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Compact Closed Cycle Brayton System for Marine Propulsion

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Sea capabilities of the United States Navy in the decades beyond 1990 will be determined largely by the research and development that was undertaken in the late 1970's. If high performance naval vessels are to realize their potential for major improvements in combat capability, it is important for appropriate R&D to be accomplished for compatible light weight compact marine propulsion powerplants. This paper summarizes some of the results from a three-year study conducted for the Office of Naval Research on the feasibility of light weight compact closed cycle gas turbine power conversion systems. In that study, critical system components were investigated and a power conversion system design concept defined and evaluated. Extended duration creep-rupture materials tests were conducted in helium at expected turbine inlet temperature for five candidate turbine blade materials. The overall results were evaluated and a representative development program defined.

INTRODUCTION

The combat capability of the U. S. Navy of the future will, to a large extent, be determined by the research and development programs that are being accomplished today. Important R&D is being accomplished which is addressed to higher performance vessels that have the potential for significant improvements in combat capability. However, if these vessels are to provide the Navy with their full potential capability, today's R&D must also be addressed to appropriate compact light weight propulsion powerplants. Furthermore, the increasing cost of fuels and the logistics difficulties that are associated with the high speeds and long distances involved in operation of these vessels will place a premium on operational propulsion plant efficiency.

The closed cycle gas turbine is a likely candidate power conversion system for many of the high performance vessels that are being considered. Such systems are now being developed for other applications and clearly have promise of suitability for naval propulsion use. The characteristics of closed cycle gas turbine systems offer significant advantages in propulsion plants where compactness, light weight and high efficiency are important criteria. Because the closed cycle gas turbine utilizes external heat addition, it is especially attractive because of its ready adaptability for use with various fuels as energy sources. Its high and relatively constant efficiency (low specific fuel consumption) over a wide range of power levels are additional characteristics making it attractive for naval ship propulsion use.

In order to determine the extent to which general appraisals of closed cycle gas turbine characteristics can be translated into realizable advanced ship propulsion units, and to provide a sound basis for subsequent exploratory engineering, the Office of Naval Research initiated a research program in 1976 (Contract No. N00014-76-C-0706). This paper summarizes some of the more important results which were obtained in this three-year Compact Closed Cycle Brayton System (CCCBS) program. Detailed results are documented in Reference 1.

The overall objective of the program was to conduct the analytical study and experimental research required to evaluate and to demonstrate feasibility of a closed Brayton cycle power conversion system for a low volume, light weight naval propulsion plant. It was also the objective to insure relevance of power conversion system study results to all candidate applications, including recognition of the various energy sources which the Navy could desire to use in the future.

In order to fulfill these objectives, the program included investigation of components that were determined to be critical to feasibility, iterative system design concept definitions, creep-rupture materials tests in helium at expected turbine inlet temperature, and overall evaluation and assessments. The results of this program therefore provide a valuable baseline of data for use by the Navy in defining the advanced powerplants which will enhance the capabilities of many of the naval vessels of the future. The results are also expected to be valuable technology inputs to considerations of closed cycle gas turbines in applications where efficiency and high power density in the power conversion system are important characteristics.

The study was accomplished by a team of Westinghouse Advanced Energy Systems Division and the Garrett Corporation's AiResearch Manufacturing Company of Arizona. AiResearch provided valuable contributions to the program in various technical areas with particular emphasis upon compact heat exchangers, gas bearings, and rotor dynamics. The program also benefited from

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technical contributions from other Westinghouse Divisions, particularly from the Combustion Turbine Systems Division and from the Research and Development Center.

REQUIREMENTS

To accomplish the objective of the program, feasibility must be evaluated in the context of a coherent set of representative top level requirements. These requirements must be representative of those which could reasonably be expected to be placed upon CCCBS plants for naval propulsion applications. The requirements must also be sufficiently stringent to support generalized conclusions regarding feasibility. Based upon these considerations, a set of top level requirements was defined and is summarized in Table 1.

TABLE 1 - TOP LEVEL REQUIREMENTS FOR CCCBS

- Reference Application - Surface Effects Ship
- Power Output per Unit - 52.2 MW (70,000 SHP)
- Specific Weight of Power Conversion Assembly - 1.2 kg/Kw (2 lb/HP)
- Lifetime - 10,000 Equivalent Full Power Hours (40,000 Operating Hours)
- Shock - MIL-S-901C (Navy)
- Heat Rejection - To Sea Water at 29°C (85°F)

In order to make the results of this study most generally useful, the scope of the power conversion system was established to include the components which are necessary to convert the input energy to output power. Interfaces with other systems which will vary from application to application were to be considered, but were not specifically included as part of the power conversion system. Thus, individual types of energy sources, loads and their transmission systems, and heat rejection systems were not included but the interfaces with those powerplant components, and the ranges of interface conditions, were considered. The most stringent interface conditions were applied as requirements in insure that the study results are valid for all likely applications.

CCCBS REFERENCE DESIGN

Based upon the plant requirements stated previously, a design concept for a reference CCCBS was developed and refined based upon detailed component and overall plant analyses. A layout view of the reference CCCBS design concept is shown in Figure 1 and the flow path for the helium working fluid is illustrated in Figure 2. Rated cycle conditions are turbine inlet temperature of 927°C (1700°F), turbine inlet pressure of 10.3 MPa (1500 psia), and a cycle pressure ratio of 3.6. The entire powertrain is encased inside a cylindrical casing approximately 5.5 m (18 ft) long by 2.33 m (7.7 ft) in diameter. The total CCCBS power conversion system weight would depend on whether one or two units are operated from one heat source. For one unit operation, the system weight would be approximately 34,000 kg (75,000 lbs); with two unit operation, the weight would increase to 42,500 kg (94,000 lbs) per unit, due to the necessity of designing the pressure casings to withstand the full system pressure. The maximum specific weight for the power conversion system is 0.8 kg/kW (1.34 lb/SHP), well under the 1.2 kg/kW (2 lb/SHP) set as a requirement.

This CCCBS design concept is the result of integration of the many design considerations that are of primary importance to naval powerplants and of the need for compactness and light weight. Throughout the concept definition, attention was given to insuring that the shock requirements can be fulfilled and to maximizing maintainability within the expected constraints of a low volume light weight powerplant. The CCCBS design concept also includes the results of research evaluations and trade studies of components which were judged to be critical. Some of the components which were examined in detail were: oil lubricated and gas bearings, various types of precoolers and intercoolers, various types of recuperators, first turbine stage blades, and others. These and other considerations give confidence in the practicality of the design concept.

The integrated assembly, illustrated in Figure 1 and discussed in detail in References 1 and 2, was conceived as a means of achieving the compactness needed for a low volume, light weight powerplant, while eliminating the need for interconnecting ducting between the various components of the system. The integrated assembly has additional benefits in maximizing resistance to shock and in minimizing design problems associated with thermal expansion, pressure retention and headering. All of the major components of the power conversion assembly are contained within a cylindrical pressure vessel made of low alloy steel. Although local pressures throughout most of the power conversion assembly are below maximum system pressure under normal operating conditions, consideration of operation with one of two parallel units shut down indicates that pressure throughout the inoperative unit may approach full system pressure as a result of leakage through shut-off valves. Therefore, the casing has been designed to withstand full system pressure.

The 18,000 rpm high pressure rotor construction consists of high pressure turbine and compressor rotors joined together by a short length of large diameter hollow drive shaft. The resulting rotor is supported at each end in gas journal bearings. Axial location of the high pressure rotor is provided by the low pressure compressor rotor gas thrust bearing through the drive shaft which connects the two rotors. The high pressure rotor employs multiple discs joined by through-bolts and curvic couplings.

The high pressure turbine rotor design has a total of five stages with cooled discs but uncooled blades. The first stage blades are made of IN100 super-alloy and are unshrouded since rotating shrouds would not confer an efficiency improvement and would increase centrifugal stresses and thus would reduce the permissible inlet temperature. The blades have extended root fixings which provide a thermal barrier between the hot airfoil section and the fir tree attachment at the disc. Cooling gas, at approximately 138°C (280°F) is directed over the extended root and disc regions to limit the temperature of the disc rim. The first stage blading was designed for 10,000 full-power hours with an inlet temperature of 927°C (1700°F) or for 40,000 hours at the reference duty cycle. The blade root and disc attachment design was defined in sufficient detail to permit a meaningful evaluation of the thermal conditions in this region.

The high pressure compressor rotor design was a total of 18 stages. The blades are attached to the discs using conventional dovetails and grooves. AISI type 403 stainless steel material has been extensively used in similar combustion turbine applications and should be appropriate in the CCCBS. The large diameter hollow drive shaft which joins the high pressure

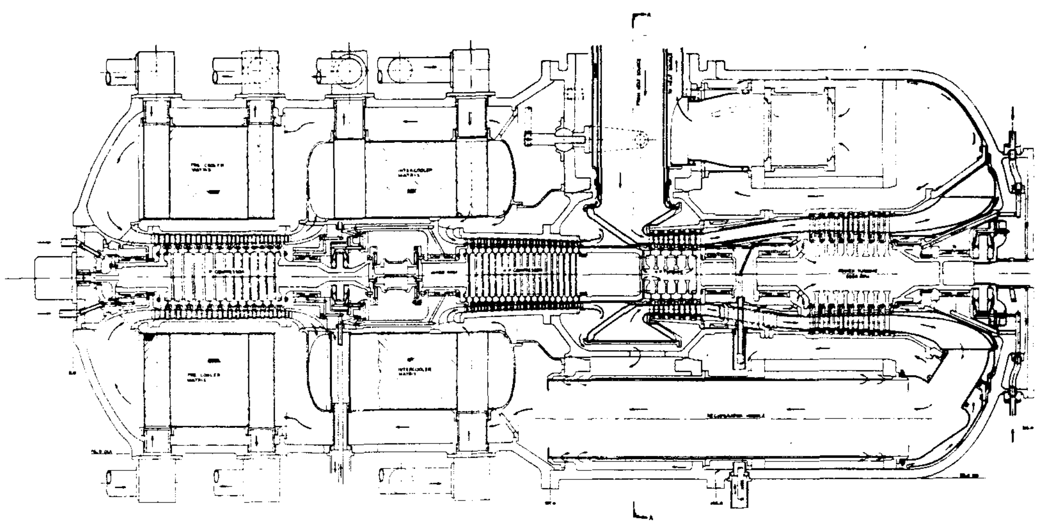

Fig. 1 - Reference CCCBS Design Concept

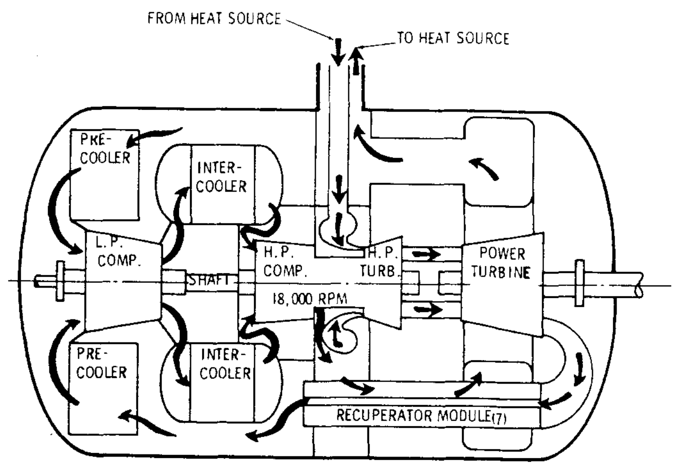

Fig. 2 - Helium Working Gas Flow Path

turbine and compressor rotors incorporates a disc-like appendage at the turbine end. Radial passages in the disc portion of the shaft allow cooling gas from the high pressure compressor outlet to flow into the inter-disc regions of the high pressure turbine rotor.

The low pressure compressor has a total of fourteen stages and is of a similar construction as the high pressure compressor. The mean blade diameter at the inlet is 409 mm (16.116 in), with a blade height of 45.5 mm (1.791 in). The rotor is supported in gas journal bearings. Axial location is provided by a gas thrust bearing at the outlet end of the low pressure compressor rotor. The low pressure compressor rotor is driven from the high pressure rotor through a short drive shaft. The drive shaft is designed to transmit axial load and to have self-aligning capability in addition to the ability to transmit torque. The drive shaft is designed to be easily disconnected to allow the removal of either the LP or HP compressor rotor. Axial rotor-to-stator gaps are 3.2 mm (0.125 in) at all locations, a value judged to be achievable with the reference thrust bearing location. The blade geometry finally selected has a tapered chord, increasing 8 percent at the root and decreasing 8 percent at the tip from the value at mean blade diameter.

A detailed analysis was performed on the first stage compressor blading. This work was largely based upon work done by Westinghouse for the Marine Gas Cooled Reactor (MGCR) program. The stage aerodynamics are based on a symmetric velocity diagram at the mean blade height and free vortex tangential velocity distribution radially. These assumptions are consistent with the results used to optimize the "MGCR" high pressure compressor.

The selection of hub to tip diameter ratio, and the maximum "wall loading factor," which places a limit on the blade and vane aerodynamic loading, is a compromise between stage efficiency, stage work, and blade and disc stresses. The following design selections were made:

Flow Coefficient at Mean	0.55
Maximum Diffusion Factor	0.40
Inlet Hub to Tip Diameter	0.80
Inlet Blockage Factor	0.98

The basic assumptions used to complete the stage study were as follows:

Shaft RPM	18,000
Fluid State Points at Inlet	
Flow	58.1 kg/s (127.9 lbs/sec)
Temperature	38^oC (100^oF)
Pressure	3.13 MPa (psia)

The design selections above result in a stage with 50 percent reaction at the mean blade height and constant axial velocity radially. To identify the stage work, the blade and vane diffusion factor was chosen at the maximum 0.4. The solidity of both blade and vane was initially chosen as 1.00, based on NACA 2-D cascade test data.

Iteration between the above aerodynamic design was done for a satisfactory overall stage configuration. To achieve this, the blade "load buildup" criteria and blade material selections were made to determine blade chords and preliminary disc shape.

Blade "Load Buildup"	1.00
Blade Material (12% chrom steel)	AISI 403

Load buildup is defined as the ratio of actual magnification factor to allowable magnifaction factor. Minimum allowable magnifaction factor is derived from a Westinghouse design curve versus blade harmonic. Actual magnification factor is defined as the ratio of material fatigue strength to blade vibratory stress times a factor of safety of 1.75. High cycle fatigue strength as a function of mean stress, in this case centrifugal plus gas bending stress, was taken from Westinghouse materials property data.

The blade which evolves from this process has a chord which is tapered 14 percent from hub to tip, and a thickness/chord which is also tapered from hub to tip. Blade gas bending and centrifugal stress result in a total steady stress of 215 MPa (31,200 psi) at the blade base section of which 134 MPa (19,400 psi) is due to centrifugal force.

The overall compressor efficiency will be a function of both the individual stage efficiency and the compressor leaving loss. The performance for the first stage predicts an efficiency of 0.920. Based on a compressor polytropic efficiency of 0.920 and a leaving loss of 4 percent, a 0.870 compressor efficiency results.

Two of the potential applications for the CCCBS plant place widely differing requirements on the low pressure turbine design. In the first of these applications, the power turbine drives a superconducting generator to provide power for an electrical transmission system. The optimum rotational speed in this case is approximately 9000 RPM. In the second application, the power turbine drives a mechanical power transmission system such as a ship's propeller reduction gear for which a speed of 3600 RPM is more suitable. Therefore, two power turbine designs were defined; one for 9000 RPM and the other for 3600 RPM application.

Figure 1 illustrates the design of the 9000 RPM power turbine. The turbine has eight stages having a mean blade diameter of 0.539 mm (21.23 in) with blade heights of 59.7 mm (2.35 in) at inlet and 86.1 mm (3.39 in). The rotor employs stacked disc construction using curvic couplings and through-bolts. Unshrouded blading is shown in Figure 1, but rotating shrouds could be employed to provide support in bending for the blading. The journal bearing stubshafts are attached to the inlet and outlet ends of its rotor by means of the through-bolts and curvic couplings. The outlet end stubshaft also carries the thrust bearing runner and the balance piston used to limit the thrust bearing maximum load.

Consideration was given to design for prevention of excessive power turbine overspeed under loss of load malfunction conditions. Preliminary analyses indicated the power turbine to be capable of withstanding a 60 percent overspeed transient, which is sufficient to withstand most malfunctions. However, the potential is dependent upon the type of energy source and also upon whether one or two power conversion assemblies are coupled to the energy source. Because the objective of this program was to evaluate and demonstrate feasibility, a modification of the Figure 1 design concept incorporating internal power turbine bypass valves was derived which could be used in those applications where a malfunction might otherwise cause excessive power turbine overspeed. This modification would also allow a second parallel unit to remain on line while the unit which suffered the loss of load is being shut off.

Figure 3 illustrates the design of the 3600 RPM power turbine. The turbine has fifteen stages having a mean blade diameter of 0.985 mm (38.76 in) with blade heights of 50.8 mm (2 in) at inlet and 71.1 mm (2.8 in) at exit. The construction is similar to that of the 9000 RPM design in its use of a stacked disc rotor with curvic couplings and through-bolts. However, the increased turbine casing diameter has made it impossible to accommodate the power turbine within the inside diameter of the recuperator high pressure outlet plenum casing in the manner of the 9000 RPM turbine. Consequently, the 3600 RPM turbine is located axially from the recuperator plenum casing in the downstream direction. The increased distance between the power turbine inlet and the high pressure turbine outlet has necessitated an annular diffuser duct of increased length connecting the two turbines. Lengthened power turbine inlet shafting and transition casings have also been required. The resulting increased length of the turbomachinery has necessitated a corresponding increase in the length of the CCCBS pressure casing.

Both gas bearings and oil lubricated bearings have been evaluated and either type can meet the CCCBS requirements. Gas bearings are inherently more compatible with closed cycle powerplants, especially those with nuclear heat sources. Gas bearings were selected for inclusion in the CCCBS design concept, although this selection does not constrain the design. Either of the two types would require essentially the same size bearing cavities. Demonstration of CCCBS feasibility, therefore, does not depend on a specific bearing type.

The rigid geometry hydrostatic concept has been adopted as a result of design and evaluations by AiResearch to limit rotor motion under shock conditions. The gas bearings originally selected to replace the oil lubricated bearings of the first CCCBS design employed a compliant foil arrangement capable of operating hydrodynamically at rated speed but employing hydrostatic augmentation at the lower speeds via a gas supply to the rotor members. The foil design, although very forgiving of thermal and mechanical distortion, is inherently flexible and would exhibit excessive deflection under Navy shock load conditions resulting in blade tip rubbing.

The journal bearings employ four pads supported from the housing through compliant diaphragm members and supplied with helium from drilled holes in the housing through metering orifices at the pads. The orifices discharge helium into a recessed area machined into the pads. The thrust bearings are similar in their use of the rigid geometry hydrostatic design but employ eight sector shaped pads. The thrust bearing stator assembly is supported from a gimbal mechanism to achieve uniform circumferential load distribution.

Sealing provisions at the bearings to limit gas leakage are confined to O-rings at the various casing joints and supply pipes. The bearing pressurizing gas, after performing its function in the bearings, bleeds directly into the primary helium flow circuits with no contamination effects.

Various types of compact gas-to-liquid (precooler and intercooler) and gas-to-gas (recuperator) heat exchangers were evaluated in the context of the CCCBS requirements. In general, the compact heat exchangers desired were determined to be within current aircraft-type heat exchanger state-of-the-art and to require no technology breakthroughs. For each heat exchanger of the CCCBS, more than one alternative was determined to be suitable, thus providing enhanced assurance of feasibility.

The reference design precooler and intercooler are similar in design and employ finned tubing arranged in a helical matrix around the turbomachinery. The tubing is headered in four equally spaced radial pipes at each

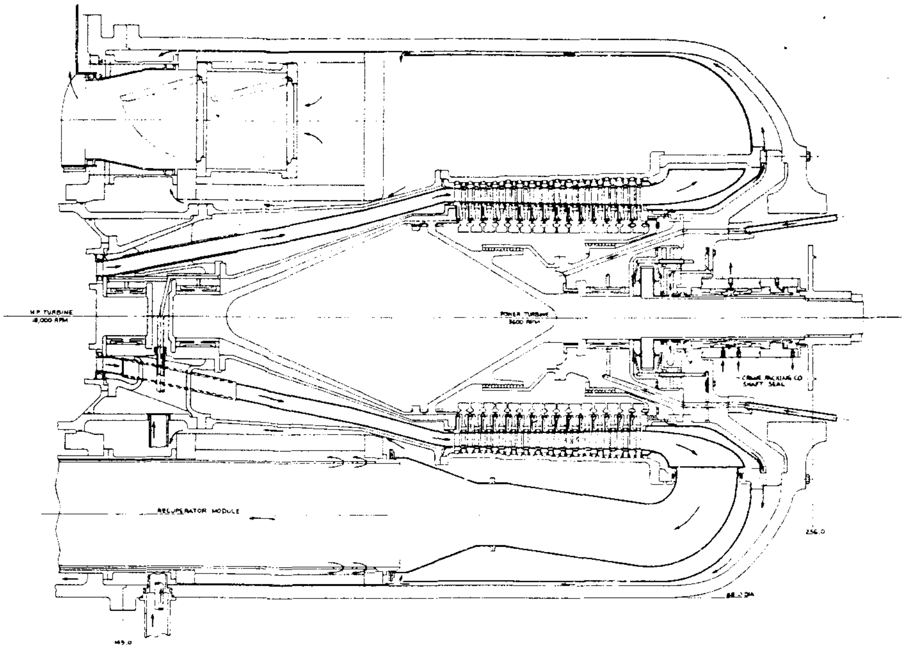

Fig. 3 - 3600 RPM Power Turbine

end of the matrix. A "four-start" arrangement of the tubing provides compatibility with the header pipes. Cooling water is piped into the radial header pipes through "bobbin" and O-ring connectors, flows through the helical tubing and leaves through the radial header pipes and connectors at the other end. The helium cycle gas flows axially across the finned tubing.

The intercooler matrix is accommodated within a cylindrical pressure shell which is terminated at each end in pressurized headers. The headers convey the helium gas from the low pressure compressor outlet into the intercooler and from the intercooler into the high pressure compressor inlet. The intercooler/header/vessel assembly is supported from a flange on the powerplant pressure vessel through a conical support member and is sealed to the turbomachinery by means of elastomer O-rings. Complete separation of the higher pressure gas in the intercooler from the lower pressure gas entering the precooler is thus achieved while permitting the turbomachinery to be freely withdrawn from the assembly through the O-ring seal interfaces. The conical support member is penetrated by a number of large diameter holes which allow the passage of the low pressure gas from the recuperator to the precooler inlet. The precooler and its headers are integrated into a similar assembly and supported from a flange on the powerplant pressure vessel.

The recuperator modules are constructed as simple thin walled cylinders containing bundles of heat transfer tubing. The flow configuration is that of a counterflow heat exchanger with the turbine exhaust helium flowing in the tubes and with the compressor exit helium flowing on the shell side. The recuperator is composed of seven modules which are approximately 415.5 mm (16.36 in) in diameter. Each module contains about 10,700 tubes with an outer diameter of 3 mm (0.120 in) and a thickness of 0.25 mm (0.010 in). The tubes are formed at their ends into a hexagonal section. The tube ends are furnace brazed together. The tube bundle is also brazed at the ends to an external cylinder and a cold end ring. Filler pieces are used to fill the gaps at the tube and cylinder interface. No tube sheet is required in this arrangement since each tube carries the local axial tension force. The tube module is free to expand and contract due to the floating end ring. O-rings are used to seal the floating cold end ring. The modules are located and sealed at the hot end by bolted flanges. This arrangement eliminates the need for tube sheets with their numerous drilled holes.

The turbomachinery is designed to be insertable into or removable from the power conversion assembly as a unit together with the generator and the rear section of the pressure vessel. In a horizontal installation, the pressure vessel rear section serves as a lifting fixture and allows the complete turbomachinery assembly to be suspended at its center of gravity from a crane or transporter device while it is removed axially from the power conversion assembly.

Alternatively, in a vertical installation, the turbomachinery can be lifted vertically from the power conversion assembly, suspended from the generator end. A small diameter hatchway, located in the deck of the ship immediately above the unit could facilitate the rapid removal and replacement of the rotating machinery.

STATE POINTS AND CONTROL

One of the primary requirements was to provide fully automatic and stable control over the full operating range up to 100 percent of full power. In addition, automatic control of the plant startup and shutdown was to be provided for. Both automatic and manual control capability were to be provided to allow for plant manageability under all forseeable circumstances. Wherever feasible, the control and protection system should make maximum practical use of diverse redundancy of sensors to maximize the system reliability.

The system state points at rated conditions are listed in Table 2. The method of control selected for the CCCBS analyses was that of a combination of temperature control and helium inventory control. In this type of control system, heat source outlet or

TABLE 2 - CCCBS STATE POINTS

	Pressure MPa	(psia)	Temperature °C	(°F)	Rate Kg/Sec	(Lb/Sec)	Power MW	Efficiency
COMPRESSOR	3.13 6.74	(454) (890)	38 151	(100) (304)	58.1	(127.9)	34.1	.87
INTERCOOLER	6.14 6.09	(890) (883)	151 38	(304) (100)	58.1	(127.9)	34.1	.98
COMPRESSOR	6.09 11.2	(883) (1625)	38 139	(100) (281)	57	(125.3)	29.7	.87
RECUPERATOR	11.16 11.12	(1619) (1613)	139 453	(281) (847)	54.6	(120.2)	88.9	.84
ENERGY SOURCE	11.01 10.43	(1597) (1512)	449 944	(840) (1730)	55.2	(121.5)	141.2	---
TURBINE	10.34 5.88	(1500) (852)	927 708	(1700) (1306)	56.4	(124.0)	63.4	.90
POWER TURBINE	5.88 3.3	(852) (478)	708 526	(1306) (978)	56.4	(124.0)	53.2	.90
RECUPERATOR	3.26 3.19	(472) (462)	511 216	(951) (420)	58.1	(127.9)	88.9	.84
PRECOOLER	3.19 3.18	(462) (455)	216 38	(420) (100)	58.1	(127.9)	53.4	.982

turbine inlet temperature is controlled as a function of the demanded power output and helium inventory in the system is adjusted as necessary to provide the demanded power output. For the CCCBS analyses, turbine inlet temperature was scheduled to linearly decrease with power demand from 927°C (1700°F) at 100 percent power to 900°C (1650°F) at 25 percent power. This temperature schedule was selected to provide maximum part-power efficiency while maintaining a turbine lifetime of 10,000 equivalent full power hours. Below 25 percent power, helium inventory was maintained constant and turbine inlet temperature varied as necessary to provide the demanded power output. It should be noted that the selection of 25 percent power for the control mode change was arbitrary - either a lower or higher power for changeover could be selected depending upon the needs of the application.

Figures 4, 5, and 6 show the steady-state plant efficiency and the system temperatures and pressures at partial powers. These figures graphically illustrate several important advantages of the closed cycle gas turbine that result from the capability to maintain the turbine inlet temperature relatively constant over a wide range of partial powers. As shown in Figure 4, the plant efficiency exhibits significantly less reduction than is characteristic of open cycle gas turbines. In fact, there is actually an increase in the plant efficiency at part power with a constant speed load due to the turbine efficiency remaining approximately constant and the heat exchanger efficiencies increasing with the reduced load. The high efficiency at partial power is particularly important for naval propulsion since the majority of the operating time is spent at part power. Thus, both unrefueled range and operating fuel costs can be significantly benefited by the use of a closed cycle gas turbine.

The relatively constant station temperatures illustrated in Figure 5 also have important beneficial effects upon both lifetime and reliability by significantly reducing the thermal cycling that components

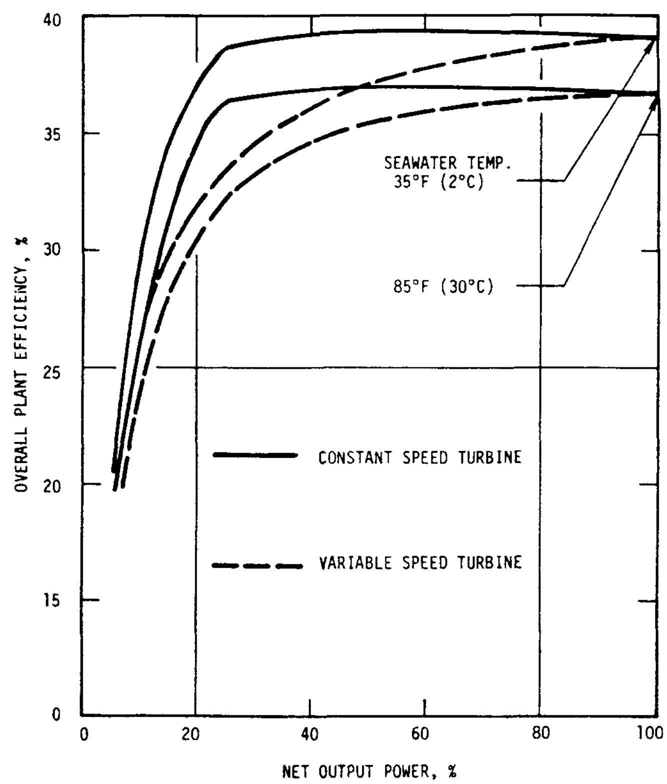

Fig. 4 - Plant Efficiency Versus Output Power

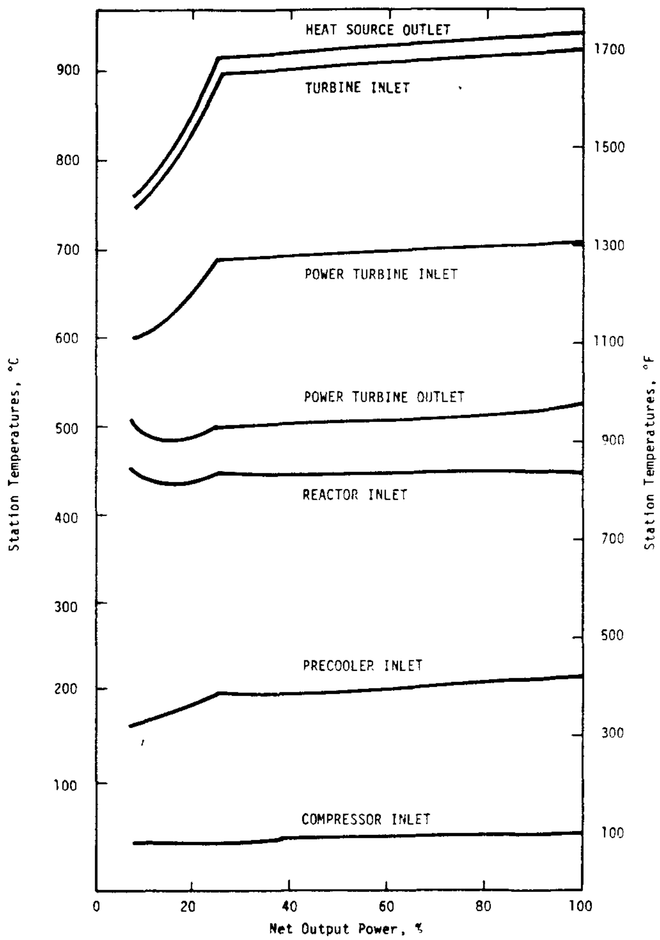

Fig. 5 - Station Temperature Versus Output Power - Constant Speed Turbine, 30°C Sea Temperature

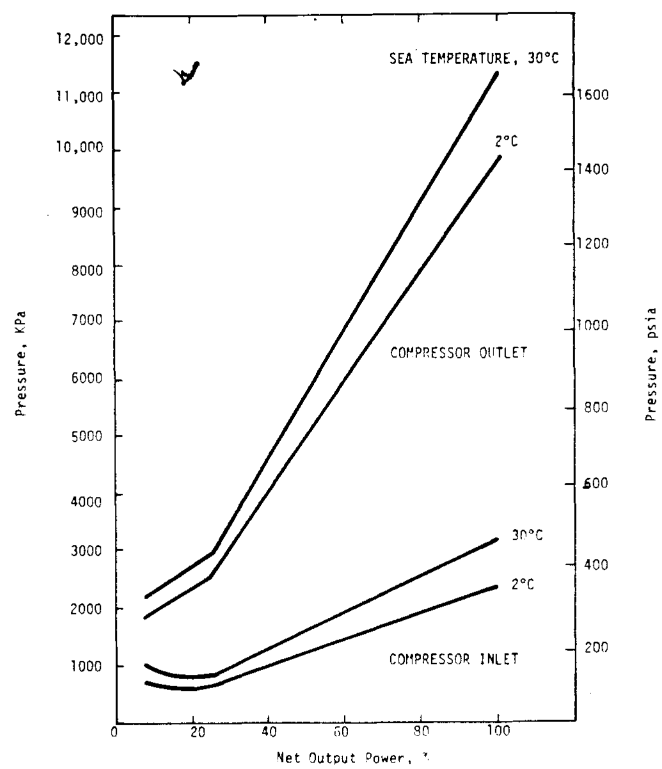

Fig. 6 - Compressor Inlet and Outlet Pressures Versus Output Power, Constant Speed Turbine

are exposed to. For instance, local temperature changes in the turbines only range from 20 to 28°C (36 to 50°F) for a power change of 75 percent.

Besides steady-state operations, the CCCBS must be operated and controlled safely during expected transients. Most common are simple throttle ramp ups and ramp downs. In addition, the system must be able to withstand malfunction transients without severe damage occurring to the equipment. A number of expected and malfunction transients were analyzed to determine their impact on the system design.

A schematic of the helium inventory control system that was used for the analyses of transients is shown in Figure 7. The necessary variations in helium inventory are accomplished by either bleed of helium from the high pressure compressor exit into a two pressure level storage system or flow from the storage system into the low pressure compressor inlet. Two pressure levels of storage, with a switchover between the two at 45 percent power, were used to minimize the helium storage volume required. Control of the heat source outlet temperature is dependent upon the type of heat source. For a fossil fuel fired heat source, the temperature is controlled by adjusting the fuel firing rate and the combustion air flow. For a gas cooled nuclear reactor, the outlet temperature is controlled by adjusting reactivity control drums or rods, thereby adjusting the reactor power.

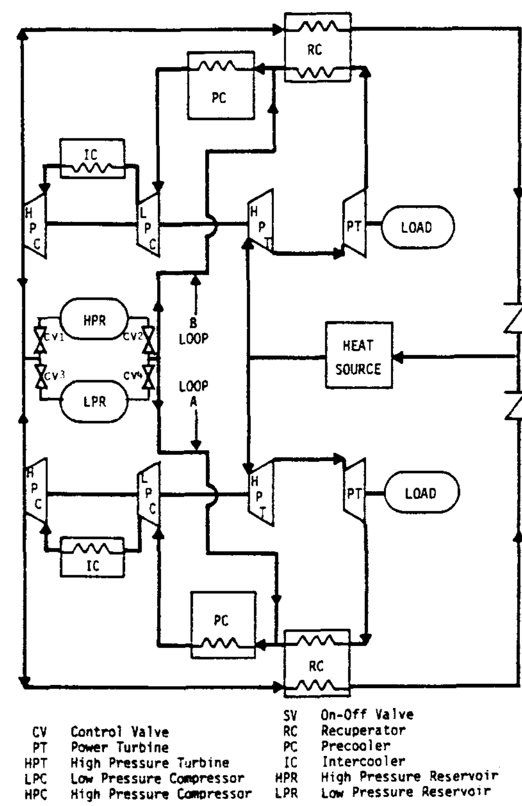

Fig. 7 - Helium Inventory Control Schematic

Normal and malfunction cases were investigated, including such as:

- Normal ramp down from 100 percent power.
- Normal ramp up from 10 percent.
- Reactor scram at full power.
- Full loss of load at full power.
- Inventory valve between compressor discharge and high pressure reservoir (CV1) fails open at full power.
- Inventory valve between high pressure reservoir and precooler inlet (CV2) fails open at 25 percent power.

The results of these analyses supported the reference design concept, and indicated that the plant could be controlled and operated using standard control methods.

MATERIALS TESTS

An important element of the CCCBS program was that of creep-rupture testing of five candidate turbine materials in helium at 927°C (1700°F). These tests were conducted to demonstrate material performance suitability through tests to confirm materials properties and to provide baseline data for the feasibility evaluations.

Historically, superalloy development has been aimed at problems associated with aviation and land gas turbines where thermal fatigue, hot corrosion, and oxidation are the primary factors which limit component life. While creep-rupture behavior is an important design consideration in such systems, hot component life is limited to a large degree of materials compatibility with combustion products, including fuel impurities such as sulfur compounds and catalytic agents (vanadium pentoxide) from fuel processing. In the case of CCCBS turbine components, creep behavior is expected to be a primary design consideration. Minor amounts of active contaminants in the inert working fluid are expected to produce surface reactions which may or may not influence creep behavior.

The objective of the CCCBS materials test program was to determine the feasibility of potential turbine materials to function under CCCBS operating conditions for the design life of the system. To achieve this objective, the creep-rupture behavior of five selected alloys were evaluated at 927°C (1700°F) in air, in static ultra-high-purity helium (UHP-He) and in a dynamic simulated CCCBS helium working fluid. Creep-rupture properties were also compared to available published data obtained from the Mechanical Properties Data Center, Traverse City, Michigan.

The alloys selected for the materials testing program are listed in Table 3. The first three alloys were selected on the basis of their commercial status and their particular applicability to the Compact Closed Brayton System requirements. Alloys 713LC, IN100, and MAR-M509 are commercially available alloys which have been used extensively in gas turbine applications over the past 10 to 15 years. Their history of metallurgical development and mechanical behavior is well documented. Thermal stability, as determined by microstructural behavior on thermal exposure at elevated temperature, is also well known and recorded. Alloy 713LC is one of a few nickel-base superalloys which does not contain cobalt as an intentional alloying addition. There are applications where cobalt is an objectionable alloying addition, and its presence is undesirable. IN100 is a "workhorse" turbine blade material designed to have a relatively low density combined with excellent creep resistance up to 1038°C (1900°F). The alloy has been successfully cast and utilized in a variety of shapes from turbine blades, vanes, and nozzles to integral wheels. Cobalt alloys are generally used as vanes, nozzles and other static hot structures because of their excellent hot corrosion (sulfidization) resistance. Cobalt alloys, however, are not generally used for rotating parts due to their lower creep and oxidation resistance compared to nickel-base alloys. MAR-M509 has good creep strength, resistance to thermal stressing, and resistance to crack propagation. The alloy can be used in the as-cast condition; no heat treatment is required. It can be successfully welded and joined both to itself and to many dissimilar metals by conventional techniques.

MA-754 is a recently developed nickel-chromium alloy produced by mechanical alloying which uniformly disperses yttrium oxide for dispersion strengthening at temperatures approaching the alloy melting point. MA-754 exhibits excellent stress-rupture properties which are equivalent or superior to cast alloys. MA-754 is currently being used as vanes in gas-turbine engines where high temperature strength is required. The superior creep-rupture properties of MA-754 provides high temperature growth potential for CCCBS turbine components.

TZM, a molybdenum-base alloy, is a refractory metal alloy which has demonstrated compatibility with inert gases at elevated temperatures and has superior creep strength at 982°C (1800°F) and above. TZM provides system growth to higher temperature capability.

Creep-rupture testing of five selected turbine materials was conducted in ultra-high purity helium and simulated CCCBS helium working fluid environment. Supplementary creep-rupture tests were also conducted in static air for alloy 713LC, IN100, MAR-M509, and MA-754. The test data generated are listed in Table 4. A total of 40,035 hours were accumulated in air, 76,583 hours in ultra-high purity helium, and 11,350 hours in a dynamic simulated CCCBS helium working fluid. Termination of the program prevented completion of the tests involving the dynamic simulated CCCBS helium working fluid environment. Test results for each alloy are described and evaluated individually in References 1 and 3. In summary, the materials tests indicated:

- Creep-rupture data obtained from the Mechanical Property Data Center for the three relatively standard gas turbine superalloys, 713LC, IN100 and MAR-M509, confirmed the dearth of published creep-rupture data that exists for materials at temperatures above 871°C (1600°F) and for test times greater than 1000 hours.
- The air test data at 927°C (1700°F) for Alloys 713LC, IN100 and MAR-M509 indicate that the materials evaluated in this program were typical of commercially available heats of the respective alloys. Air test data for MA-754 appeared to agree with the limited available data supplied by the producer of the alloy.

TABLE 3 - SELECTED ALLOYS AND NOMINAL COMPOSITIONS

ALLOY	DENSITY (lbs/in^3)	COMPOSITION (wt. %)
1. Alloy 713LC	0.289	Ni-12Cr-4.5Mo-2Cb-5.9Al-0.6Ti-0.05C
2. IN100	0.280	Ni-10Cr-15Co-3Mo-4.7Ti-5.5Al-0.9V-0.18C
3. MAR-M509	0.320	Co-23.5Cr-10Ni-7.0W-3.5Ta-0.2Ti-0.5Zr-0.6C
4. MA 754	0.300	Ni-20Cr-0.5Ti-0.3Al-0.6Y_2O_3-0.05C
5. TZM	0.368	Mo-0.5Ti-0.08Zr-0.02C

TABLE 4 - CREEP-RUPTURE DATA FOR SELECTED CCCBS MATERIALS AT 930°C (1700°F)

Material	Test Atmos.	Stress (MPa)	Stress (ksi)	Time to % Strain (Hours) 0.5%	1.0%	3.0%	Total Strain (%)	Rupture Time (Hrs)	Reduction in Area (%)	Status
Alloy 713LC	UHP He	138	(20.0)	84	192	268	6.4	269	16.7	Completed
		138	(20.0)	260	495	691	8.0	755	22.0	Completed
		124	(18.0)	128	943	1522	4.8	1527	8.4	Completed
		103	(15.0)	1060	2500	3150	5.6	3182	10.2	Completed
	Air	138	(20.0)	212	395	572	9.6	623	25.8	Completed.
		138	(20.0)	8	86	324	10.4	397	8.7	Completed
		124	(18.0)	29	119	405	9.6	476	8.3	Completed
		103	(15.0)	100	740	1870	5.6	1959	8.1	Completed
	CCCBS*	117	(17.0)	575	820	1130	8.0	1268	26.8	Completed
IN 100	UHP He	155	(22.5)	700	1170	1982	16.0	2272	17.6	Completed
		124	(18.0)	1217	-	-	-	1448	-	1
		110	(16.0)	3000	4300	7950	10.4	9452	34.5	Completed
	Air	207	(30.0)	8	70	211	6.4	271	6.7	Completed
		207	(30.0)	22	93	285	7.2	334	8.5	Completed
		172	(25.0)	1	7	490	7.2	685	4.3	Completed
		155	(22.5)	110	550	1130	7.2	1347	6.0	Completed
		124	(18.0)	1460	2890	2985	7.2	3110	12.1	Completed
		110	(16.0)	200	3800	4900	8.0	5244	16.1	Completed
	CCCBS*	124	(18.0)	1550	2000	2810	7.2	3338	23.7	Completed
		110	(16.0)	-	-	-	-	1316	-	1
		110	(16.0)	1000	2650	3920	13.5	6304	24.2	Completed
MAR M509	UHP He	103	(15.0)	2350	2440	2460	4.0	2537	0.1	Completed
		86	(12.5)	1900	5050	7900	12.8	11615	44.4	Completed
	Air	124	(18.0)	2	5	264	21.6	618	34.1	Completed
		103	(15.0)	1	570	1910	16.8	2507	19.6	Completed
		86	(12.5)	5	25	2750	7.2	4652	0.9	Completed
MA 754	UHP He	152	(22.0)	-	-	-	-	8751	-	1
		138	(20.0)	-	-	-	-	17626	-	Stopped
	Air	152	(22.0)	-	-	-	-	2100	-	2
		138	(20.0)	70	3480	15440	12.0	15712	10.4	Completed
	CCCBS*	165	(24.0)	20	100	230	5.0	289	9.2	Completed
		155	(22.5)	40	110	636	3.0	636	7.6	Completed
		155	(22.5)	100	600	1445	4.0	1445	1.9	Completed
		152	(22.0)	1	500	2640	4.4	2660	9.4	Completed
TZM	UHP He	414	(60.0)	220	425	1095	12.0	1483	98.0	Completed
		379	(55.0)	450	1010	2000	14.5	2256	98.0	Completed
		362	(52.5)	790	2000	5200	10.2	5592	76.0	Completed
		331	(48.0)	1850	4650	-	-	7818	-	Stopped

*Simulated CCCBS He Working Fluid, 400 to 500 μ atmos. H_2, 40 to 50 μ atmos. CH_4, 40 to 50 μ atmos. CO, 1 to 3 μ atmos. H_2O, CO_2 equilibrium levels.

1 Temperature Controller MaLfunctioned

2 Specimen Failed in Grips

- The ultra-high purity helium environmental creep-rupture test results indicated that all five materials were not adversely affected by the ultra-high purity helium test environment. Creep-rupture life times at 927°C (1700°F) were comparable to or exceeded life times for material tested in air. The TZM alloy being a refractory metal alloy was not tested in air due to reactivity with oxygen. TZM creep-rupture life data was comparable to extrapolated data obtained under ultra-high vacuum test conditions.

- Limited creep-rupture data obtained in a dynamic simulated helium working fluid containing 400 to 500 micro-atmospheres hydrogen, 40 - 50 micro-atmospheres carbon dioxide, 40 - 50 micro-atmospheres methane and 1 to 3 micro-atmospheres-moistures, at a test temperature of 927°C (1700°F) for three alloys, 713LC, IN100 and MA-754, indicated no effect of the test environment on material rupture life for times up to 4000 hours.

  One test was conducted for Alloy 713LC, two for IN100, and four for MA-754. Test times were less than 4000 hours, thus proof of compatibility for times greater than 10,000 hours was not obtained.

CONCLUSIONS

As a result of the analytical study and experimental research of this program, the feasibility of a closed Brayton cycle power conversion system for a low volume, light weight naval propulsion powerplant has been shown. The overall CCCBS characteristics, dimensions and weights are shown in Tables 5 and 6.

The Compact Closed Cycle Brayton System (CCCBS) program has included derivation of the most stringent representative requirements for the CCCBS power conversion system, consideration of the interfaces with the ship and with other powerplant components which were outside the scope of the system under study, investigation and evaluation of the components which are most critical to feasibility, iterative definition of a reference 70,000 HP CCCBS design concept, extensive creep-rupture tests of candidate turbine materials above the expected turbine inlet material temperature, and overall evaluations and assessments. The results have shown the feasibility and attractive characteristics of a CCCBS and have indicated that no high risk developments or technology breakthroughs are needed for the CCCBS power conversion system.

REFERENCES

1. "Compact Closed Cycle Brayton System Feasibility Study-Final Report;" Contract No. N00014-76-C-0706; Westinghouse Advanced Energy Systems Division; WAES-TNR-237; August 1979.

2. "A Compact Closed Cycle Gas Turbine for Marine Propulsion;" F. R. Spurrier; ASME Paper 79-GT-62; March 1979.

3. "Creep-Rupture Behavior of Selected Turbine Materials in Air, Ultra-High Purity Helium, and Simulated Closed Brayton Helium Working Fluid;" R. L. Ammon, L. R. Eisenstatt and G. O. Yatsko; ASME Paper No. 80-GT-173; March 1980.

TABLE 5 - TURBOMACHINERY

DELIVERED POWER	52 MW (70,000 hp)
OVERALL COMPRESSOR PRESSURE RATIO	3.6
FLOW RATE	58 kg/sec (128 lb/sec)
TURBINE INLET TEMPERATURE	927^{o}C (1700^{o}F)
COMPRESSOR EXIT PRESSURE	11.2 MPa (1625 psia)
COMPRESSOR EFFICIENCY	87%
TURBINE EFFICIENCY	90%
HP TURBINE BLADE MEAN DIAMETER	38.8/cm (15.28 in)
MEAN TURBINE WHEEL SPEED	366 M/sec (1200 ft/sec)
NUMBER OF STAGES:	
LOW PRESSURE COMPRESSOR	14
HIGH PRESSURE COMPRESSOR	18
HIGH PRESSURE TURBINE	5
LOW PRESSURE TURBINE	8 (9000 rpm) 15 (3600 rpm)

TABLE 6 - POWERPLANT SIZE AND WEIGHT

	9000 RPM POWER TURBINE	9000 RPM POWER TURBINE
OUTSIDE DIAMETER	2.33 M (92 ins)	2.33 M (92 ins)
OVERALL LENGTH	5.49 M (216 ins.)	6.5 M (256 ins)
WEIGHT		
- TWO UNIT OPERATION	42,616 Kg (93,755 lb)	50,347 Kg (110,763 lb)
- SINGLE UNIT OPERATION	34,138 Kg (75,103 lb)	40,979 Kg (90,153 lb)
SPECIFIC WEIGHT		
- TWO UNIT OPERATION	0.82 Kg/kW (1.34 lb/SHP)	0.96 Kg/kW (1.58 lb/SHP)
- SINGLE UNIT OPERATION	0.65 Kg/kW (1.07 lb/SHP)	0.79 Kg/kW (1.288 lb/SHP)