ENGINEERING RESEARCH INSTITUTE

UNIVERSITY OF MICHIGAN

NUCLEAR REACTOR AND CLOSED-CYCLE GAS TURBINE POWER PRODUCTION

A PROPOSAL FOR A 50 MW NUCLEAR REACTOR DEVELOPMENT PROJECT

By

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1.0 ABSTRACT

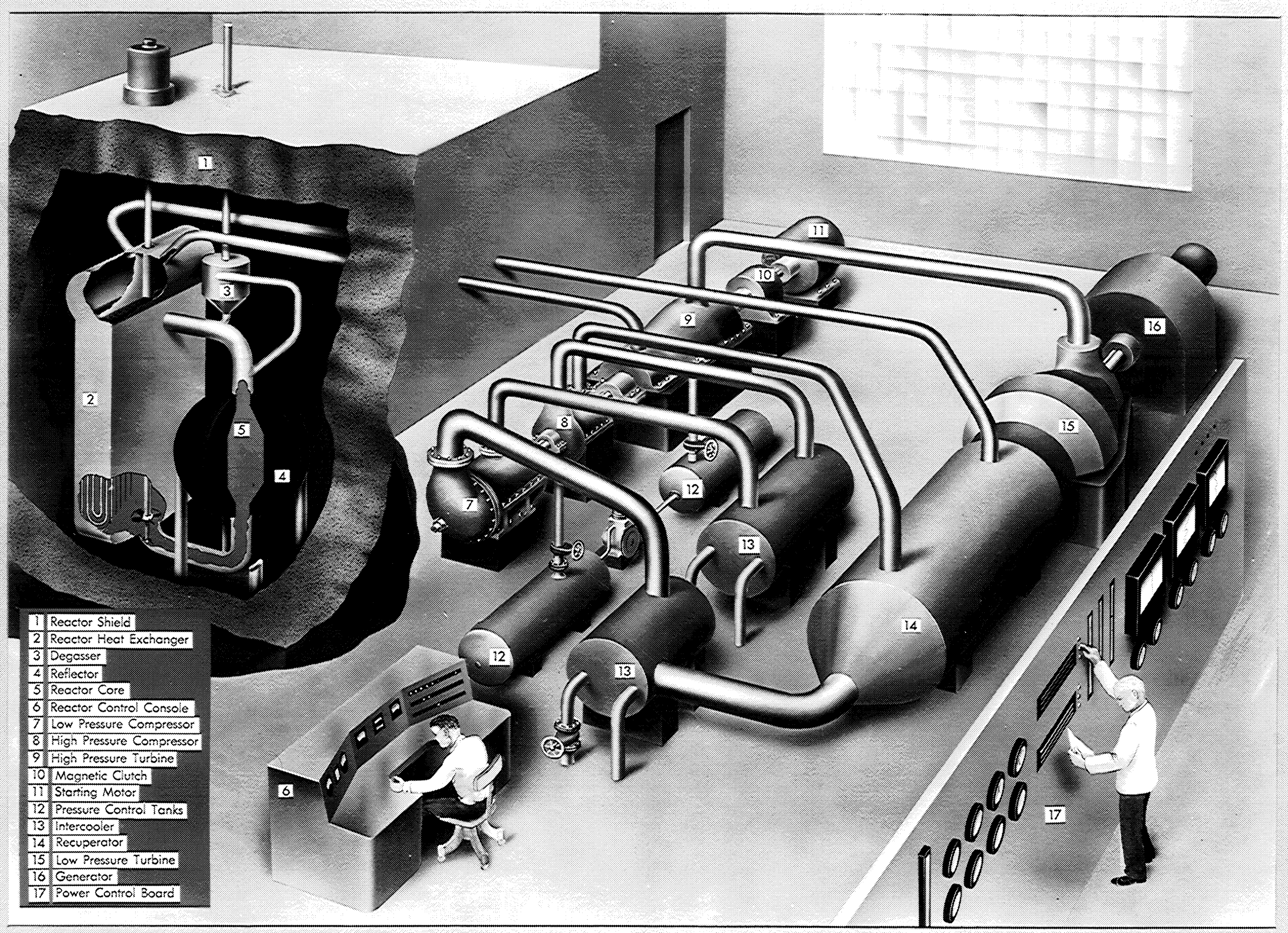
One of the most promising technological developments for production of power from a nuclear reactor is a high-temperature reactor operating in conjunction with a closed cycle-gas turbine.

This report presents a proposal for designing, constructing, and operating a nuclear power producer which is capable of generating 50 megawatts of reactor heat power and 25,000 kilowatts of electrical power.

The reactor is a modification of the liquid metal fissioning reactor developed by Brookhaven. The closed-cycle gas turbine is basically one developed by Escher Wyss, Zurich, Switzerland, and introduced in the United States by the American Turbine Corporation.

It is believed that a reactor core coupled with a closed-cycle gas turbine offers great promise for basic research, development, power production demonstration, and competitive economics.

The proposed reactor turbine combination is capable of demonstrating efficiencies in excess of 40 percent and power costs of 7 mills/kwh.



LIQUID METAL REACTOR WITH
CLOSED CYCLE GAS TURBINE SYSTEM

2.0 INTRODUCTION

It is proposed to design, erect, and operate a 50 megawatt nuclear reactor in conjunction with a closed-cycle gas turbine for the production of electrical power.

This endeavor is suggested as a joint activity between the University of Michigan under the general supervision of its Engineering Research Institute, and organizations and agencies who will provide the initial funds and operating costs for a long-range continuous development program.

The development project would have included in its physical equipment and facilities a closed-cycle gas turbine installation, necessary buildings and auxiliary facilities for fuels make-up, water processing equipment, start-up power, instrument air supply, and laboratories for routine radiochemical analysis.

This proposal considers that the facility will be located within reasonable proximity to the University of Michigan. Such location not only allows maximum use of the facility for education, and a wide variety of research, but also has the great advantage of accessibility to scores of faculty, staff, and graduate students representing skills in a wide variety of scientific and engineering fields. Such location also makes it possible to supplement new physical research equipment with the use of a variety of facilities in many of the now-existing laboratories of the University of Michigan.

In addition to the technical and scientific investigations, it is expected that this facility would provide major means by which business management problems, legal problems, and social problems can be coordinated with technological progress.

From the engineering studies which have been conducted to date, it is believed possible to engineer and construct a development reactor facility with a closed-cycle gas turbine for an initial cost between $6,000,000 and $7,000,000. Estimates of costs for operating the facility indicate expenditures of about $2,000,000 per year for a period of between five to ten years. Thus, the commitment to this development project in the form of capital funds will range between $15,000,000 and $25,000,000.

In this proposal there is described the general plan for the nuclear power generating plant and its attendant facilities. It is proposed to construct a reactor power plant with a closed-cycle gas turbine which has the characteristics shown in Table I.

TABLE I

Reactor Power	50 megawatts (maximum)
Reactor Fuel	U-235 dissolved in bismuth metal
Reactor Pressure	Atmospheric
Reactor Temperature Range	650°C to 1100°C
Gas for Turbine	Helium
Cycle Pressure	1000 psia
Cycle Temperature	90/1400°F
Electrical Power	24,000 KW
Helium Flow Rate	12.5 lbs/KW Hr.

The general flow plan of the reactor cycle integrated with a closed-cycle gas turbine power gererator is shown in Figure I. The core of the reactor power unit is fueled with uranium-235. This uranium-235 is dissolved in bismuth metal and as a homogeneous metal solution is circulated through a critical geometry.

It is believed that by resolving the problems involved in production of heat power from the core of such a reactor possibilities exist in extending the work to evaluate converter reactors, breeder reactors, as well as packaged power generating units. Possible fields of application consist of mobile propulsion, stationary power generation, closed-cycle gas turbine propulsion of merchant ships, and remotely located packaged power generating units.

In addition the peacetime uses of radiochemicals and nuclear interactions in fields of chemical reactions, petroleum, biological work, medicine, foods, metallurgy, etc., can be investigated at a wide range of levels of temperature and radiation activities.

3.0 DESCRIPTION OF THE NUCLEAR POWER GENERATION AND THE CLOSED CYCLE GAS TURBINE

The Babcock and Wilcox Company, coordinated with the efforts of many industrial organizations and the Brookhaven National Laboratory, has indicated that a liquid metal fissioning reactor requires a working experiment before many of the questions for using such a reactor for power generation can be finally determined.

The optimum experiment is defined as one which provides useful design data on mass transfer, heat transfer, effects of radiation poisoning, handling of fission products, reactor specific power, critical mass and geometry, control and safety, as well as proving itself to be a competitive nuclear reactor for power generation.

The employment of a closed-cycle gas turbine in which conventional fuel is used for providing energy to gases has been under development by Escher Wyss Ltd., Zurich, Switzerland. In this country, serious consideration to closed-cycle turbines for power generation is being given by the American Turbine Corporation and Westinghouse Electric Corporation. Due to recent developments in nuclear reactors, heat transfer equipment, turbomachinery, and materials engineering, the combination of nuclear power with a closed-cycle gas turbine appears within reach of practicality and indicates tremendous improvement over any other program for nuclear power.

3.1 Description of the Nuclear Reactor

The nuclear reactor which is proposed for this development project is the core portion of a stationary power plant. It is believed that if the problems for the core can be resolved, many applications of the core as such as well as the core integrated with added reactor designs can cover a wide range of applications. The reactor core is a liquid metal homogeneous system. The system consists of solutions of uranium in molten bismuth circulating through a critical geometry. The research and development for this type of reactor has been and will continue to be done at Brookhaven National Laboratory. Recently, a group of industrial study teams have cooperated with Brookhaven National Laboratory to achieve a practical concept of the reactor system.

The reactor system consists of a critical mass and geometry, of a solution of bismuth and uranium, with attendant heat exchangers for transferring the heat from the circulating uranium-bismuth to a gas such as helium for operation of the closed-cycle gas turbine. A circulating pump is provided to maintain proper velocities of the bismuth-uranium solution through the critical geometry.

From the experimental work done, it appears that the reactor can operate over a range from 600°C to 1100°C depending upon desired power output. In general, the higher the temperature of operation the more efficient will be the power production.

A portion of the circulating uranium-bismuth solution is by-passed through a degasser device which permits removal of a large portion of the gaseous fission product poisons.

As nuclear fuel is consumed, from time to time it will be necessary to add known quantities of fissionable material to the reactor core. Control of the reactor is essentially automatic since concentration of uranium in the bismuth is known at all times.

From preliminary design calculations made for this reactor, Table II indicates a summary of the preliminary design data. (Table II on next page).

The proposed reactor offers great advantages and improvements over other proposed power reactors. Some of the advantages which can be achieved are:

3.1.1 The reactor is small and compact with high specific power.

3.1.2 The system is self-regulating and-controlling, thus eliminating control rods in the core.

3.1.3 Temperature levels ranging from 450°C to 1100°C might be achieved, thus producing available energy at levels that give substantial improvements in overall cycle and power efficiency.

3.1.4 The heat transfer rates for bismuth to helium are high, thus requiring small heat exchange surfaces to transfer energy to the closed cycle gas loop.

3.1.5 The materials of construction for the reactor are in advanced stages of development, thus assuring that a development reactor of this size can be firmed within specified limits.

3.1.6 The critical configurations are dimensionally less for the reactor core, thus requiring a minimum of shielding.

3.1.7 The development project, as a power producing reactor, offers excellent promise for packaged power plants and mobile power propulsion.

The general objectives of building and operating a reactor at the designated power level are to develop and improve the following areas:

3.1.8 The demonstration of safe handling of bismuth-uranium solutions over a wide range of temperatures.

3.1.9 The development of a proven system for degassing fission product poisons.

3.1.10 The demonstration of long-term continuous power production without removing reactor fuels from the circulating loop.

TABLE II

SUMMARY OF PRELIMINARY DESIGN DATA

Power-50 MW heat	= 47,400 BTU/sec.
Net Electrical	20 MW
Reactor Size 2' φ	= 6.3 $ft.^3$
Piping 10" φ	= 6.3 $ft.^3$
Heat Exchanger 22 $ft.^3$(30% for Bi)	= 7.0 $ft.^3$
Total System Volume	19.6 $ft.^3$
Critical Mass	3.28 Kg. - U-235
Total uranium	10 Kg.
Total bismuth	11,000 #
Bismuth flow rate	2820 #/sec. or 1840 gpm
ΔT Bismuth	400° F.
Burn-up	50 g./day
Sp. Power - Core	15,200 KW/Kg.
Sp. Power - System	5,000 KW/Kg.
Min. Temperature for solution	420° C.
Max. Temperature for solution	Limited by materials of construction or boiling point of bismuth 1400° C.
φ Neutron flux	= 6×10^{14} neutrons/cm^2/sec.
Coolant	Helium at 1000 psi
Helium flow	265,000 #/hr. ΔT = 514° F.
Helium temperature in	886° F.
Helium temperature out	1400° F.
Helium specific heat	1.25 BTU/#
Helium film coefficient at $\frac{\Delta P}{P}$	= 1.2% 1200 BTU/hr./$ft.^2$/° F.

Approximate Size of Heat Exchanger

2650 tubes 3/8" φ x 10' x 1/4" ID
100 # helium/hr./tube
Tube Bundle Cross-Section 3' x 18"
Helium inside-bismuth shell side

Temperatures

Bismuth Core Inlet	586° C. = 1182° F.)	These temperatures can be reduced by sacrificing some thermal efficiency. The figures used indicate a thermal efficiency of 40%.
Bismuth Core Outlet	786° C. = 1450° F.)	
Helium in	886° F.)	
Helium out	1400° F.)	

3.1.11 The development of full-scale pumping and the fluid transfer system.

3.1.12 The improvement of methods of heat transfer and materials of construction.

3.1.13 The demonstration of sustained power levels under safe and stable conditions.

3.1.14 The evaluation of the circulating fuel and study methods of continuous side-stream removal of fission products and radiochemicals.

3.1.15 The evaluation of the performance of molten bismuth solutions over wide ranges of temperature drops, isotopic buildups, and fluid mechanics and transfer problems.

3.1.16 The demonstration of the economics of a highly efficient power plant.

It is believed that if these general objectives can be achieved, results of such development may accomplish the following:

3.1.17 The packaged power plant, which is particularly good from the standpoint of requiring small amounts of cooling water.

3.1.18 Merchant ship and sea-going vessel propulsion, which appears would require the smallest volume in shielding of any of the presently proposed reactors and would eliminate bulky condensers and water treatment required for steam turbines.

3.1.19 Breeder reactors, when surrounding the core with a suitable blanket, allow the demonstration of breeding gains sufficient to sustain a fuels economy within the reactor system as well as producing fuels for use in other types of fissioning reactors. The inventory of fissionable fuels over long periods of time is sustained at a minimum.

3.1.20 The development reactor, as a test reactor, appears to be the cheapest way of obtaining a high neutron flux for testing purposes, provides a source of radiation for many fields of endeavor as well as providing fields of research in neutron physics, nuclear chemistry, fluid mechanics, materials engineering, radiation damage, instrumentation and control. Thus, such a development reactor is a necessary implement and additive to the Phoenix Memorial Project, sponsored research in industrial nuclear power, and pure academic research.

3.2 Description of Closed-Cycle Gas Turbines

The American Turbine Corporation has conducted considerable development for applications of closed-cycle gas turbines in the United States. The adaption of a closed-cycle gas turbine to a nuclear reactor makes possible the use of temperatures of nuclear fission at high levels, resulting in power production at high efficiencies. The use of an inert gas as a power plant working fluid, exchanging heat with a molten metal such as a bismuth-uranium solution, eliminates the problem of a remotely operated turbine.

The closed-cycle gas turbine system consists of a dual stage turbine unit with necessary intercoolers. The plan in obtaining such a closed-cycle turbine would be to work closely with the engineering designs being achieved by Westinghouse Electric Corporation, the American Turbine Corporation, as well as the developmental activities of Escher Wyss Ltd., of Zurich, Switzerland.

Present considerations have selected helium as the working fluid. Other possibilities are air, argon, carbon dioxide. From many viewpoints, however, helium offers specific advantages in heat transfer, does not become radioactive, and permits direct operation and maintenance of the turbine equipment. The closed gas loop consists of flowing the gas through a specially designed heat exchanger in which it exchanges heat with the molten bismuth-uranium solution. Thus, the gas flowing through this heat exchanger is raised to a temperature as high as 1450°F and a pressure of about 1000 psig. The gas exhausting from this heat exchanger flows to the high pressure turbine stage which effectively drives the compressor which recompresses the gas for return to the reactor heat exchanger.

The helium exhausts from the high pressure turbine after partial expansion and flows to a low pressure turbine. The turbine drives a 3600 RPM power generator. The gas exhausted from the low pressure turbine is cooled in a recuperator exchanger and intercooler and flows to the first stage compressor driven by the high pressure turbine. The partially compressed gas flows through a second interchanger into the final compressor which is driven also by the high pressure turbine. This compressor returns the gas in a closed loop at about 880°F to the reactor heat exchanger.

For a more detailed explanation of this closed-cycle gas turbine, please refer to the report "Design Study 60 MW Closed-Cycle Gas Turbine Nuclear Power Plant," which is attached to this proposal as an appendix. These data, prepared by the American Turbine Corporation, indicate that efficiencies between 40 and 45 percent might be achieved by combining a reactor of this type with a closed-cycle turbine. It would be expected that the closed-cycle turbine and the auxiliary generating and heat exchange equipment would be provided to the project by one of the licensors of the American Turbine Corporation.

The general development problems in this field are numerous. If helium gas is selected as the working fluid for the closed-cycle gas turbine, consideration of investigations of helium as a diffusing gas into metal become a relatively important problem. The design and development of mechanical seals to contain the working fluid, the optimization of heat transfer surface, and the general characteristics of an optimum closed cycle gas are needed before assured performance for power generation can be achieved. In the event other fluids such as carbon dioxide, air, or argon are selected, measurements of degrees of radiation and amounts of shield for safe operation would be required. It is believed that through the operation of a closed cycle gas turbine in conjunction with the nuclear reactor, measured improvements in gas turbine design for this specific purpose can be achieved and many advantages demonstrated over the nominal steam cycle turbine presently being considered.

4.0 SCOPE AND DIVISION OF WORK

This proposal contemplates the engineering, construction, and operation of a nuclear power plant in conjunction with a closed-cycle turbine under the general supervision of the University of Michigan.

The funds necessary to erect and construct this facility as well as the costs involved in development programs and operations research would be provided to the University of Michigan by others.

Professor H. A. Ohlgren will be the Project Supervisor. He has had many years of experience in the fields of nuclear reactors, nuclear physics, radiochemicals, shielding problems, and fuels recovery. Assisting Professor Ohlgren will be a key group of faculty advisors in conjunction with full-time staff people for the operation of this specific development program. Graduate and undergraduate students will be employed on a part-time basis so that effectively between 10 to 15 technical man years will be consumed in the operation of this developmental facility.

5.0 PRELIMINARY BUDGET COST FIGURES

The work and investigations conducted to date on this type of system are not adequate to predict accurately the total investment required. One of the first steps in this type of a program would be to conduct engineering studies in sufficient detail to permit establishment of a more firm budget.

Order of magnitude estimates, however, have been made. In evaluating costs of a reactor core, heat exchange equipment, auxiliary systems, instruments, buildings, gas turbines, generation and switch gear equipment, the estimates which have been completed to date indicate a magnitude between $6,000,000 and $7,000,000 as the amount needed to engineer and construct this type of a development project.

The operating costs including allowances for development and improvement are anticipated to approach $2,000,000 per year. It is believed that a program for a period of between 5 to 10 years is needed to demonstrate adequately this principle of utilizing atomic power for efficient power production.

APPENDIX I

Design Study

60 MW Closed Cycle Gas Turbine

Nuclear Power Plant

Design Study

60 MW Closed Cycle Gas Turbine

Nuclear Power Plant

American Turbine Corporation December 1954.

ATC-54-12

Design Study

60 MW Closed Cycle Gas Turbine

Nuclear Power Plant

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Design Study
60 MW Closed Cycle Gas Turbine
Nuclear Power Plant

1.0 Summary

This report covers the design of the power generating equipment of a closed cycle gas turbine nuclear power plant having the following characteristics:

Output at generator terminals	60 MW
Cycle pressure	1000 psia
Cycle temperature	90/1400°F
Reactor output	148.5 MW
Heat rate	8460 BTU/KW Hr.
Helium flow rate	12.5 Lbs/KW Hr.

Reactor design is not covered in this report, it would, however, be a graphite moderated enriched fuel reactor designed for conversion or breeding and its requirements were taken into account in the selection of the power plant cycle and plant operating conditions as summarized above.

2.0 Introduction

The closed cycle gas turbine power plant using fossil fuel fired heaters has been under intensive development by the firm of Escher Wyss Ltd. of Zurich, Switzerland and their licensees for the past fifteen years. The idea of a power plant using a gas as a working fluid is not new but its development as a practical power plant has only been possible during recent years due to current developments in turbomachinery, heat transfer apparatus and high temperature materials. Adaption of this power plant cycle to a nuclear reactor makes possible the use of the temperature resulting from nuclear fission at a high level, resulting in a power plant of high efficiency. Use of an inert gas as a power plant working fluid and reactor coolant removes the problems of chemical attack on power plant components in the transport of heat from the reactor to the point of conversion into mechanical energy and since there is no phase change in the reactor coolant there is no necessity of providing for secondary containment of the reactor in event of gross temperature excursions.

Heat distribution in a closed cycle gas turbine power plant is illustrated in Figure 1 which shows the gas flow path super-imposed on a compressor/turbine assembly . The gaseous working fluid is compressed in a compressor, heated directly in a nuclear reactor and the gas expanded in a tur-

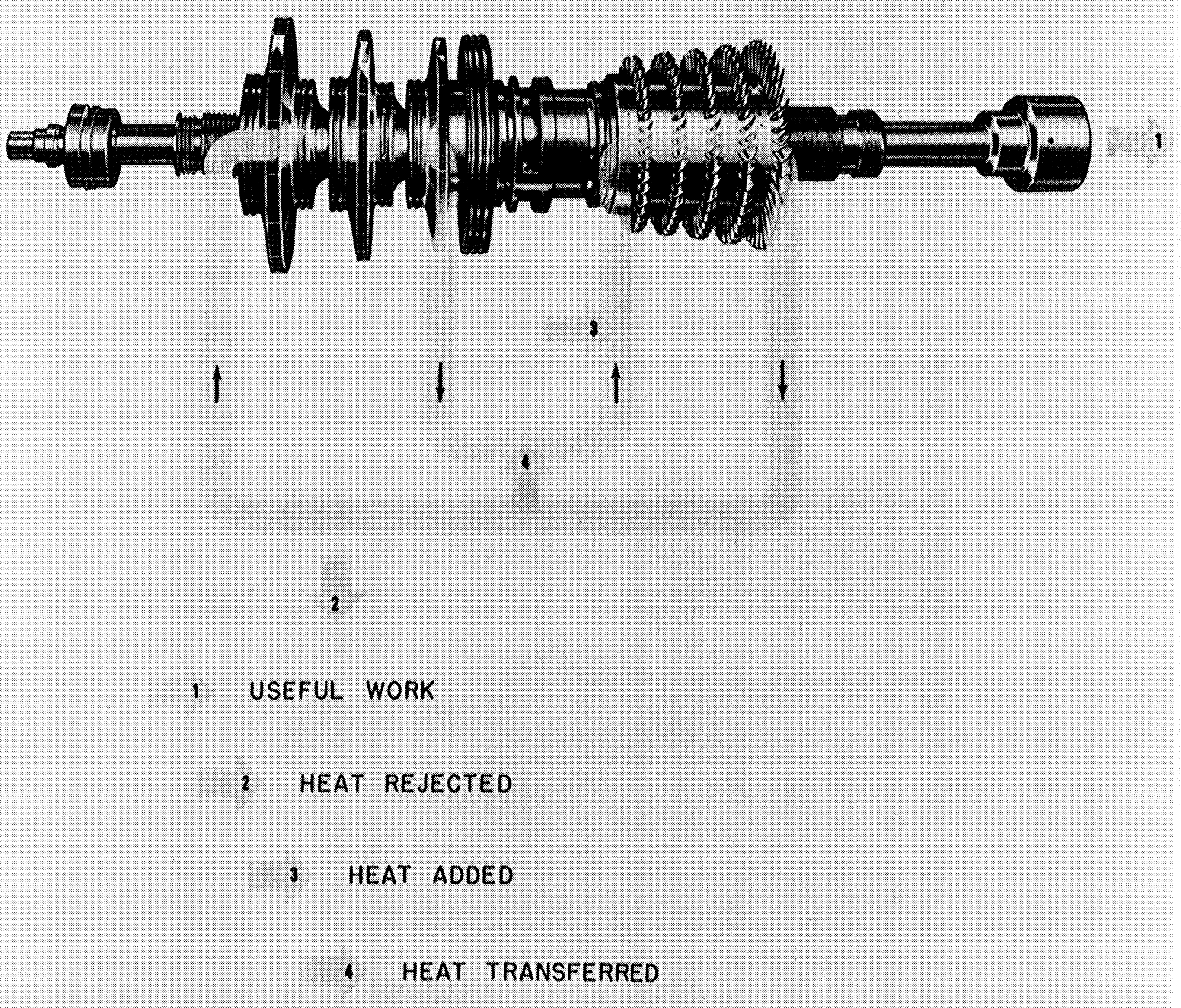


FIG. I.

HEAT DISTRIBUTION DIAGRAM
CLOSED CYCLE GAS TURBINE POWER PLANT

bine, giving up energy in expansion to drive the compressor and produce useful power. The energy input required is reduced, or the efficiency of the plant is increased, by transfer of heat from the gas after expansion to the gas after compression in a recuperator. Unavailable energy from the process is rejected in a precooler, thereby restoring the gas to its initial temperature before compression. The system is closed and the process is one of continuous flow. Load changes are effected by varying the pressure level of the system, bypassing the compressor, or both. Efficiency is substantially constant over a wide range of loads as the temperature differences remain constant as do gas velocities and peripheral speeds of the turbomachinery. Efficiency is affected by changes in sink temperature but output can be held constant by changes in system pressure level as shown in Figure 2.

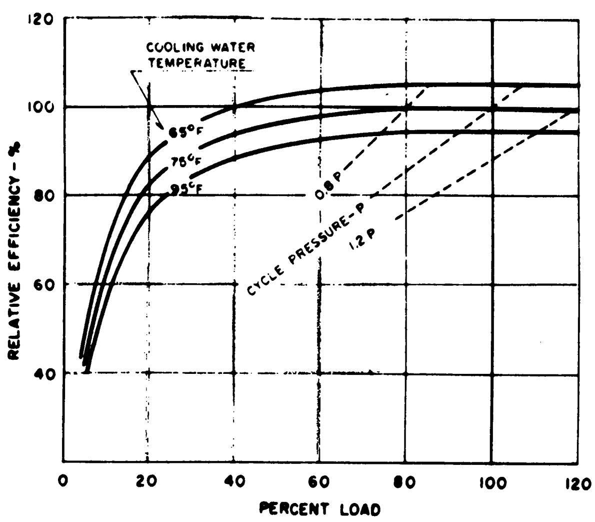


Figure 2 Part Load Efficiency

3.0 Power Plant Cycle

This power plant has been designed using helium as the working fluid and reactor coolant, due to its zero neutron capture cross section and its good heat transfer characteristics. Major equipment items are the machinery set, heat transfer apparatus and control system. The machinery set consists of a two-stage axial flow compressor set with intercooling between stages driven at constant speed by a high pressure turbine. Helium leaves the high pressure turbine after partial expansion and flows directly to a low pressure turbine directly connected to a 3600 RPM generator of 60 MW output. Heat transfer apparatus is of the extended surface type, plate fin, or finned tube, depending upon the application. The control system consists of a master regulator to maintain constant system pressure with a low pressure receiver-high pressure accumulator system and transfer pump.

A flow diagram of the complete power plant cycle, including control and helium purification system is shown in Figure 3. Table I lists the state points at the inlet to all pieces of machinery and heat transfer apparatus as well as a heat balance of the power plant.

The basic principles of design of a closed cycle power plant require the extensive use of heat transfer surface for the recuperation of heat available in the working fluid after

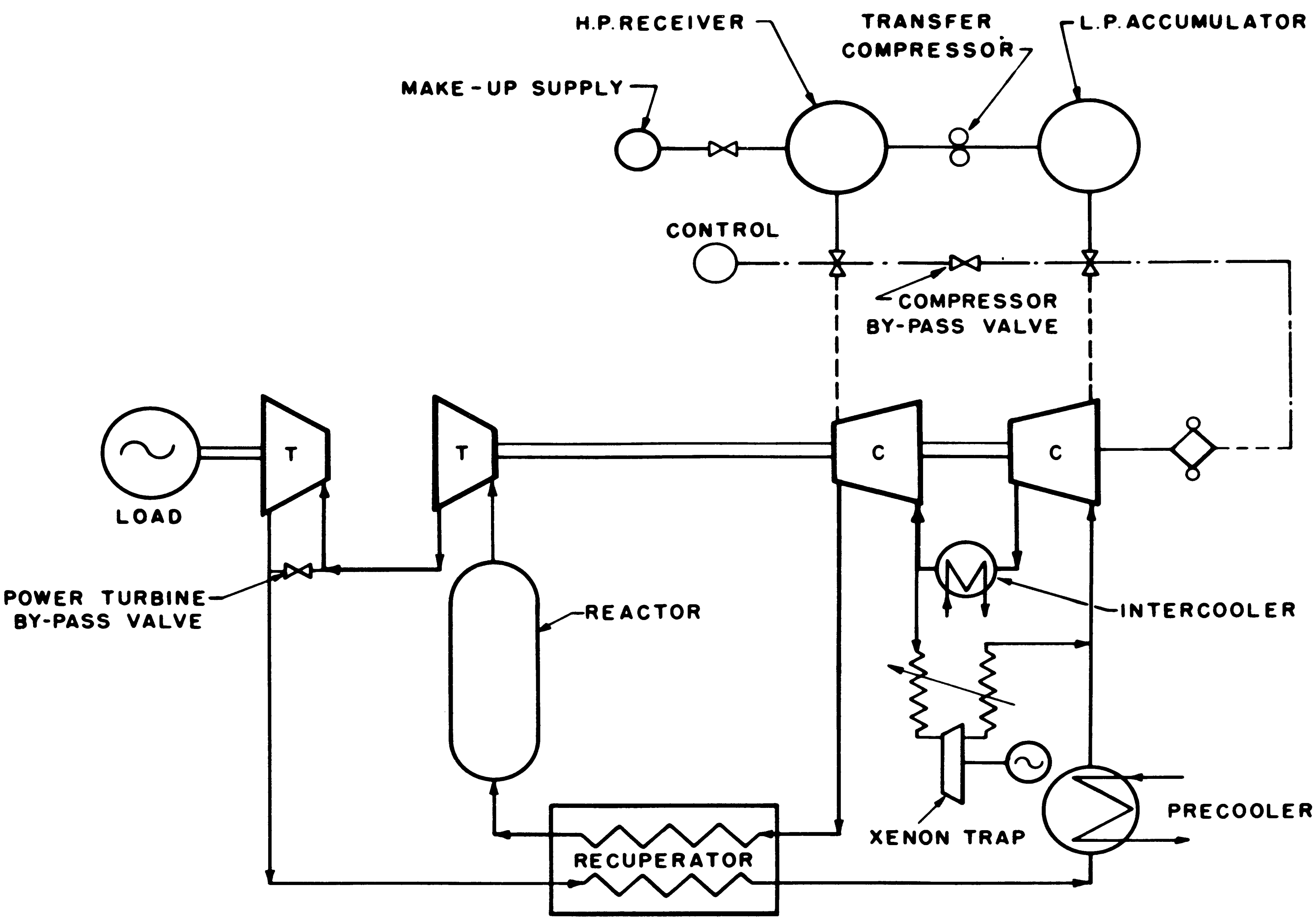


FIG. 3 - FLOW DIAGRAM

TABLE I

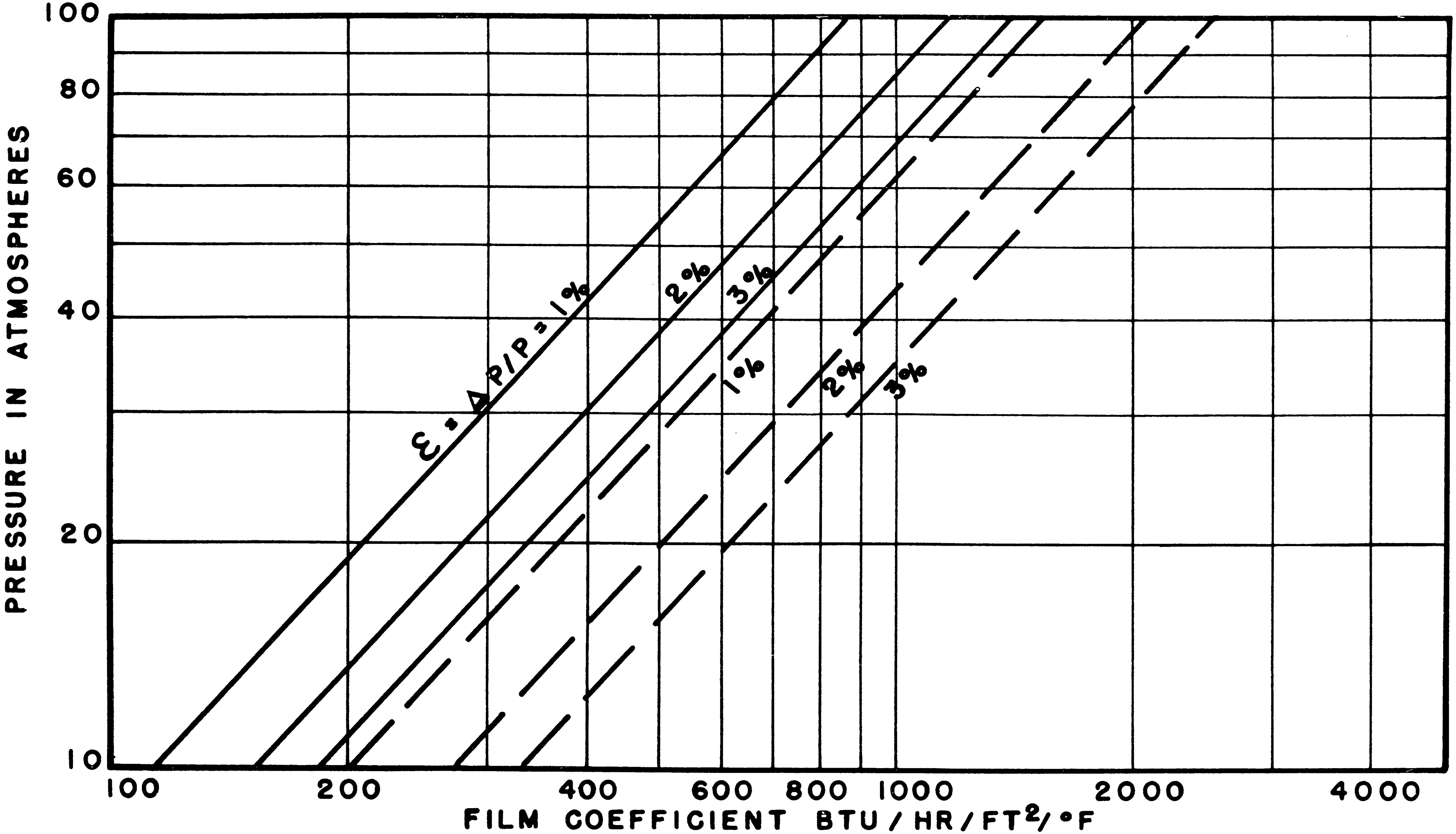
State Points & Heat Balance
W = 790,000#/Hr = 219#/sec.

Point	at inlet	T(°R)	ΔT	R_c-R_e	ϵ(%)	P/P_1	P(Psia)	ρ	Q(CFS)	+MW	-MW
1	LP Compressor	550	120	1.55		1.000	426	.289	760	34.7	
2	Intercooler	670	120		0.75	1.550	661	.368	596		34.7
3	HP Compressor	550	120	1.55		1.538	656	.445	494	34.7	
4	HP Recuperator	670	676		1.50	2.382	1015	.565	388	196	
5	Reactor	1346	514		1.50	2.345	1000	.279	787	148.5	
6	HP Turbine	1860	240	1.48		2.310	985	.1975	1110		69.4
7	LP Turbine	1620	218	1.51		1.56	665	.153	1435		63
8	LP Recuperator	1402	676		2.25	1.033	440	.1175	1870		196
9	Precooler	726	176		1.00	1.010	430	.231	950		50.8
	Rejected to oil										1.5
	Electrical Losses										1.5
	Net Output										60

expansion. The high system pressure at which a closed cycle gas turbine power plant operates, coupled with the fact that the working fluid is uncontaminated by products of combustion, makes attainment of this objective relatively easy to accomplish through the use of heat transfer surface having flow passages of small hydraulic diameter and/or the use of extended or interrupted surface.

All fossil fuel fired closed cycle power plants built to date use air as the working fluid. The use of helium with its different gas properties makes necessary the re-evaluation of the generally accepted cycles as used with air.

Compared to air, helium exhibits a considerable improvement in heat transfer properties, the film coefficient being about 180% higher under the same conditions of pressure level, pressure drop and passage geometry. This is illustrated in Figure 4, being a plot of film coefficient at various system pressure levels at constant relative pressure drops ($\Delta P/P$) for helium and nitrogen flowing in a one inch diameter tube having an l/d of 100. Since air is composed of 75% nitrogen there is no noticeable difference in the heat transfer properties of air or nitrogen. This improvement in helium film coefficient over that of air results in heat transfer equipment approximately 55% of the size required for air, or where the equipment is the same



VARIATION OF FILM COEFFICIENT
WITH PRESSURE FOR CONSTANT ΔP/P
FLOW IN 1" TUBE 100" LONG AT 1000°F
NITROGEN ——— — — HELIUM

FIG. 4.

size, an appreciable increase in the quantity of heat that can be transferred, the exact amount depending upon the effectiveness of the surface with air. For example, if the recuperator in the air cycle machine has an effectiveness of 80%, the same surface will have an effectiveness of 86% in a helium cycle. If the effectiveness in an air cycle machine is 90%, which is more of the order used in a closed cycle power plant, the same surface will have an effectiveness of 95% in a helium cycle machine.

On the other hand, the high specific heat of helium (1.25 BTU/#) makes the design of the turbomachinery difficult, the number of stages required for the same temperature rise being roughly proportional to the specific heat, or compared to air, in the approximate ratio of 5:1 (1.25:.24). However, a cycle analysis will show that the compressor temperature ratio required for maximum cycle efficiency decreases with increasing recuperator effectiveness and this fact is made use of in the design of a closed cycle plant using helium as a working fluid, trading static heat transfer surface for stages of turbomachinery.

In order for any closed cycle nuclear power plant to be attractive economically it is necessary that it be a high temperature machine, i.e. operate at cycle temperatures in excess of approximately 1200°F. All experience to date with closed cycle power plants has been at cycle temperature from 1250 to 1300°F as dictated by limiting tube wall temperature in a fired air heater. In a nuclear plant this restriction is removed and turbine inlet temperature is only limited, within reason, by reactor outlet temperature. However, our present experience would limit such temperature to about 1500°F for a power plant of conservative design. In establishing a cycle for a helium plant, the value of 1300, 1350 and 1400°F cycle temperature was assessed against a 1300° cycle temperature with 100 and 200° reheat.

Referring to Figure 5, there are three ways in which reheat can be applied in such a plant. The first is the use of dual reactors, each having a single flow circuit. These reactors would have the same outlet temperature but slightly different inlet temperatures. Their outputs would be roughly of the same order. An alternate would be the use of a single reactor with a dual flow circuit. On first inspection it would seem that it would be desirable to use an annular core taking the heat required for compressor drive from one half and the heat for power from the other half. However, such a scheme

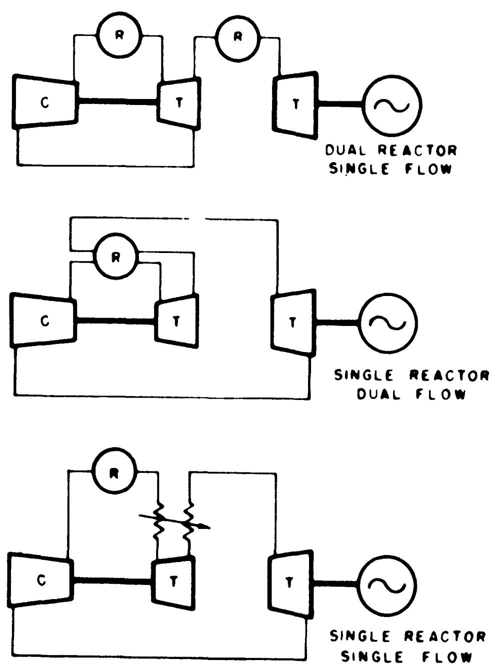


Figure 5. Reheat Systems

would appear to be unduly complicated from the standpoint of ducting required. A simple solution to the problem, from the standpoint of the reactor, is by the use of a single reactor with a single flow and a reheat exchanger. In such a scheme the reactor outlet temperature is in excess of the first turbine temperature, the working fluid temperature being reduced by a reheat exchanger as shown. Cycle studies are identified by the initial turbine inlet temperature and the amount of heat used in the second or power turbine. For example, a 1300° cycle with 100° reheat would be a cycle wherein the initial turbine temperature was 1300°F and its discharge temperature raised 100°F in a reheat exchanger. Thus the reactor outlet temperature would be 1300 + 100 or 1400°F.

A comparison of cycle efficiencies of reheat and non-reheat plants is given in Figure 6 using reasonable polytropic efficiencies for the turbomachinery and taking pressure losses in the two types of systems into account. It will be seen that there is little difference in efficiency between a 1400° non-reheat and a 1300°/200° reheat plant. Since the latter shows its optimum efficiency at a 3:1 pressure ratio compared to a 2.5:1 for the non-reheat plant and also involves the use of a reheat exchanger it is believed that the problems involved in designing a 1400° non-reheat plant are less than those of a 1300°/200° reheat plant. Therefore, a 1400° non-reheat was chosen for further study.

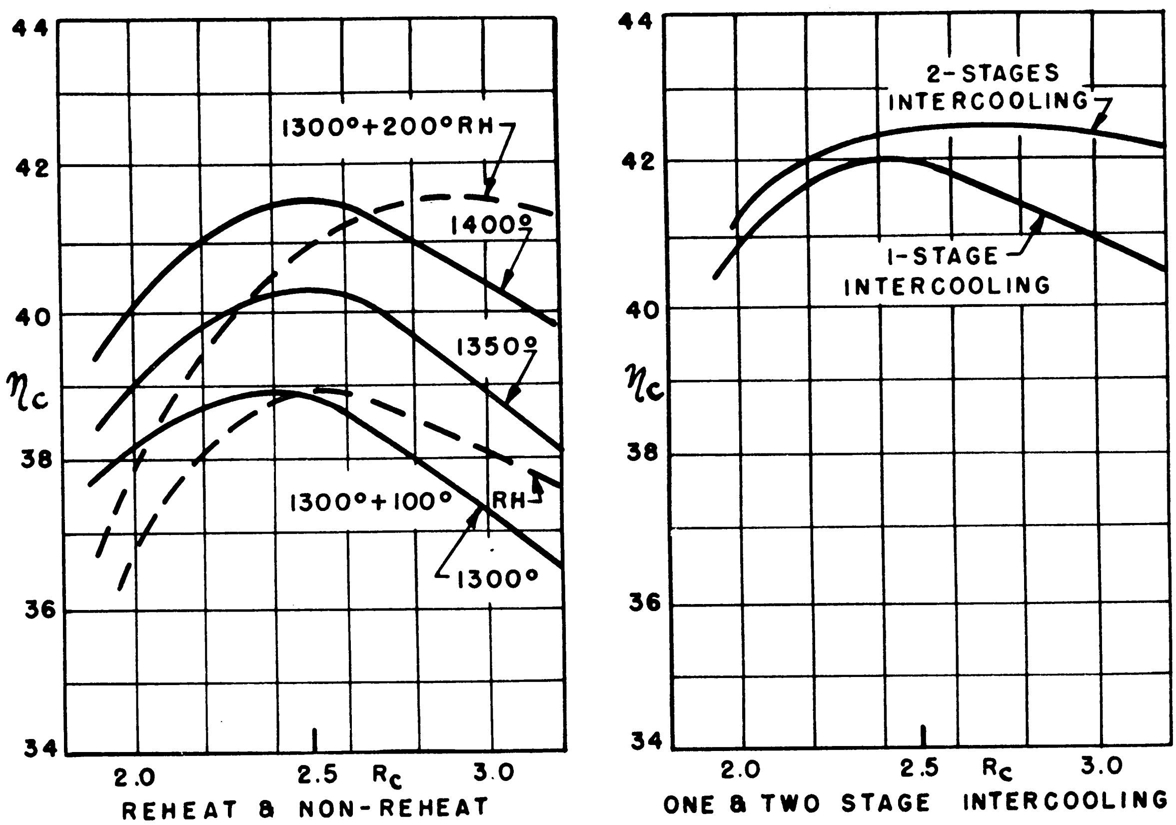


Figure 6. Cycle Efficiencies

Appreciable gains in efficiency in a closed cycle power plant are effected by moderate intercooling. The value of single vs. two stage intercooling was assessed and the results are presented in Figure 6. It will be seen that the use of two stage intercooling results in but 1/2% increase in cycle efficiency (42 to 42.5%) but at the same time additional compression ratio is required to achieve this optimum efficiency (2.4:1 vs. 2.8:1). Since even a simple non-intercooled helium compressor will present enough problems at this stage in development, and 42% cycle efficiency will result in substantially 40% efficiency at the generator terminals, we chose to follow the design of a plant with single intercooling.

The design conditions selected for the power plant cycle are as follows:

Pressure ratio	2.4:1
Pressure losses	7.0%
Expansion ratio	2.23:1
Compressor inlet temperature	90°F
Turbine inlet temperature	1400°F
Recuperator effectiveness	93%
Mechanical & electrical losses	5%
Working fluid flow	790,000 Lbs/Hr.

4.0 Design Assumption

The current state of the art of design of turbomachinery of high stage loadings indicates that it is possible to obtain polytropic stage efficiencies of 89% in compression and 88% in expansion. While stage efficiencies of turbines would normally be higher than compressors for the same work per stage, in an effort to reduce the number of stages in the turbine to an acceptable level the work output per stage is about three times that of the compressor stage. This accounts for the lower turbine stage efficiency. Figure 7 is a plot of adiabatic turbine and compressor efficiency vs. pressure ratio and are the values used in the final cycle analysis. Compression work is equally divided between the high and low pressure compressor and each has a pressure ratio of 1.55:1 resulting in an overall pressure ratio of 2.4:1.

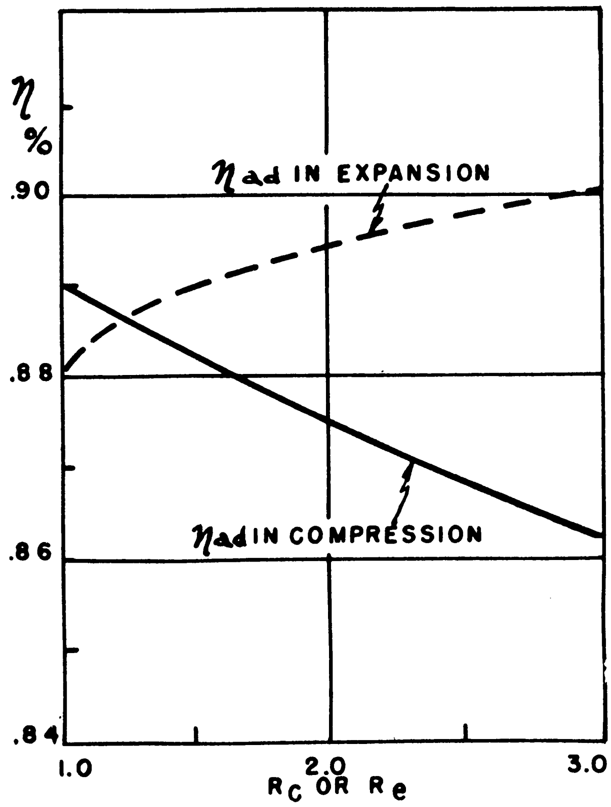


Figure 7 - Turbomachinery efficiencies.

Pressure losses in the plant have been assessed at 7%, resulting in an overall expansion ratio of 2.4x(1-.07) = 2.23:1. Pressure losses are distributed as follows:

Intercooler........0.75%
HP Recuperator.....1.50%
Reactor............1.50%
LP Recuperator.....2.25%
Precooler..........1.00%

While these pressure losses may appear to be optimistic, it is believed they can be attained with careful design and without excessively large heat transfer surface. Referred to an air cycle plant this is the equivalent of a total overall pressure loss of 11%, assigning 5.5% total pressure loss to the reactor which would be the equivalent pressure loss in a fired heater.

Compressor inlet air temperature is assumed at 90°F based on 75°F cooling water which can be obtained in most locations where water supply is critical. If such a plant were located where there was an abundant supply of water at say 55 to 60°F, the efficiency of the plant would increase about 2%.

A regenerator effectiveness of 93% has been selected for design since a plant of this type would be a base load machine. This is slightly less than the maximum that has been used in closed cycle plants (94% in the experimental Escher Wyss plant of 2000 KW) and is believed to be reasonable for a plant of this type.

Helium characteristics are taken from a critical review of available information made by General Electric and are as follows:

$$C_p = 1.25 \text{ BTU/\#}$$
$$\gamma = 1.658$$
$$R = 386.2$$
$$\frac{\gamma - 1}{\gamma} = .398$$

The variation of thermal conductivity, viscosity and Prandtl number with temperature are taken from data of the Oak Ridge National Laboratory and are presented in Figure 8.

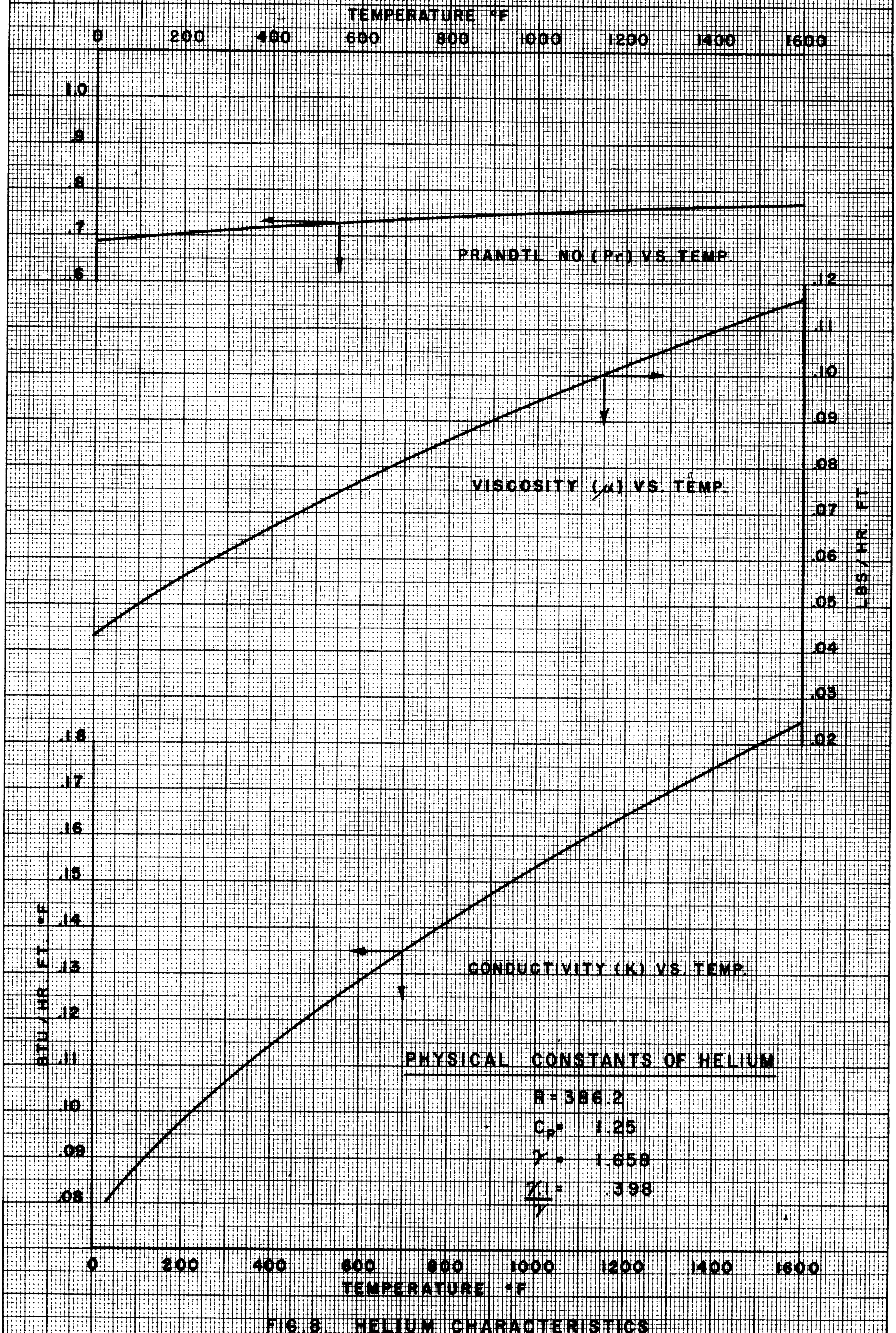


FIG 8. HELIUM CHARACTERISTICS

5.0 Turbomachinery

5.1 Introduction

Modern fluid mechanics and gas dynamics (upon which turbomachinery design is based) are founded upon the concepts of perfect fluid, potential flow and real fluid corrections to this flow. Both the theoretical and experimental developments of the science are expressed in terms of kinematic non-dimensional parameters and non-dimensional groups such as Reynolds Number and Mach Number. Therefore, if the effects of these non-dimensional variables are properly applied, much of the existing work on turbomachinery for air can be used for the design of units for another gas, such as helium.

The use of helium working fluid results in two major changes, one advantageous and one disadvantageous. The fact that the sonic velocity in helium is about three times that in air results in practical release from Mach number (compressibility) restrictions. Therefore, tip speeds are limited only by structural considerations. On the debit side, the high C_p of helium (about 1.25 against .24 for air) results in a temperature rise per stage of only one fifth that of air for the same blading diagram and enthalpy input per stage ($C_p \Delta T$).

In spite of the low ΔT per stage, the required number of stages as compared to present day air practice is low. This is because of the low temperature ratios required for the nuclear helium plant. Thus, the resulting mechanical design of a nuclear helium machine bears a close resemblance to its air counterpart.

In order to give the proposed 60 MW machine performance a strong foundation in reality, the blading and proportions of the set have been closely patterned on existing compressors and turbines. The units chosen were selected as good examples of present day practice for machines now running in industrial service. The compressor and turbine were designed for and tested in air, with air Mach Number limitations an important factor. The helium design is so closely patterned on the air machines that no attempt was made to take advantage of the release from Mach Number limitations. Hence, there should be a slight unaccounted for performance bonus, which follows the conservative design pattern adopted for this unit with the idea of providing as nearly guaranteed performance as is possible in the preliminary design of an unconventional machine.

The tip speed, solidity ratios and blade diagrams are closely patterned after those of the reference machines, often being identical. Hub-tip ratios are similar to those of the models (except where some modifications are needed to match the speed and volume flow of the helium machine).

When considering helium as a real frictional fluid capable of separation from boundary walls in an adverse pressure gradient, the question of being able to maintain the same blade velocity diagram without flow separation arises. The answer is that for a given velocity diagram, the ability to resist separation is directly proportional to a fractional power of Reynolds Number ($Re^{1/n}$). Now $Re = CL/\nu$, where L is a significant length, ν the viscosity and C, the axial velocity through the machine. Now if we assume C is the same for air and helium machines, then

$$\left(\frac{L}{\nu}\right) \text{helium} \geqslant \left(\frac{L}{\nu}\right) \text{air}$$

if we expect to utilize the same velocity diagram.

Since ν takes values in this helium machine roughly 1/3 of that for the usual air machine, then the characteristic length, say blade chord, must be at least 1/3 that of a comparable air plant. This condition is greatly exceeded with the result that this 60 MW plant operates with Reynolds Number higher than the test machines and therefore can be expected to equal or exceed the test efficiency, if all other factors are equal.

5.2 Compressors

Both the high and low pressure compressors are discussed at the same time since each has the same overall pressure ratio and blading design. The only difference is the increase in hub-tip diameter ratio and decrease in length of the high pressure unit due to the greater density and consequent reduced volume flow in this machine. The blading is the standard free-vortex, axial stator inlet type pioneered by the Escher Wyss AG of Zurich, Switzerland. This type of blading is conservative in concept and application but provides outstandingly high efficiencies and numerous constructional advantages for the many gas turbine and industrial blower installations in service. This blading is especially attractive for a helium machine since the Mach Number restrictions, which provide the limiting conditions for pressure rise coefficient in air, do not apply in the less compressible helium.

Both compressor units are based directly upon test data, in air, covered in Escher Wyss Report No. TK-45-046, pertinent data being presented in Figure 9. It should be emphasized that this test work represents conservative but efficient practice which resulted from many years research and industrial application with this type of machine

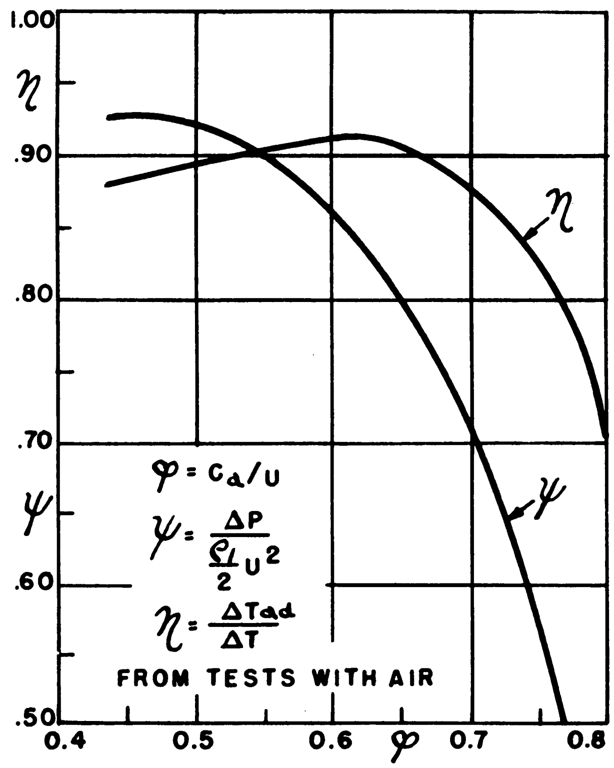


Figure 9. Compressor Stage Characteristics

and which has been further proven in service in the years since 1945.

Characteristic data for the compressors is presented in Table II and blade velocity diagrams in Figure 10. A typical eight stage compressor rotor having this type of blading is illustrated in Figure 11.

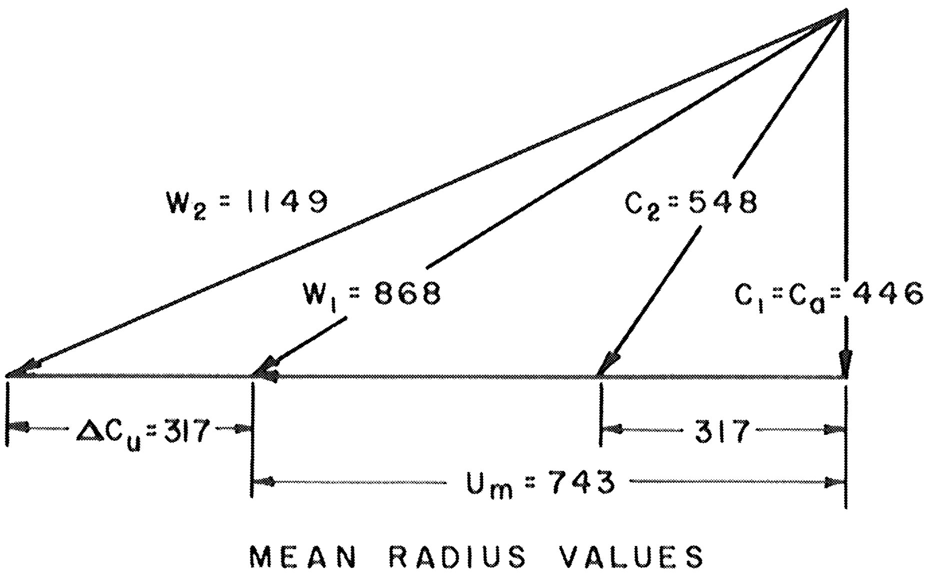


Figure 10. Compressor velocity diagram

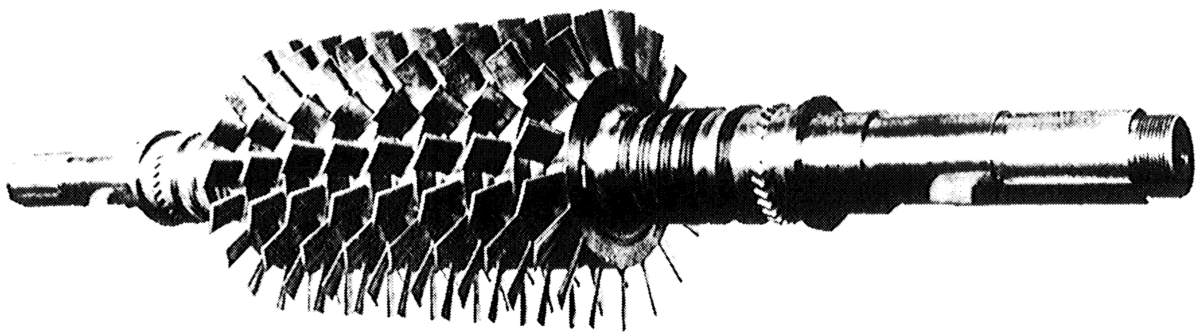


Figure 11. Photograph of typical compressor rotor

TABLE II

Compressor Data

Rotational Speed, N ....7400 RPM

Tip Speed (first stage, max. value) U_t .... 853 ft/sec

Mean radius tangential blade velocity, U_m..... 743 ft/sec

Axial velocity, C_a............................ 446 ft/sec

Pressure ratio LP unit.......................1.55

Pressure ratio HP unit.......................1.55

Overall pressure ratio.......................2.40

Number of stages, HP unit.......................14

Number of stages, LP unit.......................14

Stage pressure ratio.........................1.0318

ΔT_{ad} per stage............................7.54°F

$$\psi_m = \frac{\Delta P_{Stage}}{\rho/2\ U_m^2} = \frac{2 \Delta C_{u_m}}{U_m} = \qquad .865$$

$$\varphi_m = \frac{C_a}{U_m} = \qquad .60$$

Adiabatic efficiency assumed for design...... 88%

Polytropic (stage) efficiency assumed
for design.................................. 89%

Polytropic (stage) efficiency obtained
on test by Escher Wyss with same blading
at same operating point...................... 91%

Hub-tip ratio, LP unit, stage 1............. .74

stage 14............ .79

Hub-tip ratio, HP unit, stage 1............. .82

stage 14............ .86

Compressor Dimensions

Rotor tip diameter HP, stage 1	26.4"
stage 14	25.7"
Rotor tip diameter LP, stage 1	25.2"
stage 14	24.7"
Length of LP compressor-rotor	48.7"
Length of HP compressor-rotor	31.2"
Spacing between rotors	24.0"
Overall Dimensions, 1st stage LP to last stage HP	103.9"

The compressors were designed on the basis of constant mean radius for all stages and for the same mean radius velocity diagram at each stage. The radial velocity variations follow free-vortex principles. All non-dimensional coefficients are based upon mean radius conditions and mean radius tangential velocity.

Rotor construction is of the multiple disc type with rotor torsional and bending loads carried near the periphery of the discs. The compressors and high pressure compressor-driving turbine are housed in a horizontally split case.

5.3 High Pressure Turbine

The high pressure turbine supplies the power for operation of the compressors. It is mounted with them as a single shaft, three bearing, self-running turbo-compressor set. No useful output torque comes from this turbine. The turbine blading is a high efficiency type, also based on Escher Wyss practice, and is a type successfully used in all closed cycle gas turbine power plants built by this firm to date. A U/Co diagram of the blading performance is given in Figure 12.

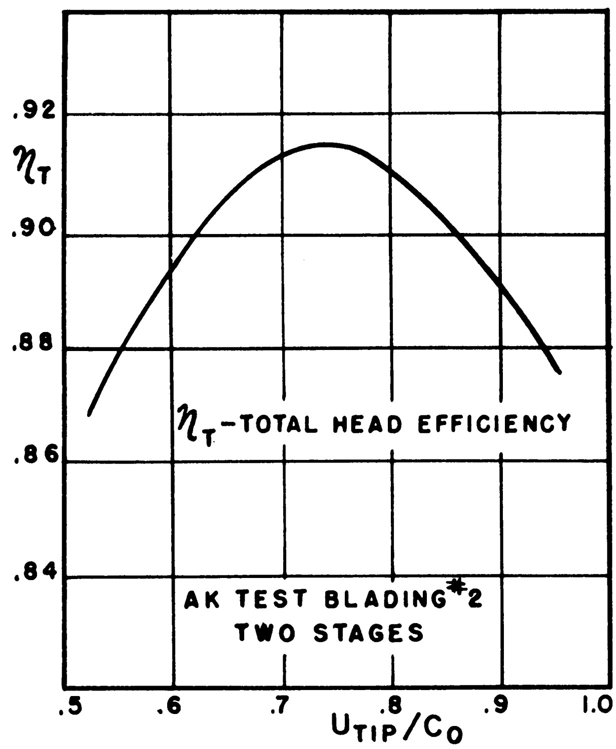


Figure 12. Turbine stage characteristics

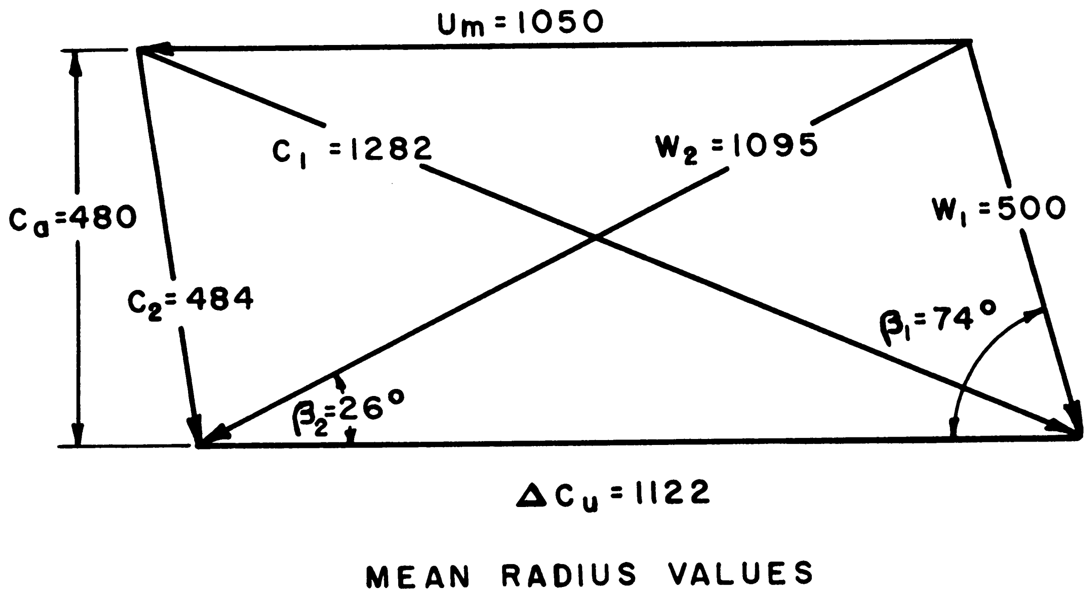


Figure 13. HP turbine velocity diagram

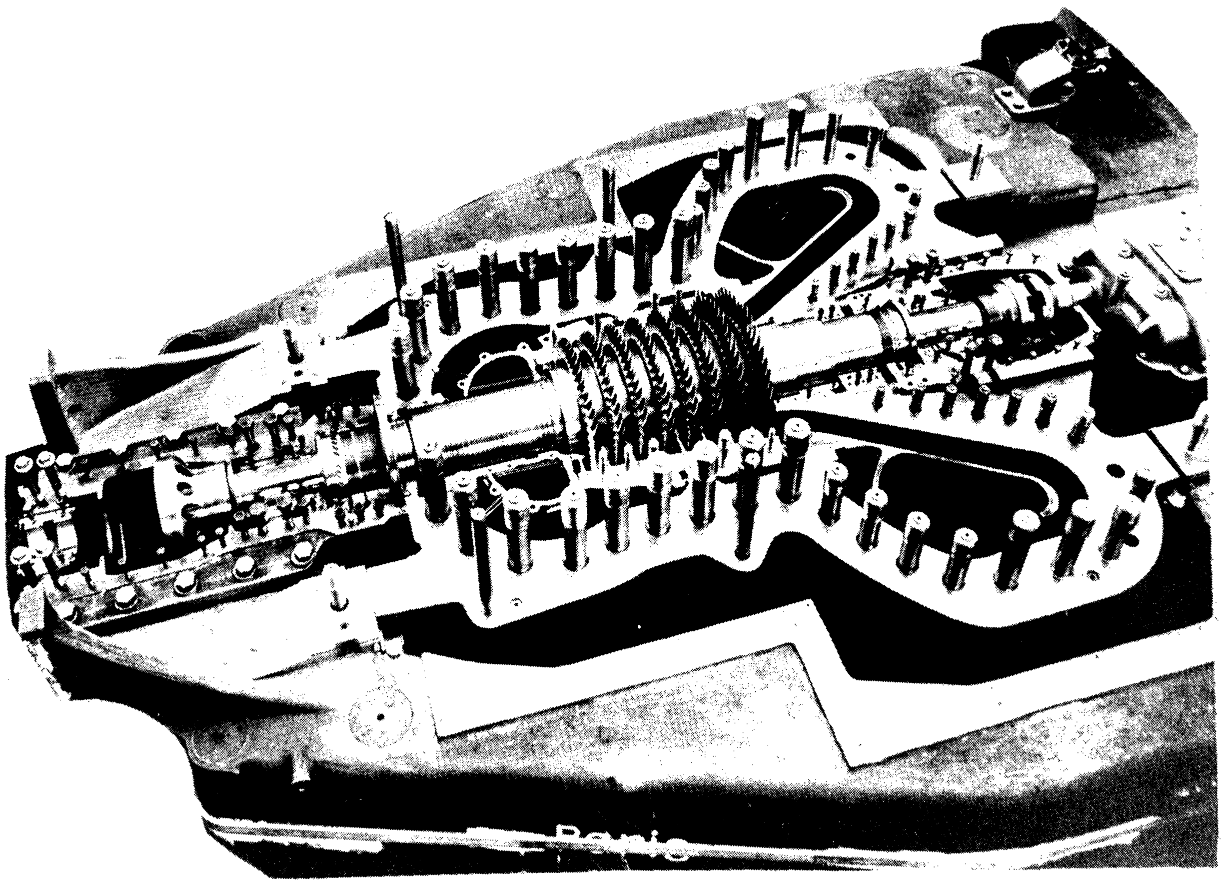


Figure 14. Photograph of typical turbine

Characteristic data for the turbine is presented in Table III and blading velocity diagrams in Figure 13. A typical multistage turbine using blading of this type is shown in Figure 14. In this design, multiple disc construction is used for the high pressure turbine.

5.4 Low Pressure Power Turbine

The low pressure power turbine is mechanically independent of the compressors and high pressure turbine, and directly coupled to the generator. The power turbine therefore rotates at a constant speed of 3600 RPM and produces the entire useful work output for the plant.

The blading is basically the same as that in the HP turbine and derives from the same test data. However, the tip speed is only about 75% that of the HP machine and the gas velocities are reduced proportionally.

Characteristic data for the LP turbine is presented in Table IV and blading velocity diagrams in Figure 15.

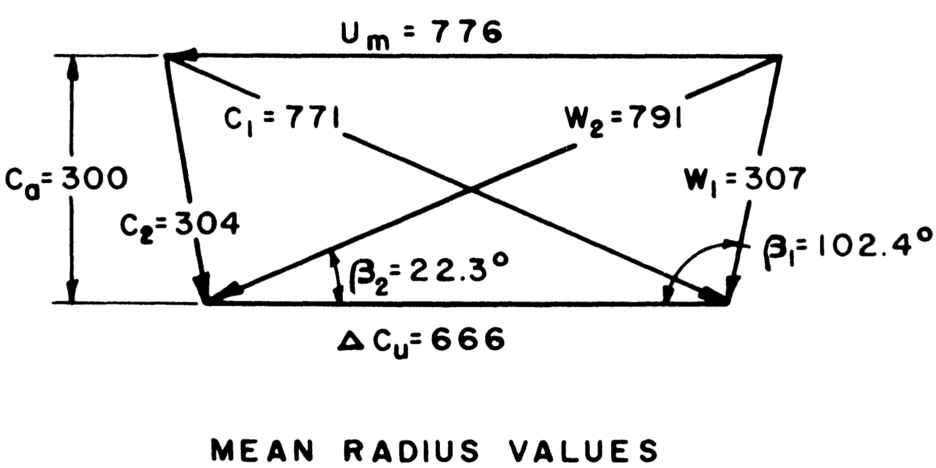


Figure 15. LP turbine velocity diagram

TABLE III

High Pressure Turbine Data

Rotational Speed, N	7400 RPM
Maximum blade tip speed (stage 6), U_t	1186 ft/sec
Mean radius tangential blade velocity, U_m	1050 ft/sec
Axial velocity, C_a	480 ft/sec
C_o	1688 ft/sec
Δh_{ad}	56.9 BTU/stage
Number of stages	6
Hub-tip ratio, stage 1	.82
stage 6	.77
Number of blades in stage 6	112
Centrifugal stress in stage 6 blades (parabolic tapered blades with 3:1 area ratio)	16,860 psi
Gas force per blade	73.2 Lbs.
Adiabatic efficiency	89%
Polytropic (stage efficiency)	88%

High Pressure Turbine Dimensions

Rotor tip diameter, stage 1	35.7"
stage 6	36.6"
Length of rotor	13.3"

TABLE IV

Power Turbine Data

Rotational Speed, N	3600 RPM
Maximum blade tip speed (at stage 14) U_t	867 ft/sec
Mean radius blade tangential velocity, U_m	776 ft/sec
Axial velocity, C_a	300 ft/sec
C_o	1057 ft/sec
Δh_{ad}	22.2 BTU/stage
Number of stages	14
Hub-tip ratio, stage 1	.84
stage 14	.79
Number of blades in stage 14	107
Centrifugal stress in stage 14 blades (essentially untapered blades)	15,200 psi
Gas force per blade	40.4 Lbs.
Adiabatic efficiency	89%
Polytropic (stage) efficiency	88%

Power Turbine Dimensions

Rotor tip diameter, stage 1	53.8"
stage 14	55.2"
Length of rotor	31.4"

The power turbine will also be of multiple disc construction. The low values of tip speed and turbine inlet temperature allow comparatively inexpensive material specification. The turbine is a two bearing machine mounted in the same casing with the high pressure compressor turbine set.

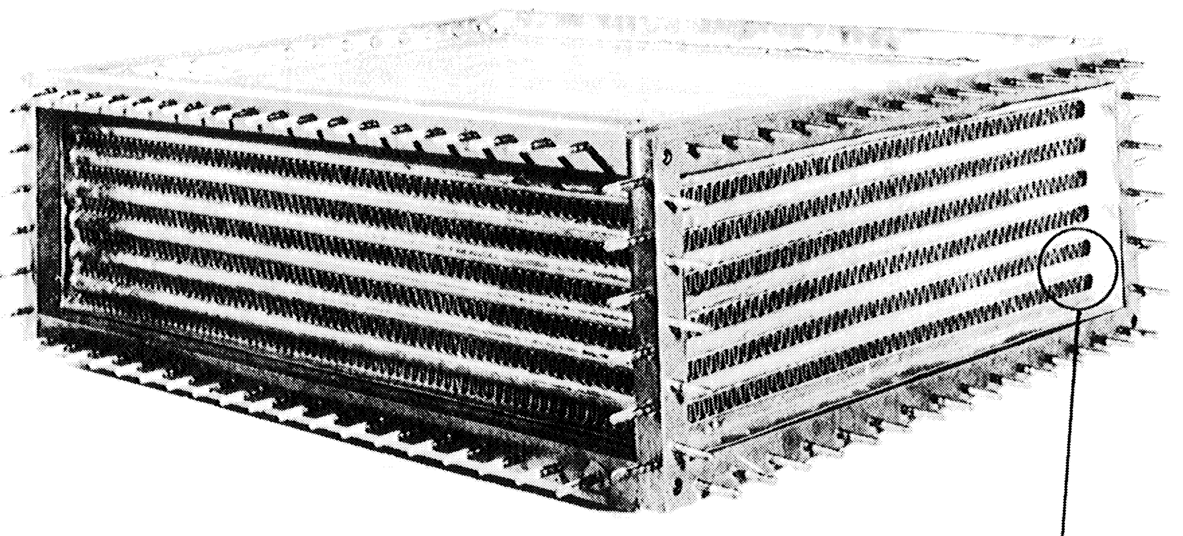
6.0 Heat Exchange Apparatus

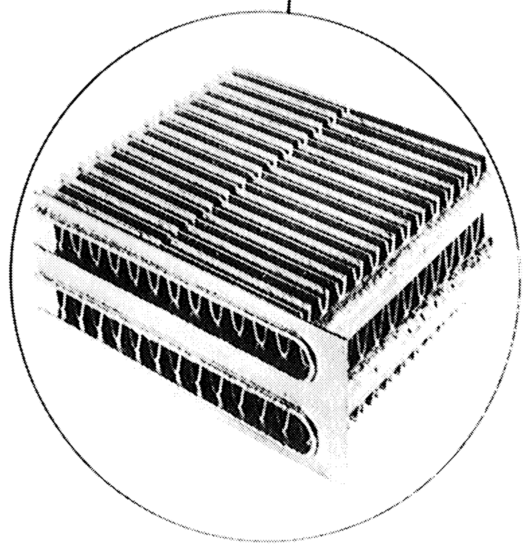
Efficient heat exchange apparatus plays an important role in the design of a closed cycle power plant. All exchangers operate at an effectiveness of at least 80% and may exceed 90%. This requires that the units be of special design if their dimensions are to be kept within reason. The problem is alleviated somewhat by the fact that the working fluid is clean, thus permitting the use of compact surfaces with flow passages of small hydraulic diameter. Many such surface types have been conceived of in recent years and a large number have been tested at Stanford University in government research.

The heat exchange surface proposed for the recuperator of this plant is a welded and brazed plate fin assembly, as manufactured by Griscom-Russell, which has been tested and found to have desirable flow friction and heat transfer properties. Surface characteristics are detailed in Figure 16.

The recuperator is composed of two units, each of which contains a rectangular plate fin section 4'2" x 6' x 17' long. Figure 17 is a drawing showing the recuperator details. Table V lists the recuperator specifications. The LP gas flows directly through the unit in a single pass. The HP gas flows in cross-counterflow, making four passes in its travel through the recuperator.

The Griscom-Russell exchanger is essentially a structure comprising a corrugated fin and a flat plate brazed together

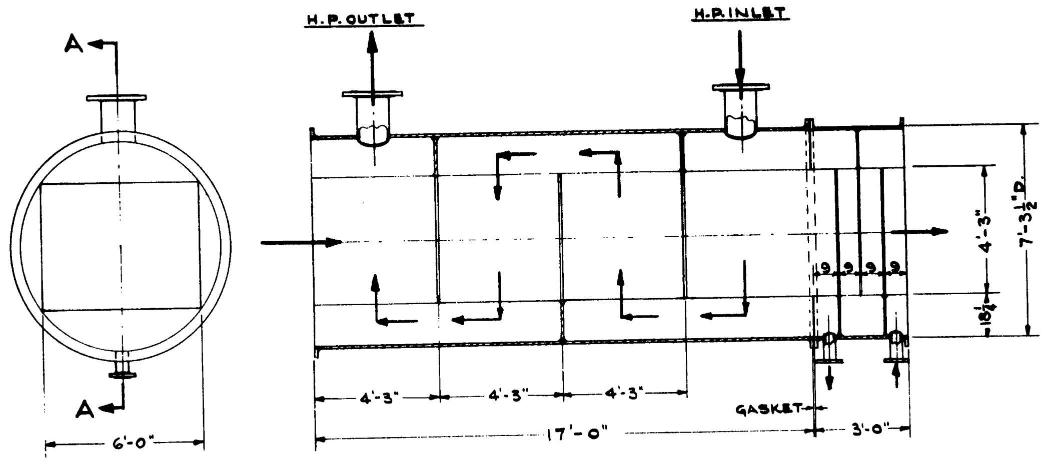




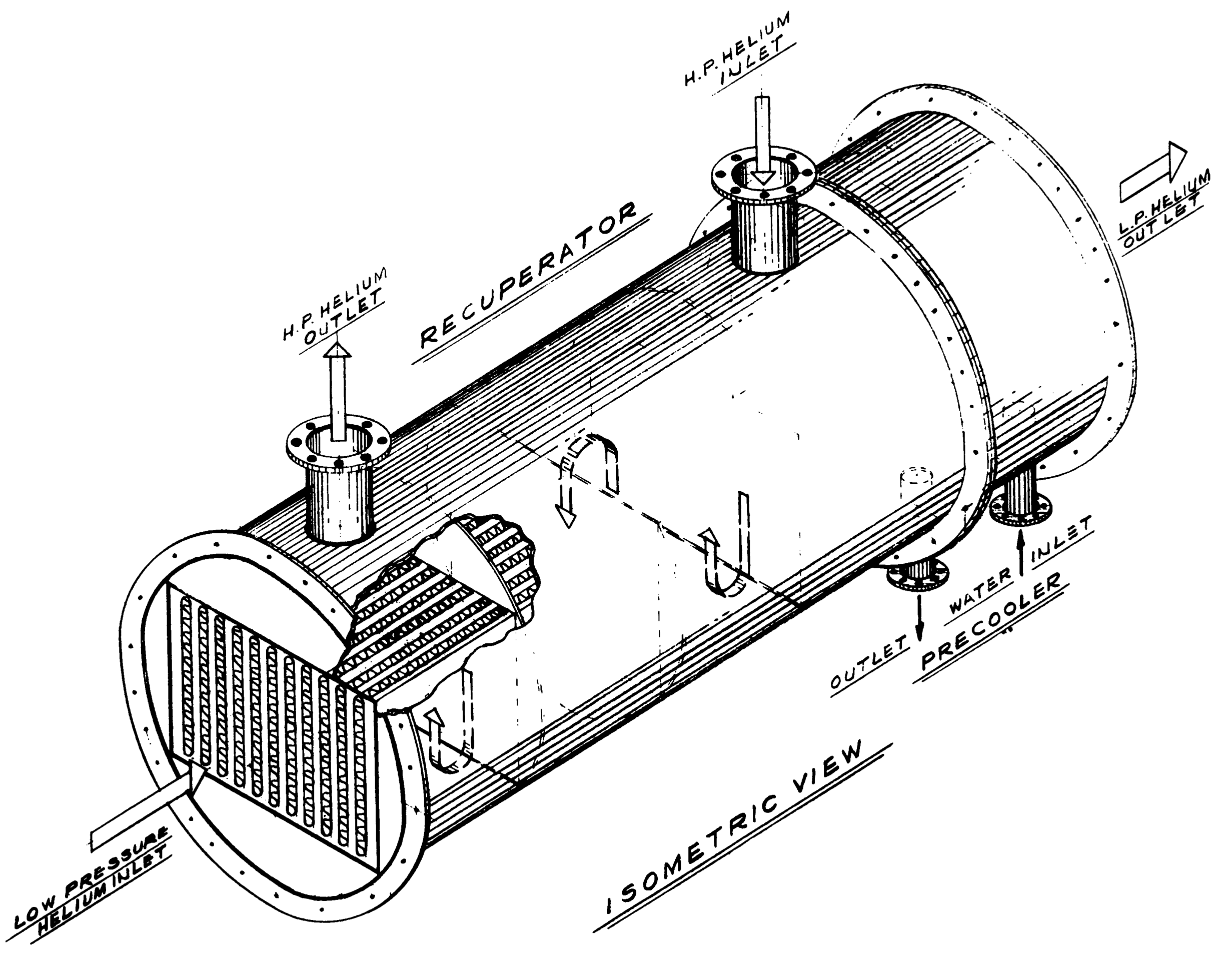
FIN MATERIAL	SA-204 CARBON 1/2 MOLY.	
FIN THICKNESS	.0145	IN.
FIN HEIGHT, EFFECTIVE	.1238	IN.
℄ TO ℄ FLAT PLATE	.2655	IN.
PLATE THICKNESS	.022	IN.
FIN PITCH TRANSVERSE TO FLOW	.1885	IN.
FIN PITCH PARALLEL TO FLOW	5.00	IN.
FIN SURFACE / TOTAL SURFACE	.7974	
EQUIVALENT HYDRAULIC DIA.	.125	IN.

FIG. 16.

GRISCOM RUSSELL PLATE FIN HEAT TRANSFER SURFACE



SECTION A-A



ISOMETRIC VIEW

FIG. 17.
ARRANGEMENT & DETAILS
RECUPERATOR - PRECOOLER

TABLE V

Recuperator Specifications

Helium flow rate, #/sec	214	
Duty, BTU/Hr (Total 2 units)	643×10^6	
Effectiveness, %	92.3	
Service	HP	LP
Temperature in, °F	210	937
Temperature out, °F	881	266
ε , %	1.5	2.25
Face area/unit, ft^2	25.5	25
Number of passes	4	1
Flow length (total), ft	16.8	17
Surface characteristics	see Figure 17	
Total surface (LP)/unit, ft^2	61,500	
Prime surface rate, BTU/Hr/ft^2	25,800	
Ft^2 prime surface /KW	0.415	

with a special nickel brazing compound. The corrugated fin material is contained in flattened tubes which are formed by bending sheet metal to the desired shape and heliarc welding the seam at the middle of the circular edge of the tube. After this welding is done, the corrugations are slipped into the tube and corrugations are placed across the tube on the outside. The tubes are then heliarc welded to the specially formed sheets. When the whole structure is assembled, it is coated with brazing compound and brazed in a reducing atmosphere. The brazing compound gives an impervious coating of nickel alloy on all surfaces and adds considerably to the high temperature oxidation resistance of the unit.

Design of the recuperator is based on test work performed by the Griscom-Russell Co. Their test work includes studies of the low temperature performance of various fin geometries over a range of Reynolds Numbers; studies of the physical properties of the brazed joint including tensile, bend and fatigue tests; and high temperature performance of completed cells. These latter tests have demonstrated the adequacy of the design at conditions comparable to those of the present application.

The Griscom-Russell Co. is presently in the process of manufacture of heat exchangers of this general type on a production line basis.

The precooler, a water-cooled heat exchanger which cools the low pressure helium to the compressor inlet temperature is simply a continuation of the recuperator with water replacing the high pressure helium. The precooler section is 4'2" x 6' x 3' long. The LP helium flows directly from the recuperator through the precooler to the compressor. The cooling water flows in four-pass cross-counterflow through the unit. Figure 17 shows the recuperator and precooler exchangers as a unit. Table VI lists the precooler specifications.

The intercooler is a conventional tube in shell heat exchanger with a folded tube bundle and a divided tube sheet. The bundle is made up of 5/8" tubes on 7/8" equilateral triangular centers. Table VII lists the intercooler specifications. The partially compressed helium flows through the tubes, and water flows across the tubes in the shell.

The pressure loss in the intercooler is slightly greater than that allowed in Table I, however it does not significantly affect the cycle analysis.

TABLE VI

Precooler Specifications

Helium flow rate, #/sec	214
Duty, BTU/Hr (Total 2 units)	169×10^6
Effectiveness, %	92.2
Surface characteristics	see Figure 17
Total surface, (He)/unit, ft^2	11,000
Prime surface rate, BTU/Hr/ft^2	37,700
Ft^2 prime surface/KW	0.745

Service	Helium	H_2O
Temperature in, °F	266	75
Temperature out, °F	90	125
ε, %	2.25	
Face area/unit, ft^2	25	4.5
Number of passes	1	4
Flow length (total), ft	3	16.8

TABLE VII

Intercooler Specifications

Tube in Shell Exchanger - Helium in Tubes, H_2O in Shell

Helium flow rate, #/sec	214	
Duty, BTU/Hr	115×10^6	
Effectiveness, %	88.8	
Total surface, ft^2	9170	
Prime surface rate, BTU/Hr/ft^2	12,550	
Ft^2 prime surface/KW	.153	
Number of tubes/pass	3020	
Length of tubes, ft	9.25	
Tube, diameter-gage	5/8" - 16 BWG	
Tube pitch, inches	7/8	
Service	Helium	H_2O
Temperature in, °F	210	75
Temperature out, °F	90	125
ϵ , %	1.07	
Number of passes	2	2
Flow length (total) ft	18.5	18.5

7.0 General Arrangement

A general arrangement of the machinery set and heat transfer apparatus is shown in Figure 18.

The high speed compressor and turbine set and the low speed power turbine are arranged in a single case, split on the horizontal centerline. The high speed shaft, consisting of the two compressor stages and the high pressure turbine, is mounted on three bearings, one at each end of the shaft and one between the low and high pressure compressor. This latter is a combined thrust and journal bearing. The low pressure power turbine is located downstream from the high pressure turbine and it is supported on two bearings, the one at the turbine exhaust being a combination thrust and journal bearing.

Viscosity or contact type seals are contemplated at the shaft ends in the casing to prevent the escape of helium from the system. Seals of this type are currently being used in compressors handling hydrocarbon gases at pressures far in excess of the pressure levels encountered at the shaft ends of this power plant.

Current requirements for the handling of gases in turbomachinery in the process industries have revived interest in hydrostatic or the so-called "piston bearings", lubricated by the fluid being pumped. An intensive program for the development of this type of bearing for use in closed cycle gas turbines is being formulated and if the development of this bearing type is successful, it will result in a complete reappraisal

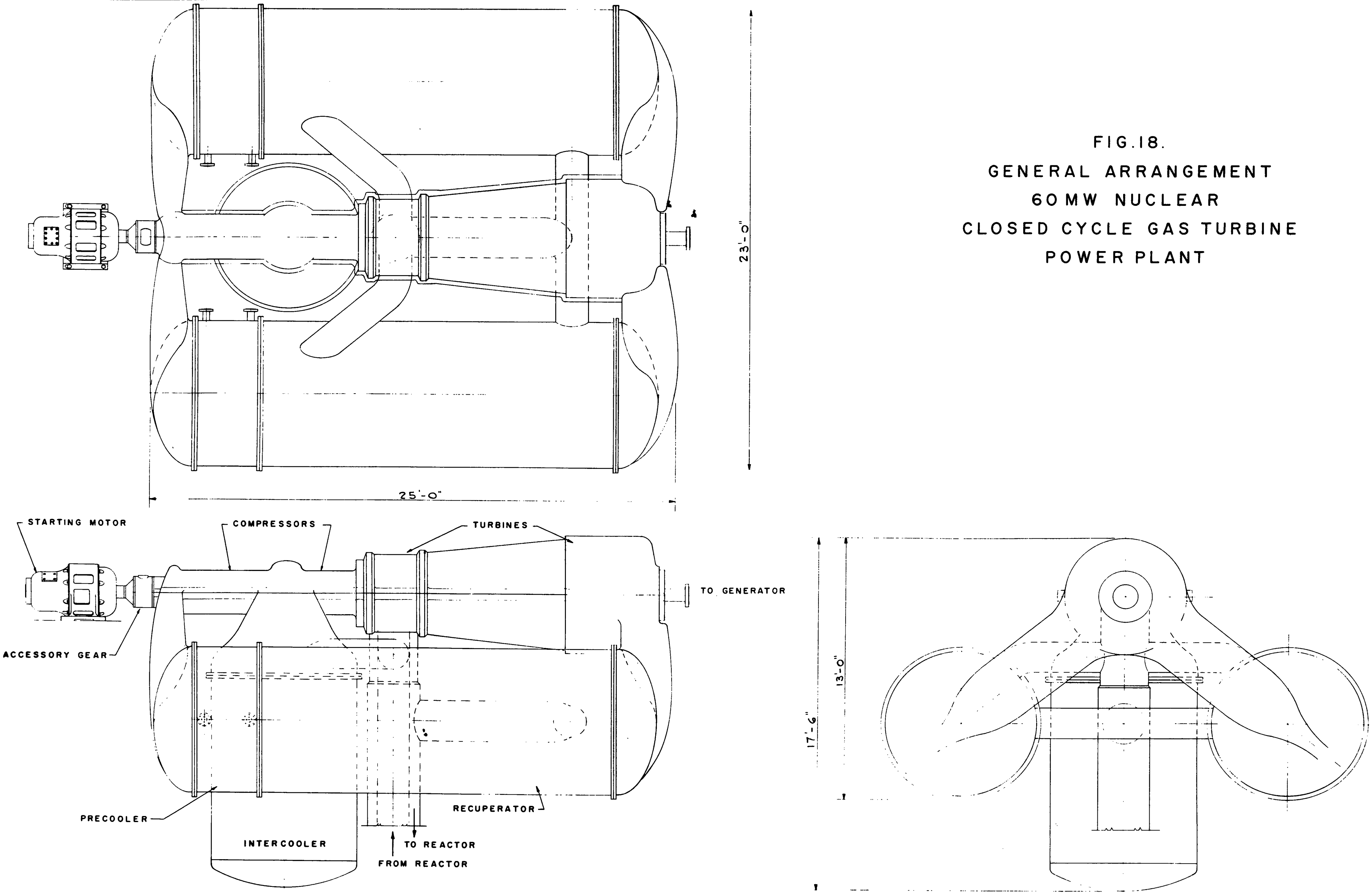


FIG. 18.
GENERAL ARRANGEMENT
60 MW NUCLEAR
CLOSED CYCLE GAS TURBINE
POWER PLANT

of the bearing and sealing system as the use of hydrocarbon lubricants will be dispensed with.

The intercooler, between compressor stages, is so designed as to be an integral part of the machinery set, thus eliminating the necessity for external lines and their associated joints which are a source of leakage. The intercooler is a hairpin tube type with helium in the tubes and water in the shell. The tube sheet, to which the tubes are welded, is bolted directly to a mating flange on the bottom of the machinery set casing between the low and high pressure compressors. By suitable diffusers the helium is conducted from the discharge of the low pressure compressor, through the intercooler and back to the high pressure compressor inlet.

A cross section through the machinery set and intercooler is shown in Figure 19.

Through a system of internal ducting, helium leaving the compressor flows in an annular passage around the high pressure turbine inlet pipe. By this means, openings in the machinery set casing are reduced to a minimum.

This pipe, illustrated schematically in section in Figure 19 employs inside out insulation, a method used on all closed cycle gas turbine plants, the use of which permits the outer pipe which is subjected to pressure only, to be fabricated of normal low alloy materials. The inner pipe conducting the high temperature gas stream is unstressed and

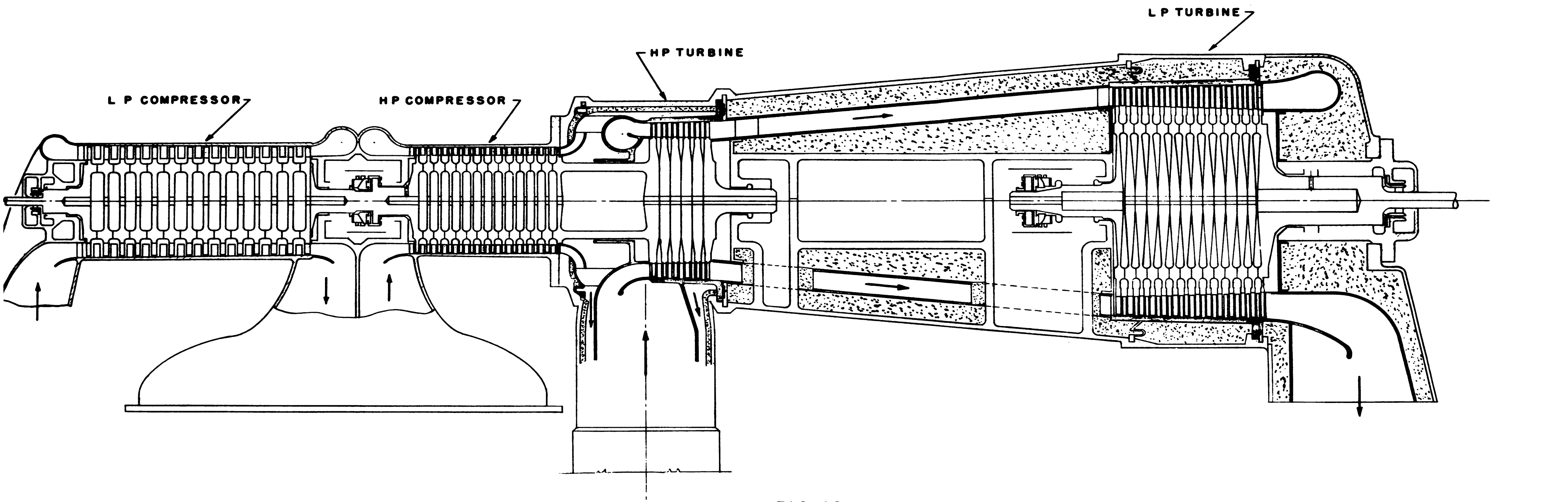


FIG. 19
SECTION THROUGH
MACHINERY SET

the material need only be selected for its corrosion resistant properties at the temperature in question.

8.0 Control System

The subject plant is designed for base load operation on a network where the frequency (speed) control is from another source. Therefore only power output (pressure level) control is required, together with a means of synchronizing the speed of the output turbine to the reference source at the time of starting. Power, or pressure level control, is effected by addition or withdrawal of working fluid from the circuit. Emergency speed control of the power turbine is effected by bypassing.

Helium that is not being circulated in the plant is stored in accumulators for subsequent use, i.e. this is a no-loss system. The accumulator system consists of two (or two groups of) storage bottles; one being the receiver and the other the accumulator interconnected by a transfer pump. In this system the total amount of helium in the power plant and tanks is constant at all times. Leakage losses are made up by addition of helium to the receiver from time to time as required. A simplified diagram of this system is shown in Figure 3.

Over the range of power output that this plant will operate, assumed to be $\pm$ 10% of normal, power output varies directly with system pressure. Therefore, the power output control consists of a set of variable datum pressure reducing valves between the system and the receiver and accumulator tanks, designed to maintain a constant preselected pressure level in the system.

An overspeed governor is provided on both the high pressure compressor/turbine set and the power turbine. The governor on the compressor/turbine set is a top speed governor only, tripping a compressor bypass valve when this set exceeds a predetermined speed limit. The governor on the power turbine is designed to come into play only in event of an emergency, the presence of which makes it necessary to shut the plant down. In action, the power turbine governor opens the power turbine bypass valve, immediately reducing the weight flow through the power turbine. Since this reduces the back pressure on the compressor drive turbine, it tends to overspeed, thus activating the compressor bypass valve. Further, the power turbine overspeed governor trips the system pressure regulator, resulting in the discharge of the contents of the system to the receiver. Simultaneously, the control rods are dropped into the reactor reducing the heat input to the system.

When the power plant load is dumped and the reactor activity level reduced, a means must be provided to cool the reactor for a period after shut-down. During the normal procedure of shutting off the plant and the emergency condition previously discussed, the compressor/turbine set will circulate helium through the reactor until the minimum self-running speed is reached. At that time, a secondary motor-driven circulating compressor with an auxiliary cooling loop is energized, circulating helium through the reactor until activity is reduced to a point resulting in a safe temperature level.

9.0 Xenon Removal

The working fluid used in this plant is helium which is available commercially at a purity of 99.99%. Impurities consist of argon, carbon dioxide and nitrogen, none of which is in sufficient quantity to be of any concern. There is, however, the possibility of contamination of the system by gaseous fission products, escaping from the reactor fuel elements. The principle volatile radioactive impurity of the fission process is xenon, and removal of the xenon from the working fluid is desirable to reduce the activity of the working fluid to an acceptable level and avoid the necessity of extensive shielding.

The xenon can be effectively removed to any degree desired by solidification in a cold-trap. One procedure for accomplishing this would be to withdraw a small stream of helium from the cold end of the compressor intercooler and pass it through a heat exchanger in which it would be cooled to whatever temperature would be necessary to reduce the xenon content to a permissible level. Since the xenon is present in such small amounts, even its complete removal would leave the helium essentially undiminished in amount. This cold helium stream would pass through a turbo-expander wherein its pressure would be dropped to essentially the suction pressure of the compressor. In passing through the expander, the helium would be cooled sufficiently so that it could act as the refrigerant for cooling the xenon cold-trap exchanger. A typical flow diagram for this cold-trap system is shown in Figure 20.

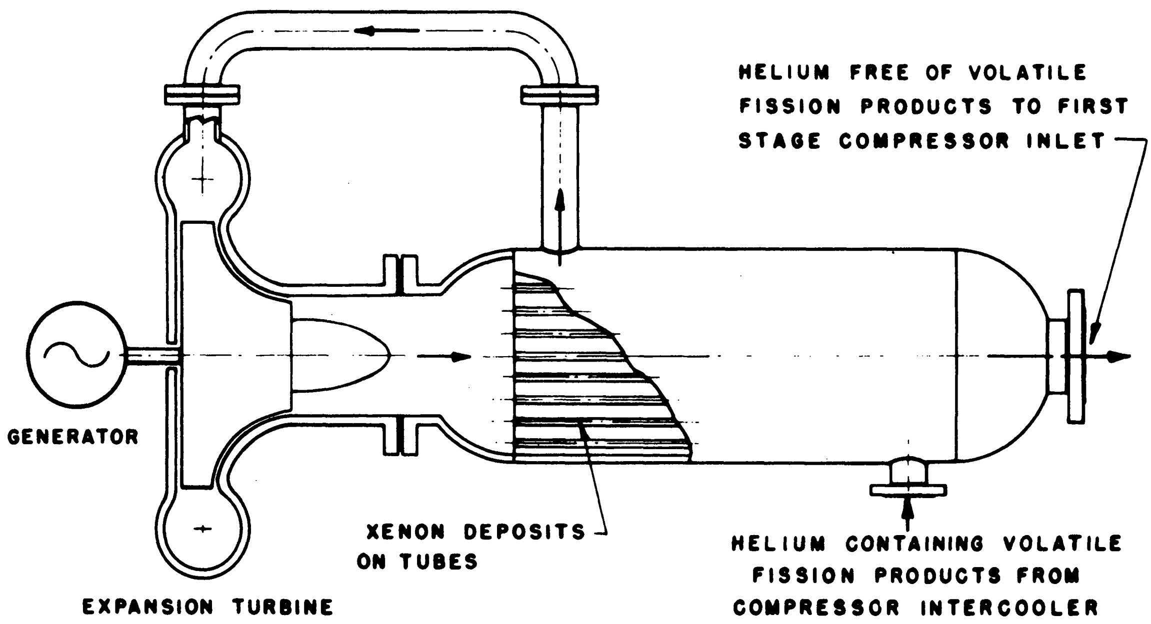


Figure 20. Xenon trap.

Since the gas flow required to hold down the xenon concentration in the working fluid of the power cycle is very small, estimated at 1% of the mass flow or about two pounds per second, the dimensions of the cold-trap heat exchanger will be small, as will be the turbo-expander required to provide the cold end ΔT. The passages of the cold-trap will gradually become plugged with xenon and its decay products and eventually it will have to be replaced by a new unit. The size of this cold trap is such that it can be cleaned or disposed of, depending upon which seems to be most desirable in the final design.

APPENDIX I

Cycle Analysis

Assumptions -

Pressure ratio	1.55^2	=	2.40
Pressure losses	$\Sigma\epsilon$	=	7%
Expansion ratio		=	2.233
Compressor inlet temperature		=	90°F
Turbine inlet temperature		=	1400°F
Recuperator effectiveness		=	93%

Compressor -

T_1	=	550°R
R_c	=	1.55:1
$R_c^{\frac{\gamma-1}{\gamma}}$	=	1.191
$1 - R_c^{\frac{\gamma-1}{\gamma}}$	=	.191
ΔT_{ad}	=	105°F
η_c	=	.88
ΔT	=	120°F
T_2	=	670°R
T_3	=	550°R
T_4	=	670°R
$\Delta T_c = 2 \times 120$	=	240°F

HP Turbine -

T_6		$= 1860°R$
ΔT_c		$= 240°F$
T_7		$= 1620°R$
η_e		$= .888$
ΔT_{ead}	$= 240/.888$	$= 270°F$
T_7'		$= 1590°R$
Θ	$= 270/1860$	$= .145$
$1 - \Theta$		$= .855$
$(1 - \Theta)^{.398}$		$= .675$
R_c	$= 1/(1-\Theta)^{2.51}$	$= 1.48:1$

LP Turbine -

Re_{tot}		$= 2.233$
Re_{LP}	$= 2.233/1.48$	$= 1.51:1$
$1/Re_{LP}$		$= .662$
$(1/Re_{LP})^{.398}$		$= .848$
$1 - (1/Re_{LP})^{.398}$		$= .152$
T_7		$= 1620°R$
ΔT_{ad}		$= 246°F$
η_e		$= .888$
ΔT_w		$= 218°F$
T_8		$= 1402°R$

Recuperator -

T_8		$= 1402°R$
T_4		$= 670°R$
δt		$= 732°F$
η_r		$= 92.3\%$
ΔT	$= 734/13$	$= 56°F$
T_5	$= 1402 - 56$	$= 1346°R$

Reactor -

T_6	$= 1860°R$
T_5	$= 1346°R$
ΔT_R	$= 514°F$

Precooler -

T_4	$= 670°R$
ΔT_p	$= 56°F$
T_9	$= 726°R$
T_1	$= 550°R$
δt	$= 176°F$

Cycle Efficiency -

ΔT_W	$= 218°F$
ΔT_R	$= 514°F$
η_c	$= 42.4\%$

Work Rate -

$W = 3415/\Delta T_W \times 1.25 = 12.5\#/KW\ Hr$

Terminal Conditions & Weight Flow -

Generator efficiency = 97.5%

Mechanical efficiency = 97.5%

Net output = 60 MW

Gross output = 63 MW

Helium Flow = 12.5x63 = 790,000#/Hr

Reactor Load = $\frac{790{,}000 \times 514 \times 1.25}{3415}$ = 148.5 MW

Terminal efficiency = 60/148.5 = 40.4%

Plant heat rate = $\frac{790{,}000 \times 514 \times 1.25}{60{,}000}$ = 8460 BTU/KW Hr

The effect of changes in cooling water temperature and overall relative pressure loss in cycle efficiency and work rate are shown in Figure 21.

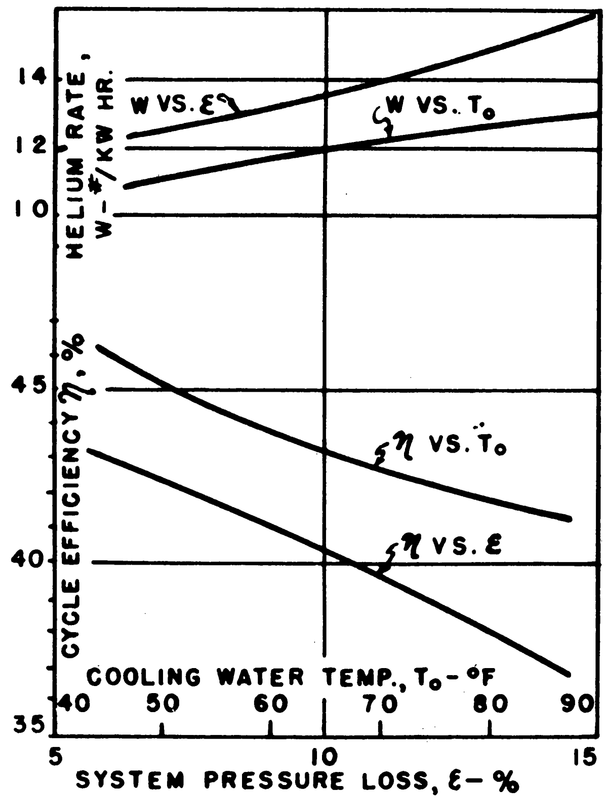


Figure 21.
Correction for off-design condition.