and relatively small aspect ratios. For the single shaft machine, with a net power output of 275 MW(e), a very compact turbine with a tip diameter of less than 80 ins. and a bladed rotor length of just over 50 ins is practical and is modest in size compared with an equivalent air-breathing turbine, since the enthalpy drop in the helium turbine is many times greater.

## GENERATOR CONSIDERATIONS

In the configuration studies the generator requirements called for 3-phase, 60-cycle, 3600-rpm generators for direct connection to the power producing gas turbines, having output capability of 275 MW(e). Since the generator and the turbomachinery are connected by a common shaft, they have corresponding requirements for installation with regard to stiffness and alignment. The generator and gas turbine are connected by a coupling which is disconnected when the generator is to be moved. The realignment of the generator on replacement over the turbine (in the case of the vertical generator) must be facilitated by the provision of locating registers and keys on dowels, and suitable hold-down fastenings.

For the plant configurations with horizontal turbomachinery a conventional generator would be used. The generators would be mounted on supports integral with the PCRV and driven from the turbines by shafts which penetrate the turbomachinery cavity closures.

For the single-shaft vertical turbomachinery configuration a vertical generator mounted on the top of the primary reactor vessel is necessary. While this represents a significant departure from existing steam turbine practice, the three turbomachinery and generator companies participating in the initial phase of the program with GGA, independently established the feasibility of the vertical generator.

## TURBOMACHINERY ARRANGEMENT

A simplified view of the single-shaft turbomachinery module is shown in Figure 10. In general the arrangement of the turbomachinery is similar to that of the compressor-turbine unit of the split-shaft turbomachinery except that the turbine has seven stages. With an overall length in the order of 30 ft, a diameter of 12 ft, and an approximate weight of 80 tons, the net power output from this unit is 275 MW(e). An idea of the gross output of the turbine is realized when considering that the compressor power required is in the order of 350 MW(e).

The compressor-turbine rotating assembly is of bolted-up construction. The compressor turbine rotors consist of solid discs held together with tie-bolts and located in accurate alignment by Curvic couplings. The main load-bearing outer

TABLE 3

DETAILS OF TURBINE DESIGNS FOR HTGR GAS TURBINE PLANT

| Machine Configuration | Single-Shaft | | Split-Shaft | | | |
|---|---|---|---|---|---|---|
| Turbine | Single-Shaft | | Compressor Drive | | Power | |
| RPM | 3600 | | 5058 | | 3600 | |
| Number of Stages | 7 | | 4 | | 3 | |
| Stage | First | Last | First | Last | First | Last |
| Tip Diameter, ins. | 74.30 | 78.70 | 57.35 | 60.22 | 76.80 | 78.50 |
| Root Diameter, ins. | 65.10 | 65.10 | 45.36 | 45.36 | 65.10 | 65.10 |
| Hub/Tip Ratio | 0.876 | 0.827 | 0.790 | 0.753 | 0.848 | 0.829 |
| Blade Height, ins. | 4.697 | 6.887 | 5.995 | 7.333 | 5.85 | 6.70 |
| Wheel Rim Speed, ft/sec | 1022.6 | 1022.6 | 1001.0 | 1001.0 | 1022.6 | 1022.6 |
| Blade Tip Speed, ft/sec | 1167.0 | 1236.0 | 1266.0 | 1329.0 | 1206.0 | 1233.0 |
| Gas Axial Velocity, ft/sec | 850.0 | 850.0 | 850.0 | 850.0 | 850.0 | 850.0 |
| Rotor Chord Length, ins. | 2.77 | 3.29 | 3.74 | 3.97 | 3.21 | 3.46 |
| Rotor Solidity | 1.72 | 1.72 | 1.81 | 1.81 | 1.742 | 1.742 |
| Rotor Aspect Ratio | 1.696 | 2.093 | 1.603 | 1.846 | 1.822 | 1.936 |
| Number of Rotor Blades | 127 | 107 | 69 | 65 | 111 | 103 |
| Number of Stator Vanes | 65 | 57 | 39 | 37 | 59 | 56 |
| Rotor Bending Stress, psi | 10,000 | | 10,000 | | 10,000 | |
| Rotor Centrifugal Stress, psi | 9,403 | 12,534 | 17,247 | 19,184 | 11,378 | 12,746 |
| Length Over Blading, ins | 51.50 | | 37.5 | | 26.0 | |

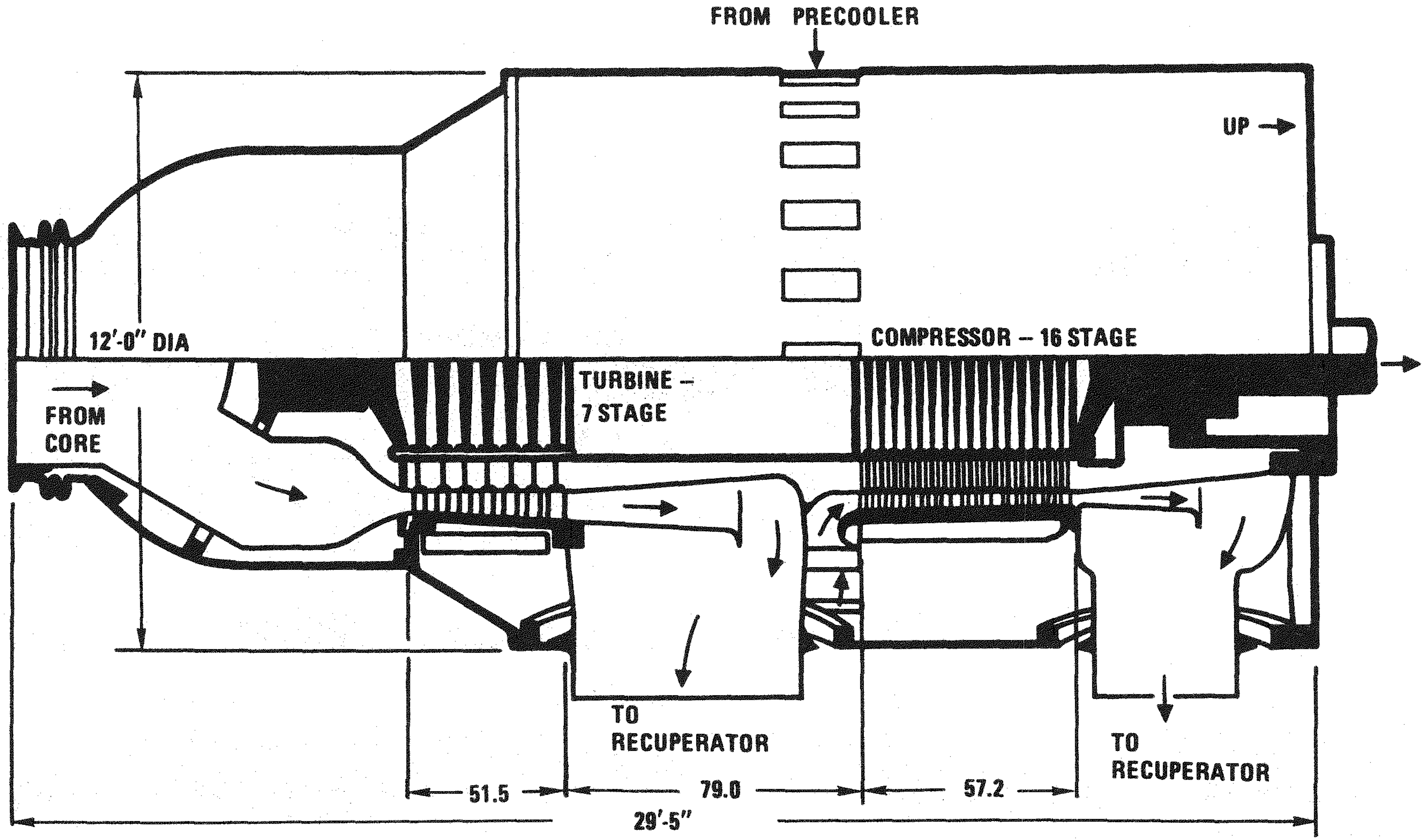



Figure 10. Simplified Cross-Section Showing Single-Shaft Turbomachinery For HTGR Gas Turbine Plant.

casing consists of a large-diameter cylindrical shell, which is always immersed in the relatively cool flow of helium from the precooler exit. The cylindrical outer casing forms the main stiffening member and maintains alignment of the bearings through the support plate inside the lower pressure vessel. The alignment of the compressor casing is maintained through its connection to the relatively cool compressor exit volute which is supported by the outer casing upper end plate. The cylindrical portion of the outer casing is large enough to contain the turbine exit volute, and holes are provided in the casing as shown at the compressor intake to permit easy passage of gas from the space around the casing into the compressor intake.

The compact nature of the helium closed-cycle turbomachinery is evident when considering that the external dimensions of a 275 MW helium gas turbine are approximately equal to those of a 25 MW air breathing open cycle industrial gas turbine.

## HEAT EXCHANGER DESIGN

The surface area of the heat exchangers in a closed-cycle gas turbine is highly influenced by the nature of the working fluid. With helium at high pressure very high heat transfer coefficients can be realized, and the surface area requirement in the recuperator, for example, is in the order of one third of that for a comparable open-cycle air breathing gas turbine. To be compatible with a four--loop turbomachinery arrangement, four recuperators and precoolers are necessary, and for the three plant configurations discussed, these are mounted vertically inside the PCRV side-wall cavities in a manner similar to that of steam generators in current HTGR steam plant. Various aspects of the precooler and recuperator preliminary designs are outlined below.

### Precooler Preliminary Design

The water outlet temperature from the precooler of 340°F, (established from precooler - dry cooling tower performance and cost trade-off studies) results in a total water flow, in the closed coolant loop for the four precoolers, on the order of 50,000 gpm.

From the structural standpoint the high gas and water pressures require a heat exchanger of tubular construction. A maintenance consideration of tube plugging in the event of a leak virtually necessitated a configuration with the cooling water flowing inside the tubes and the helium outside the tube bundle. To provide a reasonable margin for suppression of local boiling in the precooler, a water pressure of 350 psi was established. At this level the water pressure in the precooler is below the helium pressure, so that if a leak develops, leakage will be from the closed helium loop into the cooling water system.

With the high thermal effectiveness and low gas pressure loss requirements, it was found that a pure axial counterflow design embodying plain tubes gave a simple arrangement that yielded tube bundle dimensions and proportions well suited to accommodation in the cylindrical PCRV cavities. A modular type of construction was selected to facilitate manufacture, handling, installation, and to give a simple sub-headering arrangement so that each module has its own lead tube and can be plugged at the headers outside the PCRV. As can be seen from Figure 11. the precooler consists of a multiplicity of small rectangular modules mounted inside a 13 ft. diameter shroud.

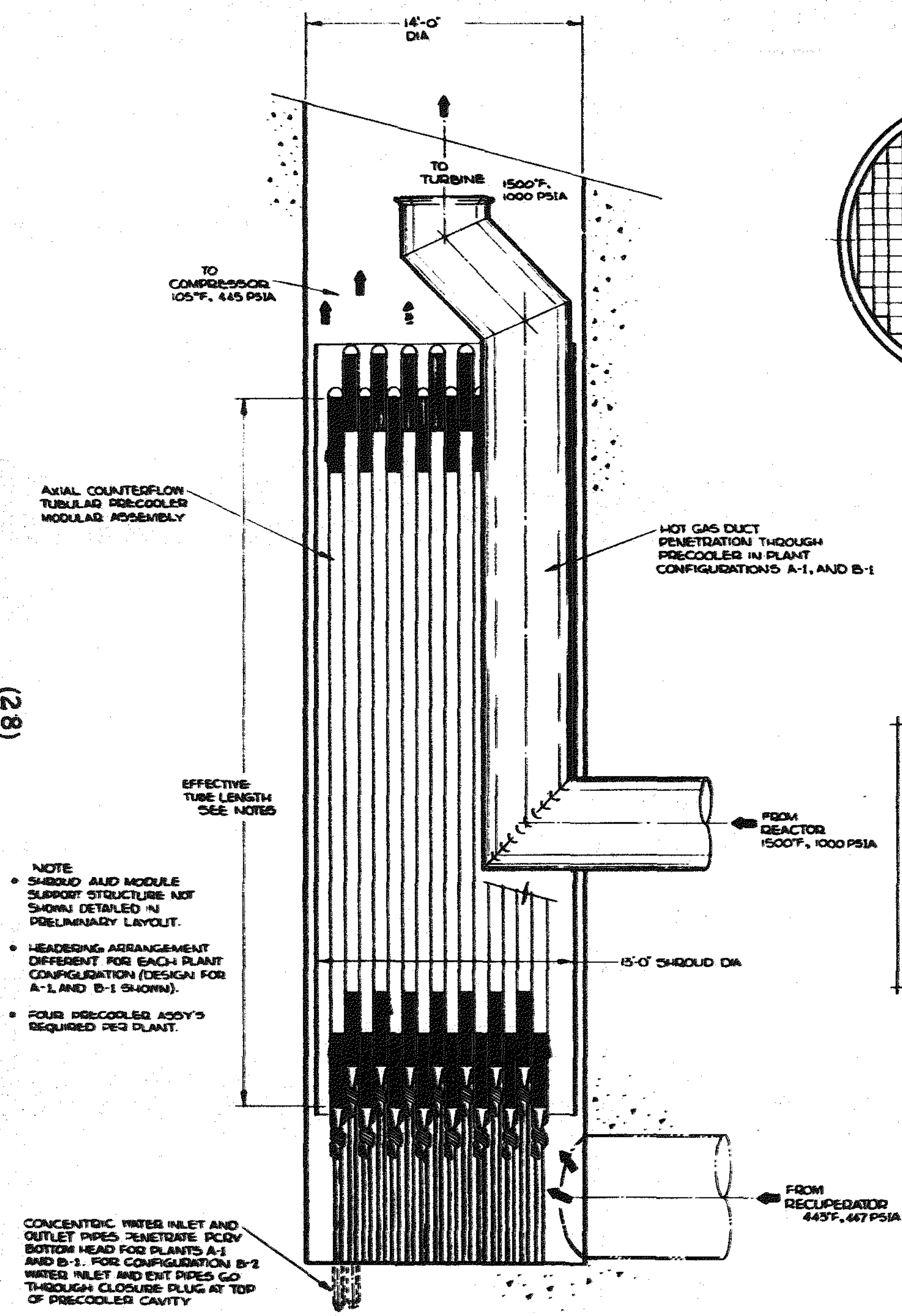
The cooling water is provided from outside the PCRV and is led in and out through a coaxial tube for each module. From the incoming water jacket water flows up through the four feeder tubes in each module and enters the tubular assembly at the top, with the water flowing downward inside the tubes to give a pure counterflow arrangement. The plain small-diameter tubes in each bundle are supported at suitable intervals by grids to avoid high stress inputs resulting from aerodynamically induced tube vibration. Details of the precooler preliminary design are given in Table 4, and it can be seen, that using a compact assembly of 3/8" O/D tubes, the effective length of the heat exchanger is 34.5 ft. Although, compared with the steam generator in the current HTGR steam plant the precooler surface area requirement is higher, the much lower tube metal temperature and lower internal pressure differential, allow extensive use of thinner and less expensive materials.

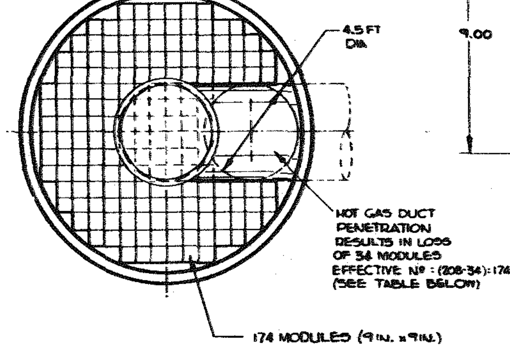
Recuperator Preliminary Design.

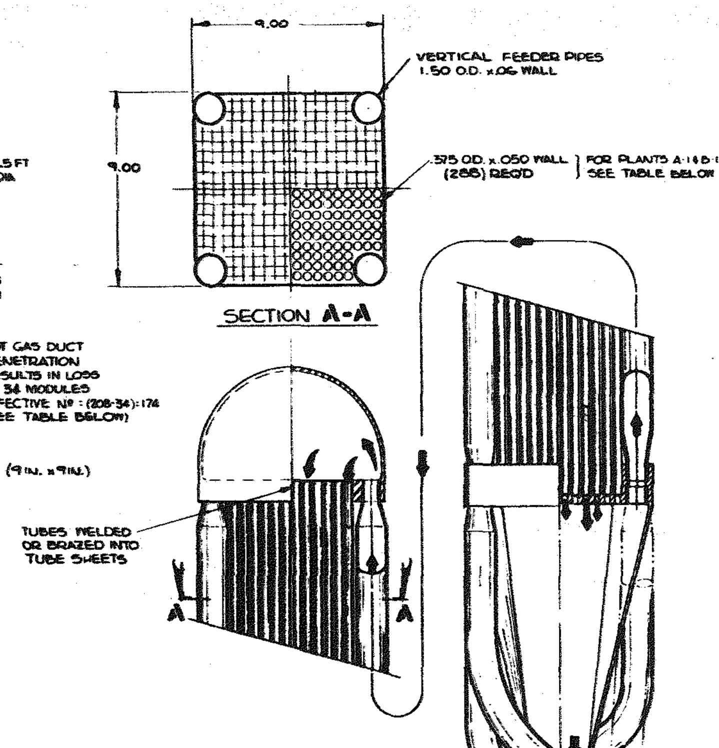
A recuperator with straight plain tubes and a pure counterflow arrangement was selected mainly because of its simplicity and the fact that the tube bundle dimensions and proportions are well suited to accommodation in the cylindrical cavities. At high levels of recuperator effectiveness a pure counterflow configuration is necessary for minimizing surface area requirements. With the high pressure gas inside the tubes a favorable relation was obtained between the high pressure and low pressure flow areas to give minimum pressure loss consistent with acceptable tube bundle dimensions.

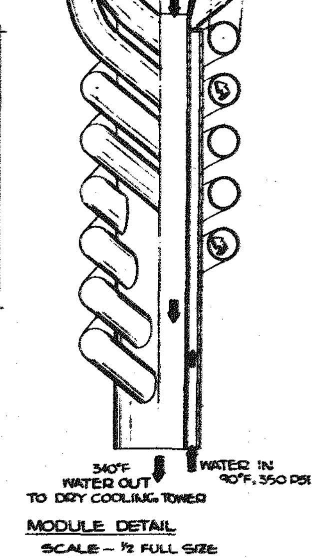
With the pure axial counterflow arrangement the heat transfer coefficient on the tube side and shell side are similar, thus there is little advantage in improving the shell-side conductance by extended surface fins on the tube unless some similar improvement can be obtained on the inside. Adding fins to the inside of a long tube with a diameter of 1/2" or less would not lead to favorable tube cost or manufacturing costs. With the pure counterflow arrangement thermal stress in the tube bundle should be minimized since the temperature gradient exists in the axial direction only.

A major problem associated with the design of large recuperators lies in achieving a satisfactory headering arrangement. As can be seen from Figure 12 an array of identical modules was chosen to simplify manufacture and installation and give an arrangement that could be readily headered with acceptable tubesheet stress levels. The design shown has a multiplicity of rectangular modules mounted inside a 13 ft diameter shroud. The low pressure entry and exit sections are longitudinally offset from bundle to bundle to provide for flow




NOTES

| PLANT CONFIGURATION | A-1 SPLIT SHAFT HORIZ. GEN. @ TOP<br>B-1 SINGLE SHAFT VERTICAL GEN. | B-2 SINGLE SHAFT HORIZ. GEN. @ BOTTOM |
|---|---|---|
| EXCHANGER TYPE | AXIAL COUNTERFLOW TUBULAR ASSEMBLY | |
| TOTAL FRONTAL AREA, $FT^2$ (4 UNITS) | 350.0 | 425.0 |
| TUBE O.D., INCHES (.050" WALL) | 0.375 | 0.500 |
| NUMBER OF TUBES (PLANT TOTAL) | 200,450 | 132,000 |
| TUBE LENGTH, FT. | 34.50 | 51.0 |
| TOTAL SURFACE AREA, $FT^2$ (4 UNITS) | 680,000 | 880,000 |
| HELIUM EFFECTIVENESS | 0.9575 | 0.9575 |
| HELIUM PRESSURE LOSS $\Delta P/P$* | .0095 | .0056 |
| NUMBER OF MODULES/UNIT | 174 | 208 |
| INDIVIDUAL MODULE WEIGHT, LB. | 2185.0 | 2500.0 |
| MODULE WEIGHT/UNIT, LB. | 380,000 | 520,000 |
| EXCHANGER UNIT WEIGHT, TON** (4 REQ'D/PLANT) | 202.0 | 270.0 |

* PRESSURE LOSS DOES NOT INCLUDE AN ALLOWANCE FOR THE TUBE SUPPORT GRIDS.
** WEIGHT ESTIMATES DO NOT INCLUDE SHROUD, SUPPORTING STRUCTURE, BELLOWS, FLANGES, WELDS, ETC.

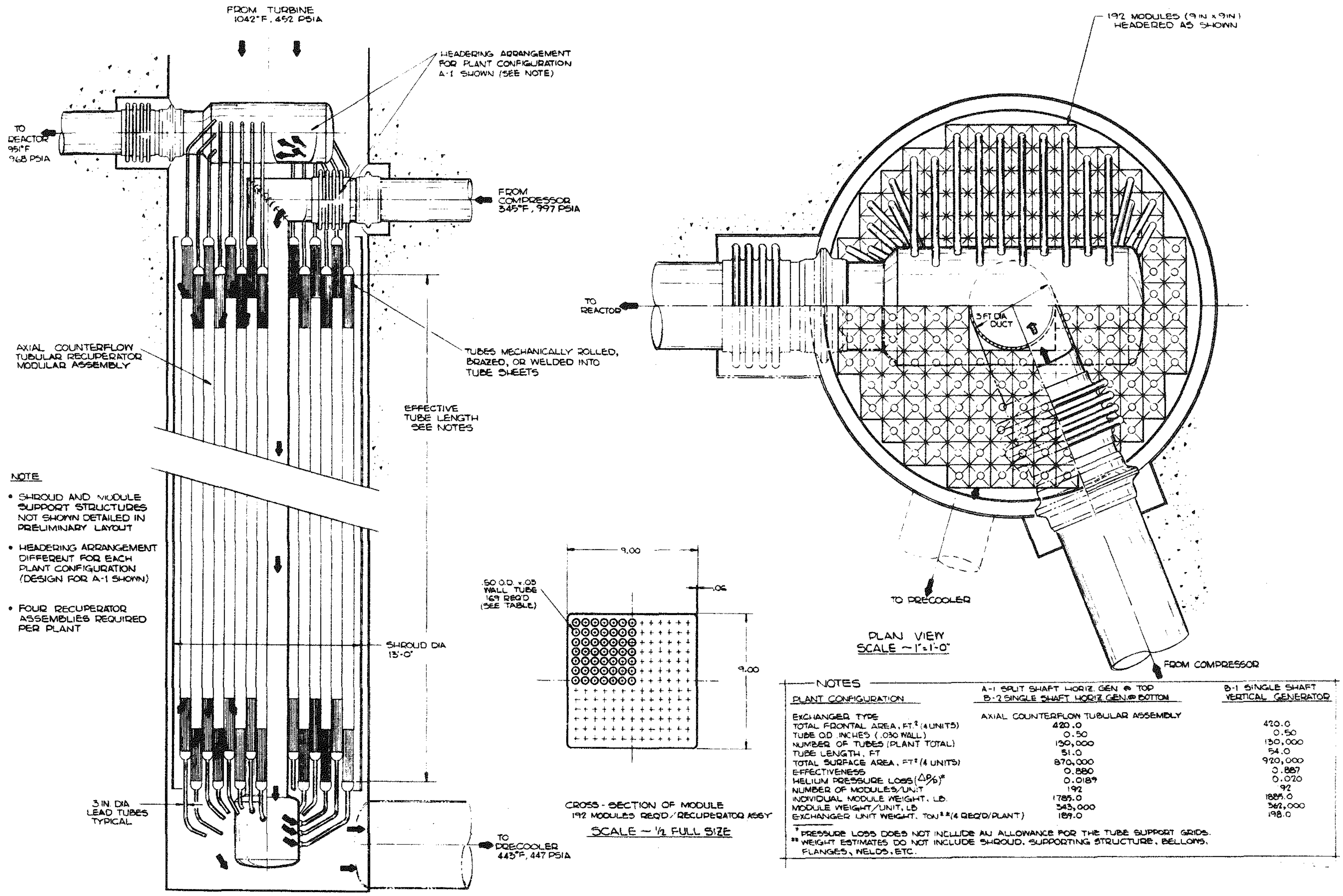
Figure 11. Tubular, Counterflow Precooler Assembly

TABLE 4

PRELIMINARY HEAT EXCHANGER DESIGNS
FOR HTGR GAS TURBINE PLANT

| Heat Exchanger | Precooler | Recuperator |
|---|---|---|
| Exchanger Type | Tubular | Tubular |
| Flow Configuration | Axial Counterflow | Axial Counterflow |
| Construction | Modular | Modular |
| Number of Exchangers per plant | 4 | 4 |
| Tube O/D, ins. | 0.375 | 0.500 |
| Wall Thickness, ins. | 0.050 | 0.030 |
| Maximum Tube Metal Temp., $^{o}F$ | 400 | 1000 |
| Material Type | A213, T-2(1/2Cr-1/2Mo) | A213, T-22(2 1/4Cr-1Mo) |
| Helium Effectiveness | 0.9575 | 0.887 |
| Helium Pressure Loss ($\Delta p/p$) | 0.010 | 0.020 |
| Total Frontal Area, $ft^2$ | 350.0 | 420.0 |
| Total Number of Tubes | 200,450 | 130,000 |
| Effective Tube Length, ft | 34.5 | 54.0 |
| Shroud Diameter, ft. | 13.0 | 13.0 |
| Total Surface Area, $ft^2$ | 680,000 | 920,000 |
| Number of Modules/Exchanger | 174 | 192 |
| Individual Module Weight, lb. | 2185.0 | 1885.0 |
| Module Weight/Exchanger, lb | 380,000 | 362,000 |
| Exchanger Unit Weight, ton* | 202 | 198 |

*Weight estimates do not include shroud, supporting structure, bellows, flanges, etc.




NOTES

| PLANT CONFIGURATION | A-1 SPLIT SHAFT HORIZ. GEN @ TOP<br>B-2 SINGLE SHAFT HORIZ. GEN. @ BOTTOM | B-1 SINGLE SHAFT VERTICAL GENERATOR |
|---|---|---|
| EXCHANGER TYPE | AXIAL COUNTERFLOW TUBULAR ASSEMBLY | |
| TOTAL FRONTAL AREA, FT.$^2$ (4 UNITS) | 420.0 | 420.0 |
| TUBE OD INCHES (.030 WALL) | 0.50 | 0.50 |
| NUMBER OF TUBES (PLANT TOTAL) | 130,000 | 130,000 |
| TUBE LENGTH, FT | 51.0 | 54.0 |
| TOTAL SURFACE AREA, FT$^2$ (4 UNITS) | 870,000 | 920,000 |
| EFFECTIVENESS | 0.880 | 0.887 |
| HELIUM PRESSURE LOSS ($\Delta P/P$)* | 0.0189 | 0.020 |
| NUMBER OF MODULES/UNIT | 192 | 92 |
| INDIVIDUAL MODULE WEIGHT, LB. | 1785.0 | 1885.0 |
| MODULE WEIGHT/UNIT, LB | 343,000 | 362,000 |
| EXCHANGER UNIT WEIGHT, TON** (4 REQ'D/PLANT) | 189.0 | 198.0 |

* PRESSURE LOSS DOES NOT INCLUDE AN ALLOWANCE FOR THE TUBE SUPPORT GRIDS.
** WEIGHT ESTIMATES DO NOT INCLUDE SHROUD, SUPPORTING STRUCTURE, BELLOWS, FLANGES, WELDS, ETC.

Figure 12. Tubular, Counterflow Recuperator Assembly

distribution. The individual modules are connected to the high pressure drum headers through bent feeder ducts. The center high pressure duct is incorporated so that the high pressure gas enters the tubes at the bottom of the assembly and flows upwards to give a pure counterflow configuration.

Although the recuperator is an important new equipment item, its operating conditions and safety implications are relatively straightforward in comparison with conventional modern steam generators. The recuperator does not from a boundary between dissimilar fluids or between separable coolant fluid loops. If a tube failure should occur in the recuperator, the helium-to-helium leak would cause only a small performance loss with no effect on plant integrity or safety. Because internal leakage is of less consequence in the recuperator than in a steam generator, the standards for tube end attachments are less exacting. Details of the recuperator preliminary design are given in Table 4, and it can be seen, that using a compact array of 1/2"O/D tubes and an available length of 54 ft., gave an effectiveness and helium pressure loss of 0.887 and 2.0 percent respectively.

## PLANT CONTROL AND SAFETY

As part of the plant preliminary design the principal objectives of the control studies were to (1) develop a basic control scheme and (2) compare the split-shaft and single-shaft turbomachinery configurations from a controls point of view. A number of alternative control systems were considered and compared on the basis of valving requirements and steady-state and transient performance. The basic control techniques that were considered included: (1) variation of helium pressure level by control of primary system inventory; (2) variation of reactor outlet temperature by reactor power level control; (3) a "compressor bypass" valve, which bypasses helium from compressor discharge to turbine discharge; (4) a "core bypass" valve, which bypasses helium from compressor discharge to turbine inlet; (5) a throttle valve at the turbine discharge; and (6) a power turbine bypass valve (applicable to the split-shaft configuration only).

Certain combinations of control valves were found to have special advantages. In particular, the combination of compressor bypass and throttle valves for the single-shaft configuration and the combination of power turbine bypass and throttle valves for the split-shaft configuration both offered the possibility of turbomachinery speed control without rapid system pressure changes. (Rates of pressure change have an important bearing on design of the PCRV thermal barrier.) However, the proper coordination of these valve combinations would require a complex control system.

Helium inventory control and reactor outlet temperature control were found to be useful in maximizing efficiency during extended part-load operation, but their response times are too great for short-term speed and load control. The prime control system must be based on fast-acting primary loop valves. The simplest control system which meets all control requirements, including loss-of-load overspeed protection, was found to be the one based on a compressor bypass valve. Two valves in parallel would be provided for each loop to meet redundancy requirements. The compressor bypass valve does give rise to rapid pressure changes during transients, but it was decided to accept this disadvantage in order to gain the advantage of a simple control system utilizing a minimum number of low temperature valves. The arrangement of the compressor bypass valve is shown in Figure 13.

The studies showed that, for control modes other than helium inventory control, a split-shaft turbomachinery configuration offers appreciably better part-load efficiency than a single-shaft configuration. On the other hand, a single-shaft turbomachinery configuration is significantly easier to control than a split-shaft configuration, due principally to the increased inertia of the single-shaft machine.

A broad review of safety aspects of the HTGR gas turbine power plant was carried out, and the areas that appeared to be of greater concern were identified for more detailed study. Significant conclusions included the following:

(1) The gas turbine plant's primary system contains metal ducts which separate regions operating at substantially different pressures, and the possible rupture of these internal ducts represents a safety problem which is not encountered in the steam cycle HTGR. Extensive analysis of internal duct ruptures and resulting pressure transients will be required.

(2) The high rotating energy of the turbomachinery and generators must be contained in the event of various postulated mechanical failures. Detailed analysis of such failures will be required.

(3) The ability to circulate helium through the reactor core with either the main compressors or the auxiliary circulators appears to offer adequate redundancy for emergency cooling.

## PLANT MAINTENANCE

A major feature of the integrated HTGR gas turbine plant is the location of the turbomachinery and heat exchangers inside cavities in the wall of the PCRV. While this arrangement has a number of advantages, it tends to limit access, and frequent scheduled or unscheduled maintenance is therefore undesirable. For this reason, an important task during the plant design phase is to ensure adequate reliability (conservative design parameters, low stress levels, etc.) to minimize required machinery maintenance. A maintenance consideration in the plant preliminary design phase has been that even the largest individual part of the system should not only be removable but also adaptable for decontamination or cleanup for limited on-site work. The shape and size of the components are such that they can be handled for off-site shipment to specialized facilities, if necessary in shielded containers.

To minimize plant down-time, a departure from the idea of on-site repair and maintenance, by substituting a spare turbomachinery module, is possible by virtue of the relatively small bulk and limited cost of multiloop components compared

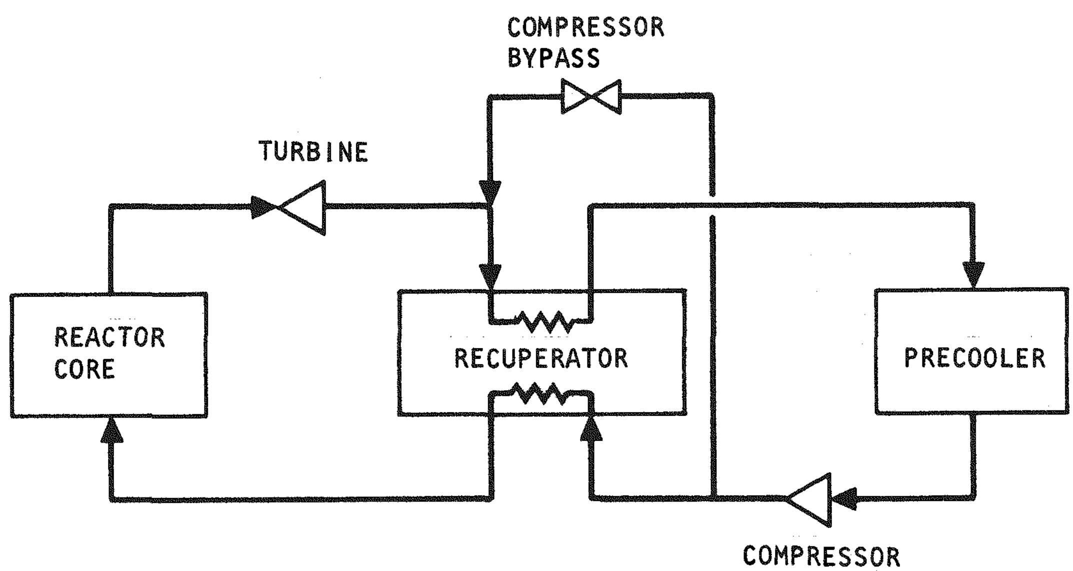



Figure 13. HTGR Gas Turbine Plant System Schematic Showing Compressor Bypass Valve Arrangement.

with existing steam plants. Maintenance must be possible, of course, to handle unforeseen events, but the design goal is to use conservative enough parameters so that, as a practical long-term objective, no maintenance should be required during the life of the equipment. It is planned to design the machinery so that remote maintenance can be performed, if ever required.

## SUMMARY


The nuclear gas turbine plant described in this paper has an output of 1100 MW(e), and on a dry cooled basis, the estimated design point plant efficiency is 36.5 percent. While continuing analytical and design studies may have a significant influence on the plant overall arrangement and component sizes, certain major design decisions were resolved during the initial phase of the program, and these are briefly outlined below.

The integrated plant arrangement (all primary circuit components and helium inventory within the PCRV) was chosen for reasons of safety and integrity. This concept is consistent with the primary circuit design of current HTGR steam plants.

A non-intercooled cycle was selected for simplicity in the primary circuit. While the incorporation of an intercooler increases the cycle efficiency by about 3 percentage points the advantage appears to be offset by the added installation complexity to the turbomachinery, ducts, and plant heat rejection system.

Several factors influenced the choice of a multiple-loop design. Independent gas turbine loops provide extra means of core cooling in emergency situations. With multiple loops the stress levels in the turbomachinery blading are reduced allowing the use of existing materials that have been extensively used in open-cycle industrial gas turbines. With four turbomachinery loops the component sizes are convenient for handling and shipping. The fossil-fired test facility for the prototype gas turbine will be more practical and less expensive for the smaller turbomachinery size associated with the multiple-loop design.

The decision in favor of a single-shaft configuration (compressor, turbine and generator on the same shaft) was based on the conclusion that this arrangement offers the advantages of design simplicity, higher reliability, easier control, and lower cost. The single-shaft machine, with its lower rotational speed (3600 rpm), has lower stress levels in the compressor and turbine blading.

Another of the major design evaluations carried out during the plant configuration studies was that concerning the orientation of the power conversion machinery. (i.e. vertical or horizontal). While this area is still being reviewed with continuing plant design studies, a decision in favor of a vertical generator orientation was made based on the following. The vertical generator configuration offers substantial cost savings, and use of a more conventional PCRV design (as being used in current HTGR steam plant) outweighed the moderate

development effort associated with the vertical generator. While this represents a significant departure from existing steam plant practice, the three turbomachinery and generator companies participating in the initial phase of the program with GGA, independently established the feasibility of the vertical generator.

The high-density working fluid used in the closed-cycle gas turbine (1000 psi helium) results in a very compact power conversion system. While the helium turbomachinery differs from air-breathing gas turbines, in that large discs and small blade heights are essentially dictated by the nature of the working fluid, many of the aerodynamic and dynamic procedures used are indentical to conventional air-breathing gas turbine practice. While no major new technology is required for the proposed helium turbomachinery, an extensive program of design, development and testing is necessary to achieve the performance and integrity levels required for long-term commercial power generation.

## REFERENCES

1) Keller, C. and Schmidt, D., "Industrial Closed-Cycle Gas Turbines for Conventional and Nuclear Fuel", ASME Paper 67-GT-10, March 1967.

2) Bell, F. R., and Koutz, S. L., "Gas Turbine HTGR and the Environment," ASME Paper 72-WA/NE-8, November, 1972.

3) Fortescue, P. "Tomorrow's Plant: Gas Turbines, Nuclear Power, Dry Cooling", Power Engineering, August, 1971.

4) Fortescue, P., Krase, J.M., Schoene, T.W., "Power Generation with the Dry-Cooled Nuclear Gas Turbine Plant", Paper presented at the IEEE Power Engineering Society Winter Meeting, New York, January 28-February 2, 1973.

5) Krase, J.M, Morse, D.C., Schoene, T.W., "The Direct-Cycle Nuclear Gas Turbine with Economical Dry Air Cooling", Paper presented at the 34th Annual Meeting of the American Power Conference, Chicago, April, 18-20, 1972.

6) Sager, P.H. and Krase, J.M., "Combined Nuclear Gas Turbine Power and Desalination Plant", Paper presented at the 7th Intersociety Energy Conversion Engineering Conference, San Diego, September 25-29, 1972.

7) Waage, J.M., Gibbons, J.P., and Sieving, K.W., "Design of 2300-MWe Twin High-Temperature Gas-Cooled Reactors for Philadelphia Electric", Proceedings of the American Power Conference, Volume 34, pp. 195-206, 1972.