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Preliminary Design of the Cooling System for a Gas-Cooled, High-Fluence Fast Pulsed Reactor (HFFPR)

Henry C. Monteith

Sandia Laboratories

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PRELIMINARY DESIGN OF THE COOLING SYSTEM FOR A GAS-COOLED, HIGH-FLUENCE FAST PULSED REACTOR (HFFPR)

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ABSTRACT

The High-Fluence Fast Pulsed Reactor (HFFPR) is a research reactor concept currently being evaluated as a source for weapon effects experimentation and advanced reactor safety experiments. One of the designs under consideration is a gas-cooled design for testing large-scale weapon hardware or large bundles of full-length, fast reactor fuel pins. This report describes a conceptual cooling system design for such a reactor. The primary coolant would be helium and the secondary coolant would be water. The size of the helium-to-water heat exchanger and the water-to-water heat exchanger will be on the order of 0.9 metre (3 feet) in diameter and 3 metres (10 feet) in length. Analysis indicates that the entire cooling system will easily fit into the existing Sandia Engineering Reactor Facility (SERF) building. The alloy Incoloy 800H appears to be the best candidate for the tube material in the helium-to-water heat exchanger. Type 316 stainless steel has been recommended for the shell of this heat exchanger. Estimates place the cost of the helium-to-water heat exchanger at approximately \$100,000, the water-to-water heat exchanger at approximately \$25,000, and the helium pump at approximately \$450,000. The overall cost of the cooling system will approach \$2 million.

ACKNOWLEDGMENTS

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PRELIMINARY DESIGN OF THE COOLING SYSTEM FOR A GAS-COOLED, HIGH-FLUENCE FAST PULSED REACTOR (HFFPR)

Introduction

As a part of the Sandia Laboratories continuing effort toward reactor design and development in support of Laboratories programs, a conceptual design has been developed for a High-Fluence Fast Pulsed Reactor (HFFPR).

The work described in this report was undertaken for the following reasons:

1. To perform a preliminary linear analysis and to estimate the nominal design parameters for the major system components necessary to the construction of a cooling system for the HFFPR,
2. To estimate the space which will be required to house the components of the HFFPR,
3. To determine the approximate cost of the major cooling system components in 1977 dollars, and
4. To evaluate the practicality of placing the HFFPR system in the Sandia Engineering Reactor Facility (SERF) building.

Preliminary Analysis

The preliminary analysis and associated research performed on the HFFPR produced some interesting results. These results are briefly discussed in the subsections which follow.

Technical Feasibility of a Gas-Cooled HFFPR

The technology of high-temperature, gas-cooled systems has been adequately demonstrated in the following instances:

1. The Peach Bottom prototype HTGR power plant,
2. The Dragon Reactor experiment, and
3. The Los Alamos Ultra High Temperature Reactor Experiment (UHTREX).

The characteristics of these reactors and their relationship to the design of the HFFPR are discussed in Appendix A.

Determination of the Nominal Design Parameters of the Cooling System

The basic cooling system design consists of the following two heat exchangers:

1. Exchanger 1 in which heat is transferred from the primary helium loop to the intermediate water loop, and

2. Exchanger 2 in which heat is transferred from the intermediate water loop to the cooling tower.

Since the possibility exists that Exchanger 1 could become contaminated, Exchanger 2 serves to isolate the cooling tower from Exchanger 1. A linear mathematical analysis of the cooling system, as shown in Figure 1, is contained in Appendix B.

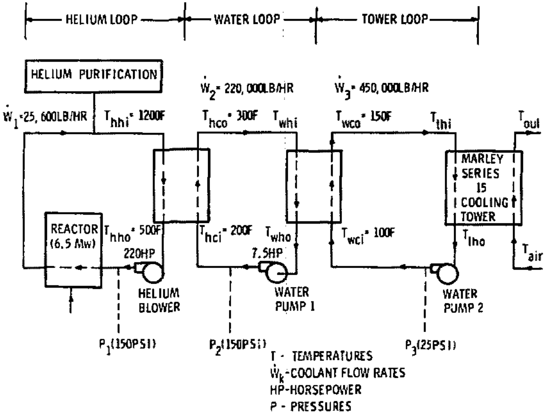


Figure 1. Cooling System Block Diagram

Heat Exchangers and Pumps

A thorough analysis of the heat exchangers is contained in Appendix C. Because of its simplicity and ability to withstand thermal stresses, a helium-to-water heat exchanger of the U-tube type was used. The structural design guide for the helium heat exchanger is Code Case 1592 (Class 1 components in elevated temperature service) found in Section III of the ASME Boiler and Pressure Vessel Code.

An analysis of the water heat exchanger is also contained in Appendix C. The mass flow rates through both heat exchangers are indicated in Figure 1. The results of the linear analysis, as well as the results of analyses submitted by different manufacturers, indicate that the size of both heat exchangers will be on the order of 0.9 metre (3 feet) in diameter and 3 metres (10 feet) in length.

One of the manufacturers, Mechanical Technology Incorporated (MTI), has considerable experience in the manufacture of helium pumps. MTI manufactured the helium blowers for the UHTREX reactor and is currently designing similar blowers for Union Carbide and Oak Ridge National Laboratory (ORNL).

The circulator can easily handle 3.3 kg/s (26,000 lb/h) of helium at 150 psia and 315° C (600° F) with a head rise between 17 kPa and 35 kPa (2.5 and 5 psi). Approximate power would be 82 kW (110 hp) at 17 kPa (2.5 psi) head and 164 kW (220 hp) at 35 kPa (5 psi) head. Water pump 1, which is located between the two heat exchangers, will operate at a speed of 1750 rpm and will have a power of about 5.6 kW (7.5 hp). Water pump 2, which serves the cooling tower, will have a power of about 37 kW (50 hp) and run at 1750 rpm. Further details on the pumps are given in Appendix D.

The Cooling Tower

The cooling tower is a Marley Series 15 forced draught tower. It is able to handle 56.7 kg/s (450,000 lb/h) of water and drop the water temperature from 65° C to 38° C (150° F to 100° F). It will cover an area 6.1 by 4.9 metres (20 by 16 feet) and will stand 5.2 metres (17 feet) high. The fan will be powered by a 15 kW (20 hp) motor. Appendix E presents a schematic of the cooling tower.

The SERF Reactor Vessel as a Component of the HFFPR

A preliminary survey of the SERF reactor vessel was made to determine if it might be used with the HFFPR. Analysis indicated that the design stresses of this vessel will be exceeded if it is subjected to temperatures on the order of 649° C (1200° F).

The vessel is locked into strong concrete and very little room is allowed for expansion; consequently, it should not be subjected to a temperature higher than 177° C (350° F). If a design scheme can be devised in which the upper vessel is retained and is not subjected to temperatures greater than 177° C (350° F), then the new lower vessel should be designed to avoid transmitting high stresses to the upper vessel.

It may be possible to devise a scheme by which the existing water cooling system for the vessel can be modified to keep its temperature below 177° C (350° F). If this is possible, then the vessel may prove usable for the HFFPR. Further details are provided in Appendix F.

HFFPR System Layout in the SERF Building

The suggested location for the helium-to-water exchanger is the beam room which is situated north of the irradiation cell and on the same level. In this location, it will be adequately shielded and at the same time it will be in close proximity to the reactor for ease of operation. The secondary heat exchanger may be placed in the pump house along with the two water pumps. (The pump house is a new addition to the SERF building.) Finally, the helium purification system should be constructed as a modular unit and placed in Lock No. 1 after it has been provided with a gas-tight liner. Lock No. 1 is located west of the irradiation cell and on the same level. Further details may be obtained from Appendix G.

Cooling System Cost Estimates

Estimated costs for all the major components in the cooling system are summarized in Table I.

TABLE I

Estimated Costs for the HFFPR Cooling System

Component	Cost
Helium heat exchanger	$100,000
Water heat exchanger	$ 25,000
Helium pump	$450,000
Helium pump controller	$ 75,000
Water pump 1	$ 1,700
Water pump 2	$ 3,000
Helium Purification	$250,000
Total	$904,700

If duplicate pumps are to be utilized in case of pump failure, an additional $455,000 must be added to the calculated total. If an extra $500,000 for the purchase of other equipment is also added in, the entire cooling system could cost as much as $2 million for hardware alone.

APPENDIX A

Technical Feasibility of a Gas-Cooled HFFPR

The technology of the high-temperature, gas-cooled reactor is sufficiently advanced to make it a viable option for the proposed HFFPR. Information from the following nuclear reactor demonstration projects suggested design criteria for the HFFPR:

1. The Peach Bottom prototype HTGR power plant,
2. The Dragon Reactor experiment,
3. The Los Alamos UHTREX, and
4. The General Atomic gas-cooled breeder reactor design.

The main design parameters for the Peach Bottom HTGR are listed in Table A-I.[1-3] The heat exchangers for this reactor had helium in the primary circuit and water in the secondary circuit. Helium was allowed to flow around the outside of the heat exchanger tubes while water flowed inside the tubes. The same type of heat exchanger concept is suggested for the gas-cooled HFFPR. High temperatures, helium leakage, and limited space impose strict design criteria upon the heat exchangers; however, these criteria were adequately met during the design of the Peach Bottom HTGR. Even though this reactor generated 115 MW(t), the steam generators were only 8.6 metres (28 feet, 4 inches) long and 2.3 metres (7.5 feet) in diameter. Since the proposed HFFPR will generate only 6.5 MW(t), it is reasonable to assume that the necessary heat exchangers can be accommodated in the SERF structure since the size of the heat exchanger is proportional to the heat energy it must transfer. Thermal stresses and material demands are also expected to be less than those encountered in the Peach Bottom HTGR; consequently, the cooling system components might be more readily available and less expensive.

Another reactor design which had some characteristics in common with the proposed HFFPR was the Dragon Reactor experiment.[4-6] Some characteristics of this reactor are presented in Table A-II. The helium temperatures for this reactor are very close to those temperatures proposed for the HFFPR. The heat exchangers of this reactor were designed to a high standard of leak tightness in order to prevent the escape of helium which could also carry some decay products.

The heat removal system for the Dragon Reactor has many characteristics in common with the cooling system proposed for the HFFPR. In the Dragon reactor, two heat exchangers were used in each coolant loop and the generated heat was dissipated through a cooling tower. In the HFFPR, a single loop with two heat

TABLE A-I

Main Design Parameters of the Peach Bottom HTGR

Reactor heat output	115.3 MW(t)
Plant net power	40 MW(e)
Core active dimensions	2.8 metres (9.16 feet) in diameter and 2.3 metres (7.5 feet) high
Fuel elements	804 (U, Th)C_2 fueled, graphite clad; (U + Th) 1 atom percent, 20 mass percent in compacts
Core fuel initial loading	200 kg_m U^{235}, 13 kg_m U^{238}, 1897 kg_m Th
Heat flux	25 kW/m^2 (8000 Btu/$ft^2 \cdot h$) average, 322 kW/m^2 (102,000 Btu/$ft^2 \cdot h$) peak
Power density, kW/liter	8.24
Specific power	580 kW(t)/kg_m U^{235}
Average conversion ratio	0.50
Excess reactivity	19 percent cold clean, 8 percent hot clean, 5 percent hot poisoned
Control rods	36 B_4C rods
Coolant	He
Coolant conditions	334° C (634° F) reactor inlet, 734° C (1354° F) outlet, 335 psig vessel pressure, 69 kPa (10 psi) rise in circulators, 55.4 kg/s (439,600 pound-mass/h) total flow rate, 0.12 kg/s (1000 pound-mass/h) purge rate.
Circulators	Two 16m^3/s (33,800 ft^3/min) horizontal single-stage centrifugal, 1.9 MW (2500-hp) motors each
Steam-cycle conditions	538° C (1000° F), 1450 psig at turbine throttle 218° C (425° F) economizer inlet
Steam-cycle equipment	Two forced recirculation steam generators, one tandem-compound, double-flow 46-MW(e) gross turbine, one single-pass divided box condenser, three horizontal U-tube feedwater heaters
Plant efficiency	39.9 percent gross, 34.7 percent net

TABLE A-II

Dragon Reactor Experiment: General Data

Thermal output	20 MW
Helium coolant pressure	2.03 MPa (20 atm or 294 lb/in^2)
Inlet coolant temperature	350° C (662° F)
Outlet coolant temperature	750° C (1382° F)
Mean power density of core	14 MW/m^3
Cooling channel voidage	13 percent
Mean surface heat flux over core	24 W/cm^2
Core length	1.6 metres (5 feet, 3 inches)
Core diameter	1.07 metres (3 feet, 6 inches)
Fuel element length	2.54 metres (8 feet, 4 inches)
Fuel rods on 6.35-cm (2.5-inch) triangular pitch	259
Fuel elements, each of which consists of a cluster of seven fuel rods	37
Reflector length	2.45 metres (8 feet, 0.5 inch)
Reflector diameter	2.89 metres (9 feet, 6 inch
Number of control rods in reflector	24
Diameter of control rod pitch circle	1.23 metres (4 feet, 0.5 inch)
Pressure vessel diameter around core	3.5 metres (11 feet, 6 inches)
Pressure vessel thickness around core	5 cm (2 inches)

exchangers is planned. The Dragon reactor cooling circuit is illustrated in Figure A-1. In this circuit, each primary heat exchanger transfers its heat to a secondary circuit loop of the forced circulation boiling type. Water is partially evaporated in the primary heat exchangers and then the steam/water mixture is carried to the secondary heat exchangers where it is condensed and recirculated by rotor pumps. Natural circulation is used to transfer shutdown heat.

The UHTREX was built by LASL in order to advance the technology of gas-cooled reactors.[7] A helium-cooled gas reactor was designed to operate at very high temperatures. Some of the design characteristics of this reactor are given in Table A-III. In this reactor, helium, which was the only coolant utilized, was passed through the cooling tower. A study of this system gives valuable insight into design methods for the handling of high temperatures, helium purification, and helium management.

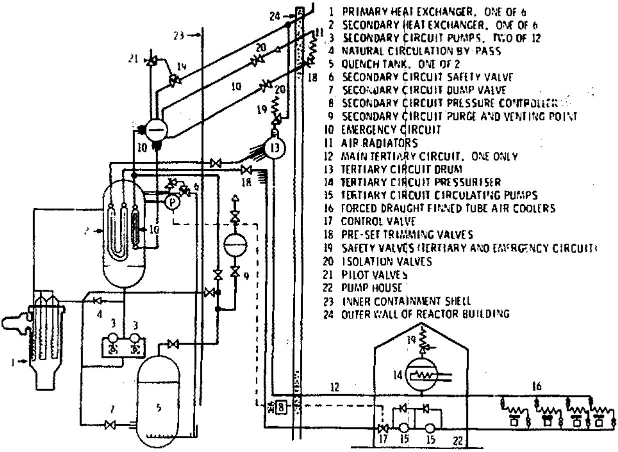


Figure A-1. Heat Disposal Circuit for the Dragon Reactor

TABLE A-III

UHTREX Design Parameters

Nominal reactor power	3 MW(t)
Coolant mass flow rate	1.25 kg/s (10,250 lb/h)
Coolant pressure	3.26 MPa (473.5 psi)
Outlet temperature	1316° C (2400° F)
Inlet temperature	871° C (1600° F)
Number of assemblies	312
Critical mass	5.68 kg, 93.6 percent enriched uranium
Core loading at design power	11.0 kg, 93.6 percent enriched uranium
Average specific power in fuel	270 kW/kg
Average power density in core	1.4 kW/liter
Burnup	10 percent to 50 percent

The design criteria which have been presented for the gas-cooled fast breeder reactor also have useful implications for the HFFPR. This demonstration facility was planned to operate at 330 MW(e).[8] The fuel rods consist of annular $(Pu\text{-}U)O_2$ pellets within a type 316 stainless steel cladding. The cladding, which is approximately 0.5 mm (20 mils) thick, is designed to handle a maximum temperature of 700° C (1292° F) at midthickness of the fuel cladding (including hot-spot factors). The design of the gas-cooled fast breeder reactor indicated that type 316 stainless steel is a feasible choice for the cladding in the HFFPR as well as in the heat exchangers.[9]

A 1000 MW(e) version of the gas-cooled fast breeder reactor has also been planned.[10] It is to have a helium pressure of about 8.6 MPa (1250 psi), a reactor inlet temperature of around 290° C (554° F), and a hot spot midclad temperature of 760° C (1400° F).

All of the design concepts which will be necessary for the HFFPR have been tested in the above projects.

APPENDIX B

HFFPR Heat Exchanger Design

The basic block diagram for the cooling system which is being proposed for the HFFPR is illustrated in Figure B-1. The heat exchangers are fundamental to the cooling system and very careful attention should be given to their design. Indeed, the proper design of the heat exchangers is extremely important to reactor operation and safety.

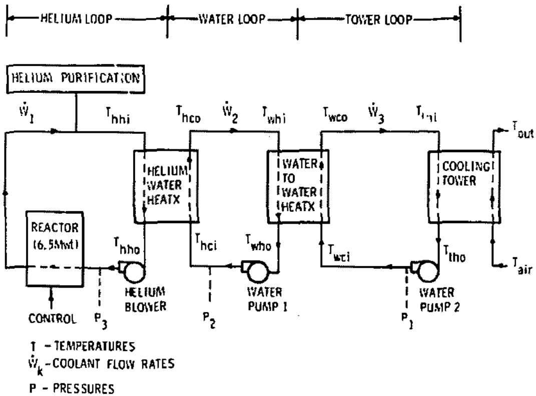


Figure B-1. Cooling System Block Diagram for HFFPR

The design of a heat exchanger is largely constrained by the temperatures at which it must operate, the amount of space available for mounting it, and the quantity of pressure it is allowed to drop. Figure H-1 shows that for a given design power level of operation and a specified outlet temperature, the volume of the heat exchanger is influenced by the temperature drop across its primary. The size of the heat exchanger will also decrease as the pressure it is allowed to drop increases.[11]

The integral and peripheral components of a high-temperature, gas-cooled reactor must be designed to a very high level of reliability. This makes it essential that materials for these components be carefully selected.

The choice of the materials to be used in the heat exchangers is dictated by the following considerations:

1. Strong mechanical properties and metallurgical stability at operating temperature,
2. Corrosion resistance against primary and secondary media under normal as well as abnormal operating conditions,
3. Ease of fabrication, e.g., in welding and bending,
4. Conformity with the necessary codes, such as the ASME and TEMA codes,
5. Practical operating experience,
6. Availability of materials, and
7. Cost of materials.

Fatigue strength is a key mechanical property of the material in the heat exchanger tubes because the tubes are subjected to thermal stresses associated with the radial temperature gradient through the tube walls. Therefore, it is necessary to consider the creep strength, yield strength, ultimate tensile strength, and fracture toughness of the tube material.

Some of the alloys which have been considered as possible candidates for the tubing material in the primary helium-to-water heat exchanger are Incoloy 800H, Incoloy 802, Inconel 601, Inconel 617, Hastelloy S, Hastelloy X, and Hastelloy C. Out of all these possibilities, Incoloy 800H is felt to be the best choice for the heat exchanger tubing material due to the following considerations:[12]

1. It has very high creep strength at 704° C (1300° F),
2. It has a low aluminum content (this is desirable because aluminum can contribute to helium reactor coolant corrosion),
3. It has excellent thermal stability,
4. It is free from cobalt and tantalum which can cause potential radioactivity problems,
5. It is readily available,
6. It has good fabricability,
7. It has the lowest cost of the candidate alloys, i.e., approximately $164 per metre ($50 per foot), and
8. It has been certified for use with gas-cooled reactors and approved under the ASME Boiler and Pressure Vessel Code.

It is recommended that the shell of the heat exchanger be an all-welded pressure boundary of corrosion-resistant material such as stabilized stainless steel. This shell should be made extra thick as a precaution against helium leakage. Calculations indicate that the helium-to-water heat exchanger will be on the order of 0.9 metres (3 feet) in diameter and 3 metres (10 feet) in length. The shell will have to support the gas pressure which is assumed to be 1.03 MPa

(150 psi). It is suggested that the certified material (type 316 steel) be used as the shell material.

The thickness of the shell will be estimated based on a life time of 100,000 operational hours. At a temperature of 704° C (1300° F), a hoop tension of 27.6 MPa (4000 psi) will cause a creep rate of 0.00001 percent per hour. The circumference of the heat exchanger shell will be 2.87 metres (9 feet, 5 inches). Thus, over an operating period of 100,000 hours, the circumference will expand by 29 mm (1.13 inches) which can be easily tolerated. The thickness of the shell can be estimated with the following formula:[13]

$$t = \frac{(D)(P)}{(2)(H)} \qquad \text{(B-1)}$$

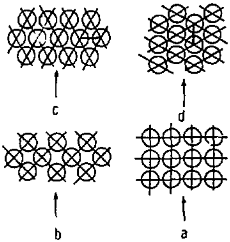
where

D = inner diameter in inches
P = pressure in psi
H = hoop tension in psi

Inserting the above values gives a thickness of 17 mm (0.68 inches). Thus, a thickness of 19 mm (0.75 inch) for the shell of the helium-to-water heat exchanger seems to be adequate.

If the water in the tubes of the primary helium-to-water heat exchanger is also pumped at a pressure of 1.03 MPa (150 psi), there will be no pressure in the tube walls. However, it is desirable to have the walls thick enough to stand 1.03 MPa (150 psi) in case coolant flow fails in either the shell or the tubes. It has been recommended that Incoloy alloy 800H be used as the tube material. At 704° C (1300° F), a hoop tension of 24.1 MPa (3500 psi) will cause a creep rate of 0.00001 percent per hour. Thus, the thickness of the tube walls is found from Eq. (B-1) to be about 0.4 mm (0.016 inches). A standard tube size with a 19-mm (0.75-inch) inner diameter and a 27-mm (1.05-inch) outer diameter is chosen. The thickness of this tube is much greater than is required to sustain 150 psia. It is believed that this tube size is also adequate to support the potential differential pressure loads and thermal loads due to axial and radial thermal gradients; however, a detailed analysis of the helium-to-water heat exchanger will be necessary in order to confirm this. The structural design guide for the helium heat exchanger will be Code Case 1592 (Class I Components in Elevated Temperature Service), Section III of the ASME Boiler and Pressure Vessel Code. This code specifies a maximum allowable stress in tension of 32 MPa (4600 psi) at 704° C (1300° F) which is well above that expected with a tube thickness of 4 mm (0.15 inch).

Tube bundles may have several kinds of arrangements (see Figure B-2).[14] The outer surfaces of the tubes may be cleaned more readily when the tubes are arranged on a square pitch rather than a triangular pitch. On the other hand, the triangular-type lattice allows greater turbulence and heat transfer surface per unit volume. This reduces the diameter of the heat exchanger, but the pressure drop may also be increased across the heat exchanger. The greater the allowed pressure drop, the lower will be the cost of the heat exchanger.



a. In-line arrangement with square pitch
b. Staggered arrangement with square pitch
c. Staggered arrangement with triangular pitch
d. Staggered arrangement with triangular pitch

Figure B-2. Several Arrangements of Tubes in Bundles

It has been found that the overall cost of a heat exchanger is minimum if the pumping power chargeable to the heat exchanger is in the range of 0.5 to 1 percent of the heat which is transferred.[14] If the pumping power is assumed to be 1 percent of the heat transferred, then for a 6.5 MW(t) reactor, the pumping power will be 32.2 kW/h (110,000 Btu/h or ~43.24 hp). A helium temperature of 260° C (500° F) is assumed to exist at the helium circulator. The reactor output temperature is assumed to be 649° C (1200° F) and the inlet temperature is taken to be 260° C (500° F). This gives an enthalpy drop of about 2.02 MJ/kg (969.4 Btu/lb). The helium mass flow rate is then calculated to be about 3.19 kg/s (25,305 lb/h). The allowed drop in pressure across the heat exchanger is then estimated with the formula:[13]

$$\Delta P = \frac{(\rho)(W)}{\dot{m}_n} \tag{B-2}$$

where

ρ = helium density in lb/ft^3
ΔP = pressure drop in lb/ft^2
W = pumping power in ft-lb/s
$\dot{m}_n$ = helium mass flow rate in lb/s

Inserting the proper values gives a pressure drop of about 13.8 kPa (2 psi). Consultation with manufacturers indicates that if a triangular lattice is used, a pressure drop of 17.2 kPa (2.5 psi) will bring about a good balance between all the above considerations.

The U-tube heat exchanger is simpler and cheaper than other types of heat exchangers and perhaps is also the best type of exchanger for this application. The U-tube structure eliminates failure due to severe thermal strains and it also allows for expansion. The U-tube structure is easier to seal than the floating head design; however, it is more difficult to clean the U-tubes by mechanical means. Adequate water and gas purification systems may make it unnecessary to clean the helium-to-water heat exchanger during its usable life. Such a system would also eliminate the contamination risk which might occur during cleaning of the primary helium heat exchanger.

Helium should be placed on the shell side of the heat exchanger for the following reasons:[15]

1. The most corrosive fluid should be inside the tubes,
2. Better heat transfer properties will result if the most inviscid fluid is on the shell side, and
3. It is desirable for the water to be at a pressure equal to or greater than the pressure of the helium and the fluid with the greatest pressure should be inside the tubes.

It has been found that a helium coolant pressure of less than 506 kPa (5 atm) results in an excessively large heat exchanger which will require much pumping power. This would increase the cost of both the heat exchanger and the pump. It is felt that a coolant pressure of 1 MPa (10 atm) is a good choice for the helium in the proposed HFFPR installation since related technology has been developed to handle much higher pressures.

The preceding considerations indicate that the helium-to-water heat exchanger should have the following design characteristics:

1. A type 316 steel shell, 13 mm (0.5 inch) thick,
2. Incoloy alloy 800H tubing material,
3. An inner diameter for the tubes of 19 mm (0.75 inch),
4. An outer diameter for the tubes of 27 mm (1.05 inches),
5. Helium flow on the shell side,
6. Water flow in the tubes,
7. A triangular lattice for the tube bundle, and
8. A U-type heat exchanger.

APPENDIX C

Major HFFPR System Components and Nominal Design Conditions

In this section, the entire cooling system will be considered as a unit and the interaction between the components will be estimated by means of simple linear analysis.

These calculations are not meant to be exact but are carried out only in order to provide a rough idea of what the nominal operating conditions might be in the system once it has been constructed. First, the basic core parameters are considered.

At the time this report was written, a complete analysis of the HFFPR reactor core had not been performed; consequently, it is necessary to make the following assumptions regarding the design parameters:

1. It is assumed that the helium will occupy 25 percent of the total core volume,
2. It is assumed that the core inlet temperature will be 260° C (500° F),
3. It is assumed that the core outlet temperature will be 649° C (1200° F), and
4. It is assumed that the core radius will be 0.8 metre (2.5 feet).

It is interesting to note that the Peach Bottom HTGR had an outlet temperature of 734° C (1354° F) and an inlet temperature of 334° C (634° F), that the Dragon reactor had an outlet temperature of 750° C (1382° F) and an inlet temperature and pressures (~150 psia) close to those chosen for this preliminary cooling system design of the HFFPR. The HFFPR reactor fuel element design, however, is expected to be quite different from these other reactors and may impose additional restrictions on the cooling system parameters.

Helium properties in the reactor core will be taken as those at the mean temperature of 454° C (850° F). Since the reactor power is 6.5 MW(t), the helium mass flow rate is found from the formula

$$\dot{m}_n = \left(\frac{dQ}{dt}\right)\left(\frac{1}{C_p \Delta T}\right) \qquad \text{(C-1)}$$

where

ΔT = temperature drop across the reactor in degrees Fahrenheit

C_p = specific heat of helium in Btu/lb·°F

$\frac{dQ}{dt}$ = heat transfer rate in Btu/h

Inserting the appropriate values yields a helium mass of 3.19 kg/s (25,305 lb/h). The portion of the reactor core's cross-sectional area which is covered by the helium is simply the cross-sectional area of the core times the volume fraction. Thus the approximate mean core velocity can be obtained from the formula

$$V_m = \frac{\dot{m}_n}{\rho A_f} \qquad \text{(C-2)}$$

where

ρ = helium density in lb/ft^3

A_f = helium cross-sectional flow area in the core in ft^2

This formula yields a mean core velocity of about 12.4 m/s (40.7 ft/s). In the following calculations, a mean velocity of 12.2 m/s (40 ft/s) will be used for the core. Since the specific volume of the helium is different at different temperatures, the core outlet velocity will not be the same as the core inlet velocity. This velocity difference affects the enthalpy change across the reactor according to the formula:[16]

$$h_o - h_I = C_p \left(T_o - T_I\right) + \frac{\left(V_o^2 - V_I^2\right)}{2g} \qquad \text{(C-3)}$$

where

h_o = output enthalpy in Btu/lb

h_I = input enthalpy in Btu/lb

T_o = output temperature in degrees Fahrenheit

T_I = input temperature in degrees Fahrenheit

C_p = specific heat of helium in Btu/lb·°F

g = acceleration of gravity in ft/s^2

V_o = output helium velocity

V_I = input helium velocity

Using the formulas

$$V_o = \frac{V_m S_o}{S_m} \qquad \text{(C-4)}$$

and

$$V_I = \frac{V_m S_I}{S_m} \tag{C-5}$$

where

S_o = output specific volume in ft^3/lb

S_I = input specific volume

S_m = mean specific volume

it is found that the change in enthalpy due to the velocity difference is only about 93 J/kg (0.04 Btu/lb) and can be neglected compared to the entropy change of 2.02 MJ/kg (869.4 Btu/lb) caused by the temperature drop.

The basic design for the helium-to-water heat exchanger will be of the U-tube type, as illustrated in Figure C-1. The design parameters of this heat exchanger will be estimated with a linear model.

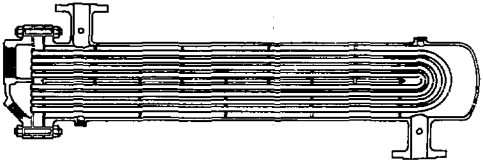


Figure C-1. U-Tube Heat Exchanger (Figure from Reference 11)

First, find the logarithmic mean temperature difference from the formulas[17]

$$\Delta T_1 = T_{h_1} - T_{c_1} \tag{C-6}$$

$$\Delta T_2 = T_{h_2} - T_{c_2} \tag{C-7}$$

and

$$\Delta T = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1/\Delta T_2\right)} \tag{C-8}$$

where

T_{h_1} = high temperature on the hot side

T_{h_2} = low temperature on the hot side

T_{c_1} = high temperature on the cold side

T_{c_2} = low temperature on the cold side

For the hot side of the heat exchanger, T_{h_1} is 649° C (1200° F) and T_{h_2} is 260° C (500° F); for the cold side, T_{c_1} is 149° C (300° F) and T_{c_2} is 93° C (200° F). All of these values are assumed. The logarithmic mean temperature difference ΔT is then found to be 286° C (546.4° F). The correction factor for the heat exchangers operating at these temperatures is very close to unity; consequently, the ΔT value will be assumed as approximately correct.

The properties of helium at the mean shell temperature of 454° C (850° F) and 150 psia are as follows:

ρ = 0.66 kg/m^3 (0.041 lb/ft^3) (density)

C_p = 5.2 kJ/kg·K (1.242 Btu/lb·°F) (specific heat)

M = 38 μPa·s (9.234 x 10^{-2} lb/ft·h) (viscosity)

K = 0.9 W/m^2· (0.159 Btu/h·ft^2·°F) (thermal conductivity)

The outside film heat transfer coefficient for the helium around the tubes can be estimated with the formula

$$h_o = \left(C_p \rho V\right)\left(\frac{a_4}{F_s}\right)\left(\frac{1}{R^{1-N_o}}\right)\left(\frac{1}{P^{2/3}}\right) \tag{C-9}$$

where

$$R = \frac{V \rho D_o}{\mu} = \text{Reynolds number} \tag{C-10}$$

$$P = \frac{\mu c}{k} = \text{Prandtl number} \tag{C-11}$$

D = outside tube diameter

Here the helium shell velocity will be assumed to be 9.1 m/s (30 ft/s). Eq. (C-9) then yields an outside film heat transfer coefficient of 364 W/m^2·K (64.1 Btu/h·ft^2·°F). For a gas, a_4 was taken to be 0.33, F_s to be 1.25, and N_o to be 0.6.[15]

In order to get an estimate of the inside film heat transfer coefficient, the following properties of water at a mean temperature of 121° C (250° F) are needed:[15]

ρ = 985 kg/m^3 (61.5 lb/ft^3) (density)

C_p = 4.23 kJ/kg·K (1.01 Btu/lb·°F) (specific heat)

μ = 245 µPa·s (0.165 x 10^{-3} lb/ft·s) (viscosity)

k = 2.24 W/m^2·K (0.394 Btu/h·ft^2·°F) (thermal conductivity)

The inside film heat transfer coefficient is determined from the formula

$$h_i = \left(C_p \rho V\right)\left(\frac{a_3}{F_s}\right)\left(\frac{1}{R^{1-N_i}}\right)\left(\frac{1}{P^{2/3}}\right) \qquad \text{(C-12)}$$

where a_3 is chosen as 0.023, F_s as 1.25, and N_i as 0.8.[15] If the preceding values for water properties are substituted into this formula and the velocity of the water in the tubes is assumed to be 0.6 m/s (2 ft/s), an inside film heat transfer coefficient of 4.134 kW/m^2·K (728 Btu/h·ft^2·°F) results.

If fouling resistance is neglected, the overall inside heat transfer coefficient for the helium-to-water heat exchanger is given by the formula[15]

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{\left(D_o - D_i\right)/2}{K_w\left(D_o/D_i\right)} + \frac{1}{h_o\left(D_o/D_i\right)} \qquad \text{(C-13)}$$

where

U_i = overall heat transfer coefficient in Btu/h·ft^2·°F

D_o = outside tube diameter

D_i = inside tube diameter

K_w = thermal conductivity of the Incoloy 800H tube walls at a mean temperature of 121° C (250° F)

Given a value of 96 for K_w, the overall inside heat transfer coefficient is found to be 450 W/m^2·K (79.2 Btu/h·ft^2·°F).

The following formulas are needed to determine the overall heat transfer area:

$$U_o A_o = U_i A_i \qquad \text{(C-14)}$$

$$U_o A_o = \frac{dQ/dt}{\Delta T} \qquad \text{(C-15)}$$

where

U_o = overall outside heat transfer coefficient in $Btu/h \cdot ft^2 \cdot °F$

A_o = overall outside heat transfer area in ft^2

dQ/dt = rate of heat transmission in Btu/h

U_i = overall inside heat transfer coefficient

A_i = overall inside heat transfer area

Eq. (C-15) yields U_oA_o as 40,264 Btu/h·°F; the value of A_i is then found from

$$A_i = \frac{U_oA_o}{U_i} \tag{C-16}$$

to be 47.2 m^2 (508.37 ft^2)

The mass flow rate of the water in the tubes is found from the equation

$$\dot{m}_w = \frac{dQ/dt}{(h_o - h_i)} \tag{C-17}$$

where

h_o = enthalpy of output water in Btu/lb at 300°F

h_i = enthalpy of input water in Btu/lb at 200°F

If the values of enthalpy are taken to be 269.87 and 168.40, respectively, the water mass flow rate is 27.32 kg/s (216,813 lb/h). Based on this, the overall cross section as seen by the water in the heat exchanger can be found from

$$A_c = \frac{\dot{m}_w}{\rho V} \tag{C-18}$$

This formula gives a value of 0.05 m^2 (0.49 ft^2). The number of tubes necessary to show this cross section is determined by

$$N = \frac{(A_c)(4)}{(\pi)(D_i)^2} \tag{C-19}$$

The number of tubes in the heat exchanger is 160. The length of the tubes in the heat exchanger can now be found from the formula

$$L = \frac{A_i}{\pi D_i N} \tag{C-20}$$

which yields a value of approximately 5 metres (16.3 feet). The tubes which are bent into a U-shape will then give a length of about 2.4 metres (8 feet). Inclusion of the shell, piping fixtures, etc., will bring the final length to about 3 metres (10 feet).

The cross section of the heat exchanger depends on the diameter, spacing, and arrangement of the tubes. It has been assumed that a triangular lattice (illustrated in Figure C-2) will be used. The pitch ratio, σ, is defined by the formula

$$\sigma = \frac{Z}{D_o} \tag{C-21}$$

where D_o is the outer diameter of the tubes and a clearance between the tubes of 6 mm (0.25 inch) is assumed.[15] Since D_o is 27 mm (1.05 inches), Z will have a value of 33 mm (1.3 inches); therefore, σ has a value of 1.23.

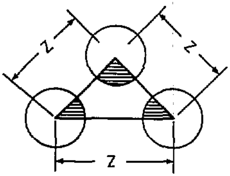


Figure C-2. Triangular Lattice

The ratio of the heat transfer surface to the total volume is defined by the equation[15]

$$S = \frac{3.63}{\sigma^2 D_o} \tag{C-22}$$

If the preceding values given for σ and D_o are substituted into Eq. (C-22), the value of S is 27.42 ft^2/cm-ft. The heat transfer surface can be found from the formula

$$A_o = N \Pi D_o L \tag{C-23}$$

which yields 66.6 m^2 (716.9 ft^2) when the proper values are inserted. Let A_{crt} be the cross-sectional area of the tube bundle in the heat exchanger. The area can be derived from the formula

$$A_{crt} = \frac{2A_o}{SL} \tag{C-24}$$

When the preceding values are substituted into it, Eq. (C-24) yields 0.3 m^2 (3.2 ft^2) for the cross-sectional area of the heat exchanger. This is equivalent to a diameter of about 0.6 metre (2 feet). The shell will increase the diameter so that the actual heat exchanger will be around 0.8 metre (2.5 feet) in diameter.

These calculations indicate that the helium-to-water heat exchanger will have parameters of the sizes indicated in Table C-I.

TABLE C-I

Helium Heat Exchanger Parameters

Parameters	Values
ΔT	285.8° C (546.4° F)
h_o	364 $W/m^2 \cdot K$ (64.1 $Btu/h \cdot ft^2 \cdot °F$)
h_i	4.137 $kW/m^2 \cdot K$ (728.5 $Btu/h \cdot ft^2 \cdot °F$)
U_i	450 $W/m^2 \cdot K$ (79.2 $Btu/h \cdot ft^2 \cdot °F$)
U_o	324 $W/m^2 \cdot K$ (57 $Btu/h \cdot ft^2 \cdot °F$)
N	160 tubes
L	3 metres (10 feet)
D	0.8 metre (2.5 feet)

When the same methods of calculation given in Table C-I are applied, the parameters of the water-to-water secondary heat exchanger are determined to be approximated by those indicated in Table C-II.

TABLE C-II

Water Heat Exchanger Parameters

Parameters	Values
ΔT	50.5° C (123° F)
h_o	4.68 $kW/m^2 \cdot K$ (824 $Btu/h \cdot ft^2 \cdot °F$)
h_i	4.137 $kW/m^2 \cdot K$ (728.5 $Btu/h \cdot ft^2 \cdot °F$)
U_i	2.57 $kW/m^2 \cdot K$ (452 $Btu/h \cdot ft^2 \cdot °F$)
U_o	1.89 $kW/m^2 \cdot K$ (333 $Btu/h \cdot ft^2 \cdot °F$)
N	163 tubes
L	2.7 metres (9 feet)
D	0.9 metre (3 feet)

Finally, the pressures and mass flow rates for the primary and secondary heat exchangers are given in Table C-III.

TABLE C-III

Pressures and Mass Flow Rates

Helium Heat Exchanger

Helium pressure primary	150 psia
Water pressure secondary	150 psia
Mass flow of helium	3.2 kg/s (25,305 lb/h)
Mass flow of water	27.3 kg/s (216,813 lb/h)

Water Heat Exchanger

Primary water pressure	150 psia
Secondary water pressure	25 psia
Mass flow in primary	27.3 kg/s (216,813 lb/h)
Mass flow in secondary	55.5 kg/s (440,528.6 lb/h)

Since Sandia Laboratories has no computer codes for the analysis of heat exchangers, the decision was made to solicit computer code analyses from several different manufacturers. Since the above analysis is ideal and involves many assumptions, the results obtained from the computer codes developed by the manufacturers would seem to be more reliable.

An analysis was performed by APCO Nuclear to determine approximate design parameters for the helium-to-water heat exchanger. The following are the basic design parameters which were obtained:

- Helium
  - Temperature in -- 704° C (1300° F)
  - Temperature out -- 315° C (600° F)
  - Pressure -- 1 MPa (10 atm)
  - Flow Rate -- 3.15 kg/s (25,000 lb/h)
- Cooling Water
  - Temperature in -- 93° C (200° F)
  - Temperature out -- 149° C (300° F)
  - Pressure -- 1 MPa (10 atm)
  - Flow Rate -- 24.1 kg/s (191,000 lb/h)
- Tower Water
  - Temperature in -- 65.5° C (150° F)
  - Temperature out -- 38° C (100° F)
  - Pressure -- 25 psia
  - Flow Rate -- 48.4 kg/s (384,500 lb/h)

- <u>Size</u>

  Outer Diameter -- 0.8 metre (2.5 feet)

  Length -- 3.2 metres (10.5 feet)

  Space Required -- 1.2 by 1.2 by 3.4 metres (4 by 4 by 11 feet)

APCO estimates the cost of the primary heat exchanger to be from $75,000 to $100,000.

APCO has indicated that the secondary water-to-water heat exchanger is practically an off-the-shelf item and could be supplied for about $25,000. It would have an outer diameter of about 0.6 metre (2 feet) and a length of about 3 metres (10 feet).

Westinghouse has proposed the system shown in Figure C-3. The heated helium enters the primary heat exchanger where heat is transferred to pressurized water. The heated water is then transferred to a reboiler heat exchanger where the heat is transferred by vaporization of the water. The resulting steam is then passed to a cooling tower where the steam is condensed and subcooled.

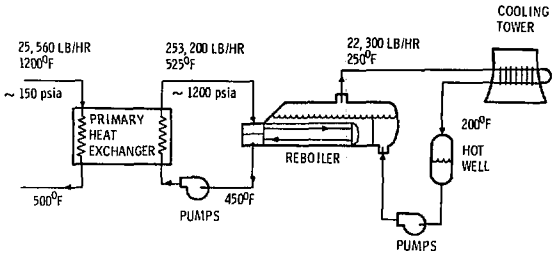


Figure C-3. Westinghouse Cooling System with Reboiler

[illegible] 6.5 MW [illegible] heat load, the primary heat exchanger would require about [illegible] tubes, each with an outside diameter of 19 mm (0.75 inch) and [illegible] metres (60 feet). The heat exchanger matrix would have a diameter [illegible] metre ([illegible] feet) and a length of 1.8 metres (6 feet). The shell would [illegible] metre ([illegible] feet) in diameter and 2.7 metres (9 feet) long.

[illegible] would require about 350 tubes, each with an outside diameter of [illegible] and a length of 4.6 metres (15 feet). The reboiler shell would [illegible] metre ([illegible] feet) in diameter and 6 metres (20 feet) long. The water [illegible] the primary and reboiler heat exchangers could be 13 cm (5 inches) [illegible] Westinghouse did not provide any cost information.

Wyatt Industries, Inc., suggested a two-pass U-bend with gas flow directed via internal pipe within the channel into the center of the tube field. The gas would exit through tubes on the periphery of the field and leave by means of a conventional channel nozzle. The internal pipe would be designed for a differential pressure of about 137.9 kPa (20 psi). This type of design offers the advantage of not having a shell expansion joint for operation without water flow. In addition, a minimum amount of weldment is exposed to the 704° C (1300° F) gas. This might be particularly advantageous if the helium is difficult to confine. This unit would have an outside diameter of about 76 cm (30 inches) and have an overall length of 3.6 metres (12 feet). The unit would require about 0.1 m^3/s (1750 gal/min) of water and would cost about $50,000.

In all these calculations, including the very elementary linear calculation, the size of both heat exchangers ranges from 0.6 to 0.9 metre (2 to 3 feet) for the diameter and from about 2.7 to 4.6 metres (9 to 15 feet) for the length. Consequently, it is certain that there will be no trouble in fitting these units into the SERF building and its proposed additions.

Based on all these considerations, a number of nominal operating conditions for the proposed HFFPR have been suggested. These are illustrated in Figure C-4.

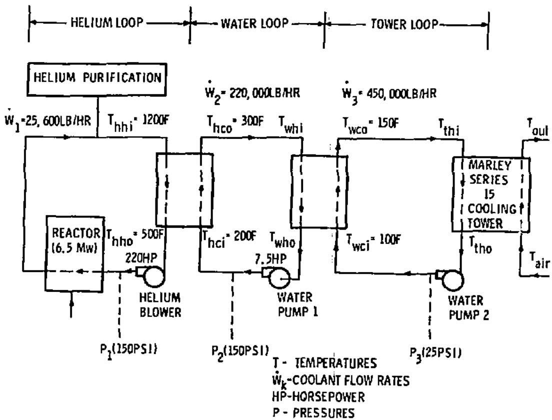


Figure C-4. HFFPR System Components and Nominal Design Conditions

APPENDIX D

Pump Design for the HFFPR Cooling System

MTI has had considerable experience in the manufacture of helium blowers. It was this company which manufactured the blowers for the UHTREX experiment at Los Alamos and is currently designing similar blowers for Union Carbide and ORNL. The new blowers have the same structural design as those used in UHTREX. This design is illustrated in Figure D-1.[7] The exact measurements of the new blower are presented in Figures D-2 and D-3.[18]

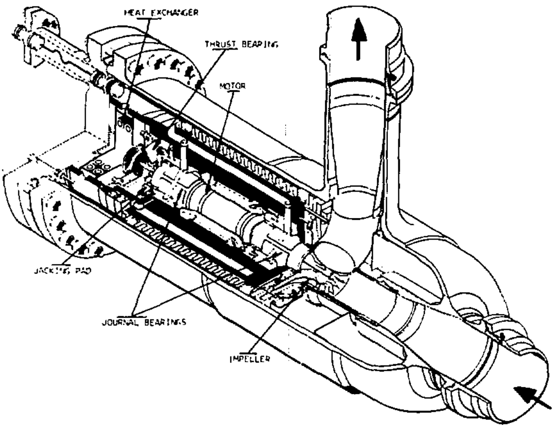


Figure D-1. Coolant Loop Blower

The circulator can easily handle 3.3 kg/s (26,000 lb/h) of helium at 150 psia and 315° C (600° F) with a head rise between 17 and 34 kPa (2.5 and 5 psi). Power would be approximately 82 kW (110 hp) at 17 kPa (2.5 psi) head and 164 kW (220 hp) at 34 kPa (5 psi) head.

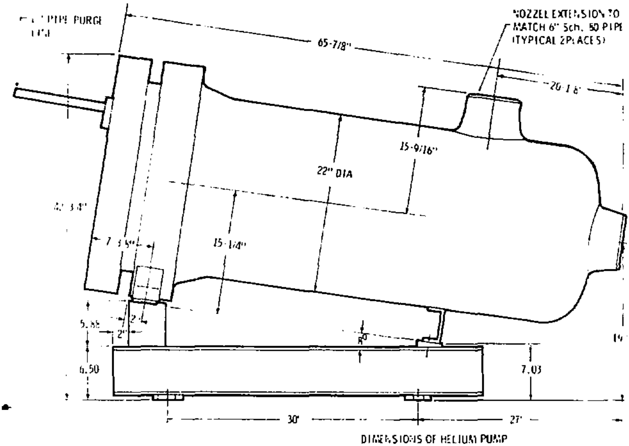


Figure D-2. Dimensions of Helium Pump

The estimated cost of the blower will be approximately $450,000. This price does not include the high-frequency 200-Hz power supply which would cost approximately $75,000.

The pump between the heat exchangers, referred to as pump 1, will have to handle about 28 kg/s (220,000 lb/h). The cooling tower pump, referred to as pump 2, will handle 57 kg/s (450,000 lb/h). The dynamic heads for these pumps can only be determined after a complete analysis of the cooling system.

For the purpose of cost estimation, pump 1 will be assumed to have a dynamic head of 15 metres (50 feet). This requirement can be satisfied by an Aurora Pump, 3 x 4 x 10B, series 410. The motor will be 5.6 kW (7.5 hp) and will cost about $1700. Pump 2 will be assumed to have a dynamic head of 46 metres (150 feet). This pump can be an Aurora Pump, 5 x 6 x 15, series 413 and will cost $3,000.[19] These pumps will be not more than 0.9 metre (3 feet) tall and will occupy a space not greater than 0.2 m^2 (2.5 ft^2). They will stand in a vertical position.

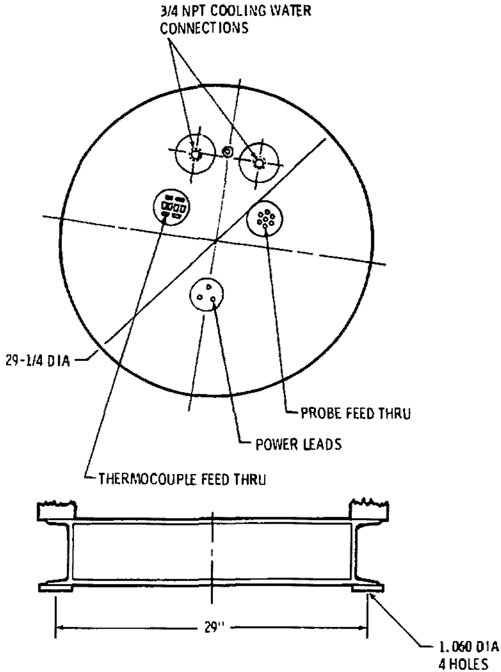


Figure D-3. Cross-Sectional Dimensions of the Helium Pump

Figure D-4 presents three views of the water pump configuration. The pumps used at HFFPR will operate at a speed of 1750 rpm. Pump 2 will be rated at about 37 kW (50 hp).

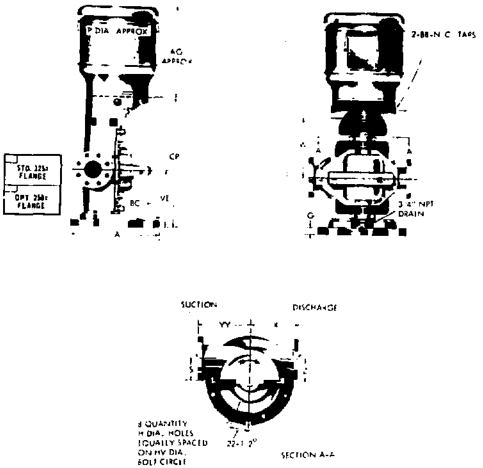


Figure D-4. Water Pump Configuration

APPENDIX E

The HFFPR Cooling Tower

The cooling tower will have to cool 57 kg/s (450,000 lb/h) of water ranging in temperature from 38° C to 65° C (100° F to 150° F). It should be a forced draught cooling tower of the configuration illustrated in Figure E-1. The requirement will be satisfied by a Marley, Series 15, tower, which sells for about $12,000. It will cover an area 6.1 by 4.9 metres (20 by 16 feet) and will stand 5.2 metres (17 feet) high. The motor which drives the fan will be 15 kW (20 hp).[20]

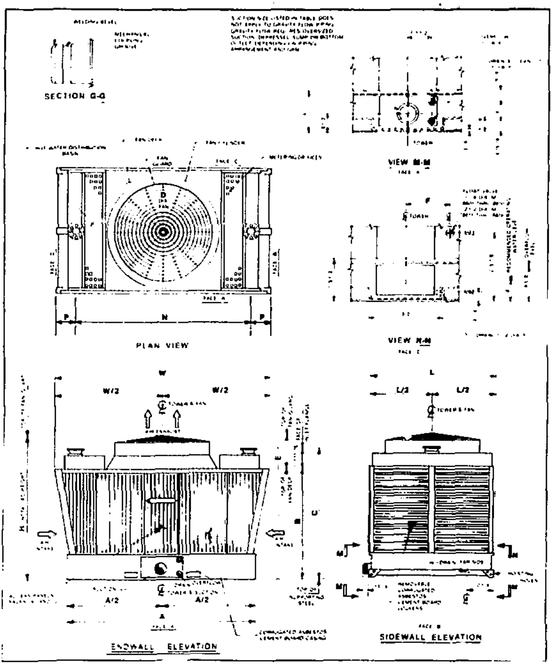


Figure E-1. Cooling Tower Configuration

APPENDIX F

The SERF Reactor Vessel as a Component of the HFFPR

The SERF reactor vessel, located in Building 6580, is encased in very strong concrete and will be both difficult and expensive to remove. Consequently, it would be desirable to include it in the HFFPR design.

The original reactor generated very low temperatures on the order of 49°C (120° F). The new reactor will have an outlet temperature of 649° C (1200° F). The SERF reactor vessel was designed to withstand 1 MPa (150 psi) without distortion and not to rupture below 2.76 MPa (400 psi).[21] It is important to determine what kind of pressures the vessel would be subjected to if an attempt were made to use it in direct conjunction with the HFFPR.

The portion of the SERF vessel embedded in concrete is illustrated in Figure F-1. If it is assumed that the lower portion of the vessel is subjected to a temperature of 649° C (1200° F), a quick estimate of the hoop pressure the vessel will experience can be obtained from the formula

$$P = \frac{Eh(\alpha)\Delta T}{R} \tag{F-1}$$

where

P = hoop pressure

E = Young's modulus ($\simeq 30 \times 10^6$ psi)

α = expansion coefficient ($\simeq 7 \times 10^{-6}$ °F^{-1})

ΔT = temperature difference

R = shell radius

h = shell thickness $\simeq$ 0.5 inch

If a 593° C (1100° F) drop of temperature is assumed across the shell, the pressure obtained from Eq. (F-1) is 22.1 MPa (3208 psi). This is a far greater pressure than the 2.76 MPa (400 psi) it was designed to endure.

The shell is locked in concrete and little or no room is allowed for expansion. An estimate of the relative amount of expansion between the concrete

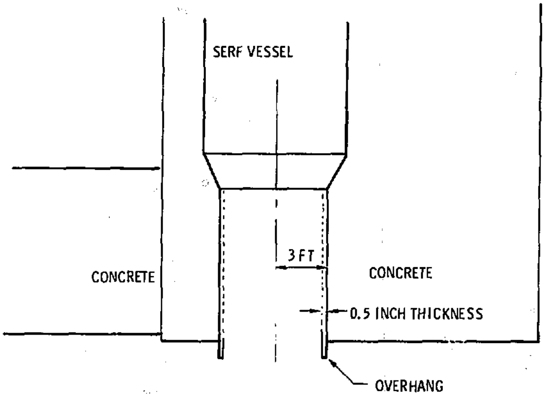


Figure F-1. Constraints on SERF Reactor Vessel

and the cylinder can be obtained from the equation[22]

$$U(a) = \frac{-p\left(1 - \mu^2\right)\left(\frac{a^2 - b^2}{a}\right)}{E\left[(1 + \mu) + (1 - \mu)\left(\frac{b^2}{a^2}\right)\right]} \qquad \text{(F-2)}$$

where

a = radius of the cylinder

P = pressure in the cylinder

μ = Poisson's ratio

E = Young's modulus ($\sim 4 \times 10^{-6}$ for concrete)

b = cylinder thickness

For the concrete, an infinite thickness will be assumed; therefore,

$$\lim_{b \to \infty} U(a) = \frac{Pa}{E}(1 + \mu) \qquad \text{(F-3)}$$

When the proper values are substituted into Eq. (F-3), a value of 0.9 mm (0.037 inch) for the amount of concrete expansion results. Eq. (F-2) gives the expansion of the cylinder or reactor vessel as approximately 10 mm (0.396 inch). Thus, it is obvious that great stresses will be produced.

Next, it is assumed that the shell is cut off at some distance L from the concrete, as illurtrated in Figure F-1. This produces what is commonly referred to as an overhang. An approximate idea of what happens here can be obtained by considering the following equation for a clamped thin cylinder:[22]

$$\frac{d^4W}{dx^4} + 4\beta^4W = \frac{Z}{D} \tag{F-4}$$

where

$$\beta = \sqrt[4]{\frac{hE}{4(a)^2D}} = \left[\frac{3\left(1-\mu^2\right)}{a^2h^2}\right]^{\frac{1}{4}} \tag{F-5}$$

and

$$D = \frac{Eh^3}{12\left(1-\mu^2\right)} \tag{F-6}$$

The letter Z indicates components of either external or inertia load per unit area. In this instance, Z will simply be taken as the thermal pressure in the shell. A solution for Eq. (F-4) takes the form

$$\begin{aligned} W(x) &= e^{\beta x}\left(A_1 \sin \beta x + A_2 \cos \beta x\right) \\ &\quad + e^{-\beta x}\left(A_3 \sin \beta x + A_4 \cos \beta x\right) + \frac{P}{4\beta^4 D} \end{aligned} \tag{F-7}$$

Eq. (F-4) takes the same form as the equation for a beam on an elastic foundation. The moment is obtained by differentiating Eq. (F-7) to the 2nd derivative. An evaluation of the constants at boundary conditions gives a moment equation as

$$M_x = \frac{P}{2\beta^2}\left(\frac{\cosh^2 \beta L - \cos^2 \beta L}{\cosh^2 \beta L + \cos^2 \beta L}\right) \tag{F-8}$$

Application of the preceding equations reveals that if L is as small as 1 inch, the moment is 17,385. The stress is obtained from the formula

$$\sigma = \frac{6M}{h^2} \tag{F-9}$$

This formula yields a stress of 2.88 GPa (417,240 psi). Thus, if the temperature reaches 649° C (1200° F), no overhangs can be allowed. This means that if the vessel is allowed to remain in the building with the HFFPR, it should be cut off at the concrete and the shell should never be subjected to a temperature greater than 177° C (350° F).

These restrictions which must be placed on the use of the SERF reactor vessel lead to the question: How should the SERF reactor vessel be used with the HFFPR if indeed it should be used at all? In the original reactor, the vessel had many functions. It provided reactor containment, core access, and support and guides for the control rod drives. It also served as a shield against reactor radiation and as a part of the cooling system. In addition, it provided access parts for detectors, internal wiring, and other instrumentation, it permitted underwater fuel discharge to cutoff pool, and it provided storage space for partially spent fuel elements or other radioactive sources. It may be possible to use the old vessel for some of the above uses if its temperature can be kept below 177° C (350° F).

The existing water cooling system could be modified to keep the SERF vessel below 177° C (350° F); however, appropriate assurances should be made to prevent water from penetrating the primary reactor containment of the HFFPR and flooding the core. If such precautions are not taken, criticality consequences could occur.

The water cooling system might be modified to cool not only the old reactor vessel but the new one as well. A thorough thermal analysis of both vessels should be carried out with a view to reducing the heat transfer between them. The water flowing from the SERF vessel would move counter to the heat flow and could conceivably serve as an emergency cooling circuit in case the helium loop breaks down. Once again it must be emphasized that this will be possible only if water can be kept out of the gas-cooled core.

APPENDIX G

HFFPR System Layout in the SERF Building

The overall structure of the SERF building is illustrated in Figure G-1. Floor plans for this facility are given in Figures G-2, G-3, and G-4.

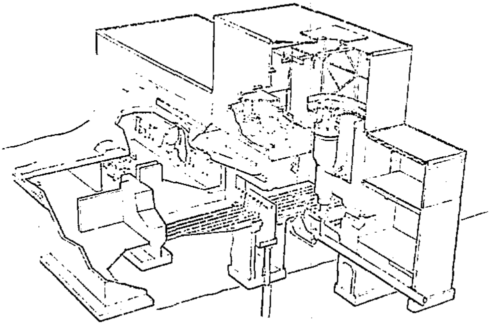


Figure G-1. Sandia Engineering Reactor Facility (SERF)

It is assumed that the helium-to-water heat exchanger will utilize a concentric pipe such as is used in the steam generators at the Peach Bottom HTGR for the helium. For this configuration, only one hole needs to be cut from the lower part of the beam room to the reactor vessel. The helium-to-water heat exchanger should be located in the beam room since this room can provide some shielding against the radioactivity that might be present. This layout is illustrated in Figure G-5. The secondary water heat exchanger may be placed in the pump house together with the two water pumps in the manner illustrated in Figure G-6.

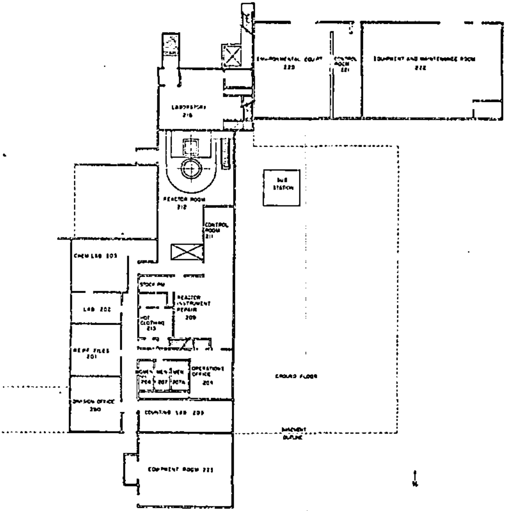


Figure G-2. Ground Floor Plan

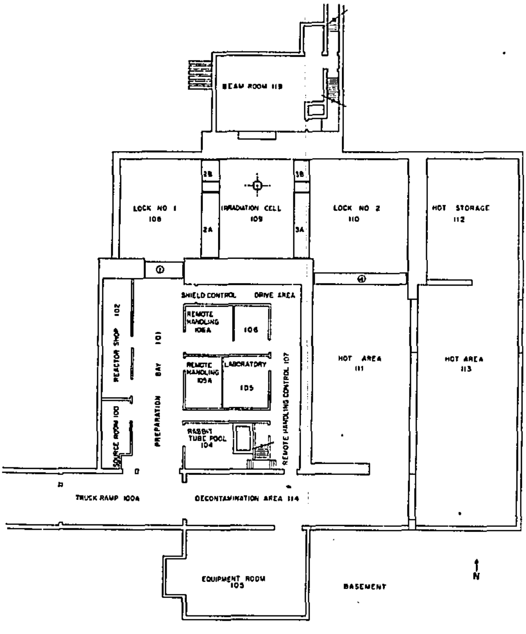


Figure G-3. Basement Floor Plan

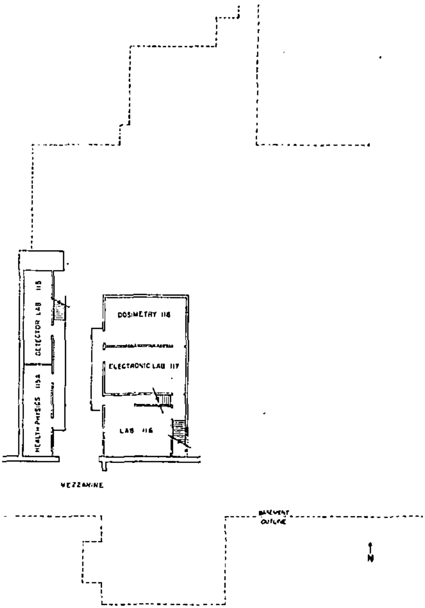


Figure G-4. Mezzanine Floor Plan

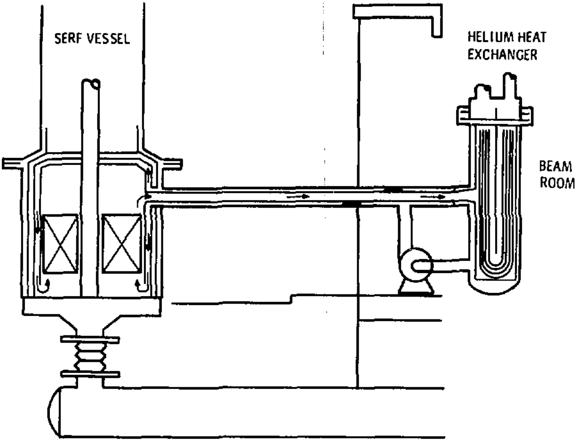


Figure G-5. HFFPR Helium Cooling Circuit

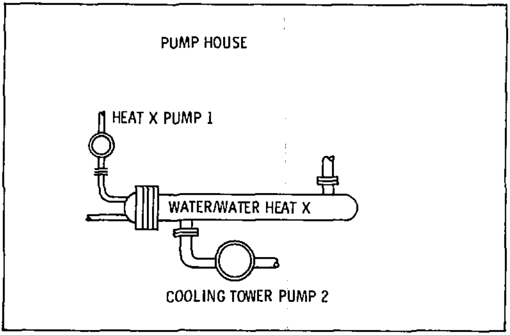


Figure G-6. Location of Water Heat Exchanger

The helium purification system should be built as a modular unit and be placed in Lock No. 1 (see Figure G-7). A gas-tight liner complete with a gas-tight door should be constructed in Lock No. 1. In addition, Lock No. 1 should be operated at a negative pressure in relation to other adjacent rooms in the building. These safety precautions are necessary to prevent the escape of any gas which could be slightly radioactive. The price of the helium purification system is estimated to be approximately $250,000.

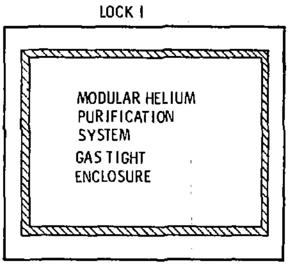


Figure G-7. Location of Helium Purification System

APPENDIX H

Some Analytical Conclusions and a Discussion of System Cost

Figure 1 (page 8) summarizes the nominal operating condition of the proposed HFFPR cooling system. The input and output temperatures chosen for the hot side of the helium heat exchanger were based on computer calculations obtained with a simple linear model of the cooling systems devised for this report.

In Figure H-1, a graph plotted from computer output shows the relationship between heat exchanger volume versus temperature difference across the heat exchanger. The high temperature on the hot side was held constant while the low temperature was dropped as indicated. The helium velocity in the heat exchanger did not vary. This indicates that in this case the heat exchanger increases in volume as temperature across the heat exchanger increases.

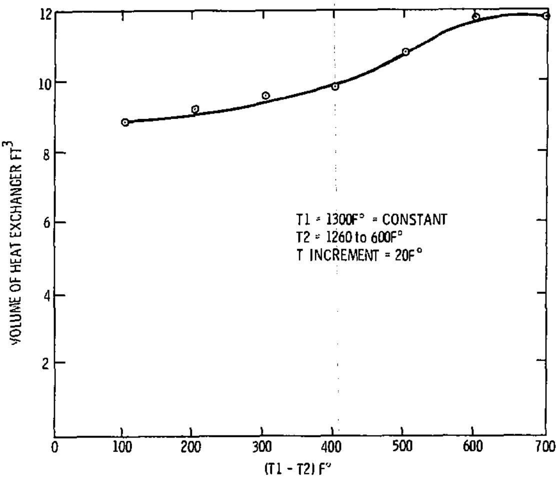


Figure H-1. Heat Exchanger Volume Variation with Temperature Increment

In the graph plotted in Figure H-2, the low temperature on the high side is kept constant, while the high temperature on the high side increases as shown. In this instance, the heat exchanger volume decreases as the temperature drop across the heat exchanger increases.

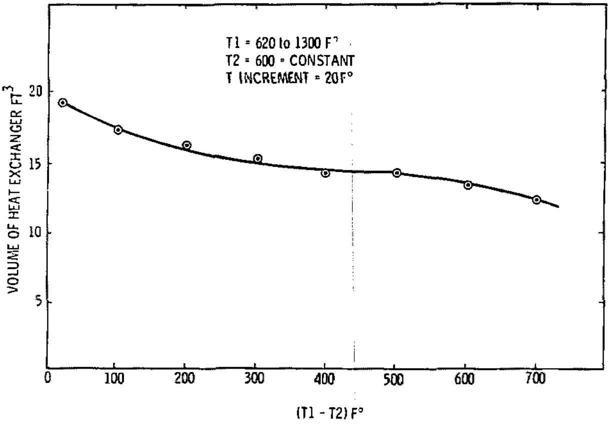


Figure H-2. Heat Exchanger Volume Variation with Temperature Increment

Figure H-3 shows that in both of the preceding cases the helium flow rate decreases as the temperature drop across the heat exchanger increases. Since the helium pump is the most expensive item, the temperature difference which gives the lowest helium flow rate and which can be handled by the materials involved represents the most economical temperature drop for the system. The reactor output temperature of 649° C (1200° F) can be easily handled by the materials which have been chosen, and the 260° C (500° F) inlet temperature gives enough temperature drop to assure a helium flow rate that can be handled by readily available helium pumps.

Table H-I summarizes the estimated prices of all the major components in the cooling system. Hence, the total cost of the overall cooling system can be estimated to be approximately $1.5 million. In addition, it may be necessary to purchase a standby helium pump which can be brought online in case the main pump fails. If so, the cost of the cooling system will increase to approximately $2 million.

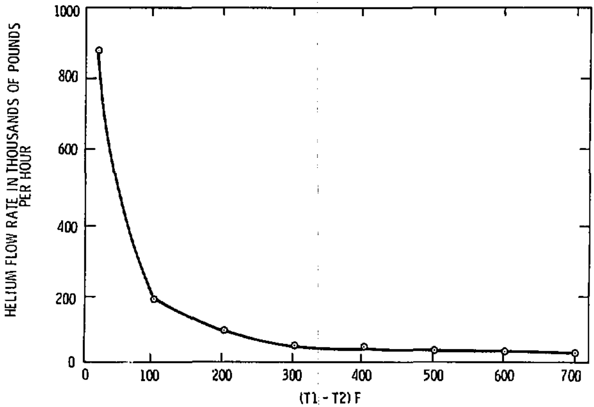


Figure H-3. Helium Flow Rate with Temperature Increment

TABLE H-I

Estimated Prices of Major Components

Helium-heat exchanger	$100,000
Water-heat exchanger	25,000
Helium pump	450,000
Helium pump controller	75,000
Water pump 1	1,700
Water pump 2	3,000
Helium purification system	250,000
Total	$904,700

It is possible that a nominal 12-inch concentric pipe which has cold helium on the outside and hot helium on the inside will be sufficient for the helium circuit. A 5-inch pipe will carry the water between the heat exchanger and a 6-inch pipe should be adequate to transport the water to the cooling tower.

It appears that all the necessary equipment needed to construct the cooling system for the gas-cooled HFFPR can be placed in the old SERF building and its proposed additions. The sizes of the heat exchangers should not be greater than 0.9 metre (3 feet) in diameter and 3 metres (10 feet) in length. The pumps will be very small and the helium purification module will fit into a room the size of Lock No. 1 which is 9.1 by 7.9 by 3.4 metres (30 by 26 by 11 feet) in size.

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