Advanced Energy Systems Division

NUCLEAR-POWERED ARTIFICIAL HEART PROTOTYPE SYSTEM DEVELOPMENT PROGRAM

MASTER

QUARTERLY PROGRESS REPORT

October 1, 1976

Through

December 31, 1976

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NUCLEAR-POWERED ARTIFICIAL HEART
PROTOTYPE SYSTEM DEVELOPMENT PROGRAM

PHASE III
QUARTERLY PROGRESS REPORT

OCTOBER 1, 1976 THROUGH DECEMBER 31, 1976

WESTINGHOUSE ELECTRIC CORPORATION
ADVANCED ENERGY SYSTEMS DIVISION
P. O. BOX 10864
PITTSBURGH, PENNSYLVANIA 15236

PREPARED FOR
UNITED STATES ENERGY RESEARCH AND DEVELOPMENT AGENCY

UNDER CONTRACT NO. EY-76-C-02-3043

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INTRODUCTION AND SUMMARY

The Nuclear Powered Artificial Heart Prototype System Development Program is being conducted by the Westinghouse Advanced Energy Systems Division and its major subcontractor, the Philips Laboratories division of the North American Philips Corporation, as Phase III of the U. S. ERDA's Artificial Heart Program under Contract No. EY-76-C-02-3043. This report presents the technical progress and accomplishments on the active program tasks for the period from October 1, 1976 through December 31, 1976.

Effective with the start of this reporting period, the work program has been directed to the completion of the IVBM system and the design of the Mark I Prototype System. New task numbers were assigned to all segments of the total program and the general objectives of each task active in the past quarter are stated in this summary.

The objectives of Task 49, Thermal Insulation and Protection Research and Development, are to confirm the sealed multifoil vacuum insulation technology to assure achievement of Prototype System requirements, to develop a selective vent for helium generated in the isotope capsule and to evaluate thermal insulation coverings for heat transmitting components. Major accomplishments in this period were the outgassing of static vacuum package (SVP) No. 2, the completion of fabrication of SVP No. 3 and No. 4, and continuation of nickel foil cup emittance evaluation.

The goals of Task 50, Thermal Converter Component Research and Development, are the advancement of materials processing and fabrication technologies of the Stirling engine and its components to ensure an engine design meeting the Mark I Prototype requirements. Significant accomplishments were the development of a process to aluminum clad titanium alloy samples by vacuum melting, the development of a screw seal for the magnetic coupling housing, a redesign of the speed control valve, and the development of a magnetic fluid seal for a reciprocating shaft.

The general objectives of Task 51, Blood Pump and Coupling Research and Development, are to improve the blood pump geometry and to advance the pump and flexible shaft life expectancies to satisfy the Mark I Prototype System requirements. Failure analysis of previously failed shafts was completed and three new shafts were placed on life test. Bearing wear tests in both Jojoba and Vitrea 21 oils continued without incident. Performance characteristics of IVBM Blood Pump No. 5 were determined and transient performance response of Bjork-Shiley valves was evaluated.

The goal of Task 52, Waste Heat Rejection Research and Development, is to identify and confirm the viability of improvements to existing approaches or new approaches for dissipation of thermal converter waste heat. In this period, design specifications for an oil coolant circulating pump were issued and two preliminary gear pump designs were completed. A parametric study of a heat exchanger was completed and a design concept was developed.

The objective of Task 53, Systems Studies, is to assure that the subsystems of the Mark I Prototype System are properly defined and integrated. In the October through December period, several meetings were held with Philips Laboratories to arrive at mutually satisfactory design interface definitions. A detailed study of the heat source/hot finger interface was performed leading to conclusions on surface finish and contact pressure relationships required for satisfactory heat transfer. A decision was made to proceed with a pumped circulating oil system for the Mark I Prototype System thermal converter waste heat removal while continuing development of the TOP system as the primary backup. The converter gear box/magnetic coupling and insulation package/thermal converter interface drawings were prepared.

Task 54, IVBM Modification Design, has as its objectives the incorporation of selected design changes to the IVBM blood pump electric drive, coupling mechanism and waste heat removal system to achieve better implantation experience and/or improved performance. In this period, the compliance chamber required to accommodate a blood pump with an oil

filled mechanism cavity and a reduced right ventricle stroke volume was designed and all detailed drawings required for manufacture were completed.

The major objective of Task 55, IVBM Fabrication, is the provision of spare, replacement and modified parts for the IVBM system. The vacuum foil insulation package for the electrically heated IVBM thermal converter was completed and checked out.

The purpose of Task 56, IVBM System Performance Testing, is to conduct checkout and performance testing of the IVBM subsystems, total system and auxiliaries and to verify implantation procedures by performance of mock implantations. The cause of previously reported thermal converter rollsock failures was identified and corrected. IVBM thermal converter No. 2 was run continuously for 146 hours at Philips Laboratories and delivered to Westinghouse where it was tested in the electrically heated mode. IVBM thermal converter No. 1 was reassembled and is being run in at Philips Laboratories. The leakage rate of the sterilant, ethylene oxide, through neoprene as used in the gloves of the sterilizing chamber was determined.

The objective of Task 57, IVBM System Life Testing, is to demonstrate the life capabilities of the IVBM system and subsystems. Blood pump No. 6 has operated 2332 hours to date without incident. The first implantable electric drive for the IVBM blood pump was subjected to a 2000 hour endurance test which was completed without any change in performance.

Task 58, Field Support, provides for technical assistance to in-vivo experiments utilizing both IVBM systems and individual blood pumps. Two additional IVBM blood pumps, two implantable electric motor blood pump drives and several spare flexible shafts were furnished to the University of Utah.

Quality Assurance activities covered by Task 59 continued consistent with the program needs.

Task 61, Mark I Thermal Insulation Design, provides for initiation of drawings and specifications of the multifoil insulation system for the Mark I Prototype System. A source for titanium alloy electric conductor feedthroughs was identified and a layout of feedthrough locations for an electrically heated Mark I system was prepared. The Mark I thermal converter/insulation package interface schematic was completed.

The objective of Task 62, Mark I Thermal Converter Design, is to provide the design and detailed drawings and specifications for production of the thermal converter for the Mark I Prototype System. An optimization of the Mark I thermal converter thermodynamic design was completed. Design of an oil cooled heat exchanger for removal of thermal converter waste heat to the blood pump was initiated. A parametric study of various pumps for circulation of cooling oil was completed and a specific configuration was selected for detail development. Sketches of various methods of securing the isotope heat source to the hot finger were completed for comparative evaluation. The baseline engine structural configuration was selected.

TASK 49 - THERMAL INSULATION AND THERMAL PROTECTION RESEARCH AND DEVELOPMENT

Work accomplished during this quarter included; 1) successful outgassing of static vacuum package (SVP) No. 2; 2) placing SVP No. 2 on life test; 3) fabrication and bakeout initiation on SVP No. 3 and No. 4; and 4) cup emittance evaluation of two cups on the modified emittance test apparatus.

STATIC VACUUM INSULATION TEST PACKAGES

The status of the static vacuum test packages to the end of this reporting period is summarized in Table 49-1. SVP No. 1 is being maintained at 839°C with a power input of 9.5 watts. SVP No. 2 is being maintained at 846°C with a power input of 10.1 watts. The top outside temperatures for SVP No. 1 and SVP No. 2 are 41°C and 43°C , respectively. A time-temperature history of the bakeout cycles for the static vacuum packages to the end of this reporting period is presented in Figure 49-1.

ELECTROFORMED CUP EMITTANCE EVALUATION

Total hemispherical emittance of two cups manufactured by Gar and one cup manufactured by Servometer, all in the as-received electroformed condition, were measured. The results are plotted in Figure 49-2. The Servometer cups were measured on the emittance test rig before it was modified to decrease the ratio between the guard and measure heater power. The Gar cups were measured on the modified test rig. Because of the different test apparatus configurations, is not possible to assume from these results that the Servometer cups exhibit the lower emittance. Additional testing will determine if this difference is due to the differences in electroformed nickel or in the experimental apparatus.

HELIUM VENT DEVELOPMENT

Preliminary calculations have shown that both the silica vent and the silicone rubber vent proposed in the component development plan would prevent contamination of the PuO_2^{16}

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TABLE 49-1
STATIC VACUUM PACKAGE PROCESSING HISTORIES

Insulation Package	Bakeout Time (days)	Static Vacuum Time Prior to Pinchoff (Days)	Pinchoff Date
No. 1 - 60 Cups No Getter	53	36	10/6/76 (86 days)
No. 2 - 60 Cups Al-Zr Getter	20	43	12/22/76 (9 days)
No. 3 - 60 Cups No Getter	14	-	-
No. 4 - 30 Cups No Getter	Started 12/28/76		
No. 5 - 45 Cups Al-Zr Getter	Being Fabricated		
No. 6 - 60 Cups 27 cups with plasma sprayed ZrO_2 spacers Al-Zr Getter	Being Fabricated

Figure 49-1. Bakeout History for Static Vacuum Packages (to 12/31/76)

Figure 49-2. Total Hemispherical Emittance Test Results

fuel form by limiting the rate of permeation of O^{17} and O^{18} back into the fuel. Oxygen sources considered in the calculations have included the H_2O and O_2 from the body fluids. The effects of temperature excursions during bench testing, where the system is alternately heated and cooled, were considered. An analysis of a silicone vent 1 mm in diameter and 1 cm long has shown that the change in internal pressure on heating and cooling would require about 120 days to reach equilibrium if the isotope was heated from 400 to 1000°K and would take about 300 days to achieve equilibrium if the isotope was cooled from 1000 to 400°K. During the heating cycle the pressure gradient would expel gas from the isotope capsule. Alternately, during the cooling cycle, the pressure gradient would force gases into the cavity.

If one considers the case where O^{17} and O^{18} are continuously being exchanged with the isotope for 10 years and thereby permeating continuously through the vent, this would represent only about 6.8×10^{-3} percent of the O^{17} and O^{18} atoms that would be contained if the PuO_2 was made from natural oxygen. This small percentage of O^{17} and O^{18} would not measurably affect the radiation dose rate of the medical grade PuO_2 .

This preliminary analysis suggests that a silicone rubber vent would be adequate for use on the MK I system. Plans to evaluate this vent concept and verify the analysis are being made.

TITANIUM INSTRUMENTATION FEEDTHROUGHS

A source for ceramic-titanium feedthroughs has been located. Once the MK I feedthrough design is approved, these feedthroughs will be ordered and evaluated.

THERMAL PROTECTION

Testing continued on the effort to determine whether foam polyethylene insulation will function adequately at the 150°F operating temperature of the developmental methanol

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circulated TOPS coolant system. To date, over nine months of testing has been completed under conditions which simulate TOPS in-vivo operation without any sign of insulation deterioration. The average heat flux measured at the outside insulation surface has remained unchanged at 0.008 watts/cm².

TASK 50 - THERMAL CONVERTER RESEARCH AND DEVELOPMENT

HOT FINGER MATERIALS

In the past quarter, two activities were initiated. The Rollmet Co. was asked for a quote and for the preparation of more samples. Philips Laboratories started the design of a setup to non-destructively test thin wall cylinders.

The rolled tubing received from Rollmet Co. looks very encouraging although it was mishandled. New blank pieces were prepared for the fabrication of adequate samples. These consisted of two pieces of cast and swaged stock and one piece of powder metallurgical stock. The latter will be especially interesting since it might be easier to fabricate and lead to a substantial savings in materials since the blank could be hot extruded to size.

A quote was received from Rollmet on fabricating various configurations of cylinders open on both ends, for possible future production of hot fingers. The quotes ranged from \$8,600 for 25 pieces to \$22,500 for 100 pieces.

The design of a non-destructive test setup was started. It is based on the interference of light in a small air gap between the polished Udimet 700 test cylinder and a concentric cylindrical lens. The change of the interference pattern with the application of internal pressure will be observed. Any inhomogeneities in the uniform strain should be easily observed since the resolution of the setup will be in the order of a fraction of the wave length of light. Such a high sensitivity requires the construction of a sturdy setup with very fine controls in three dimensions.

Polishing of ground hot finger cylinders should be beneficial for the creep rupture life of thin walls since it reduces the possibilities for crack nucleation. Experimentation in methods for this final manufacturing operation were initiated.

A new fabrication technique for the regenerator liner was investigated this quarter. It consists of using 0.5 mil thick Inconel X-750 foil wrapped into a cylinder with an overlapped seam. The

seam would be ultrasonically welded. IVBM liners have previously been machined from bar stock with a final honing operation to reduce the wall to 2 mils thickness. This is not only costly but results in a thicker wall than required to sustain the cyclic negative buckling pressure of 1.0 psi placed across it by the regenerator pressure drop.

Inconel X-750 was chosen for this new liner primarily because its thermal conductivity is very low (nearly identical to that of Udimet) and its modulus of elasticity is high - which is directly proportional to buckling pressure capability. This allows a very thin foil to be used. Inconel 0.5 mil foil is also readily available and successful ultrasonic bonding of other Inconel superalloys has been previously demonstrated.

The following formula predicts a collapsing pressure of only 0.1 psi for a 0.5 mil thick Inconel X-750 liner.

$$P = 0.807 \frac{E t^{2.5}}{L^{1.5}} \left(\frac{1}{1 - \nu^2} \right)^{3/4} \quad (1)$$

- P Buckling pressure, psi
- E elastic modulus, psi
- ν poisson ratio
- r cylinder radius, in.
- t foil thickness, in.
- L cylinder length between stiff round supports, in.

This pressure was verified using a test cylinder with an Eastman 910 glued joint and is 10 times less than the pressure requirement. Since the above formula does not take into account the added stiffening effect of the closely packed regenerator mesh around the periphery of the liner, tests were conducted to determine the increase in pressure capability with mesh installed. With a 1 mil diametral mesh clearance (slip fit), using 60 percent of the regenerator mesh in place, the pressure increased to 0.8 psi. By using a 1.5 mil press fit of this same amount of evenly spaced mesh, the pressure increased to 1.8 psi, which is very encouraging. Tests using

100 percent of the regenerator mesh have yet to be done. If these results do not provide a greater safety factor after compensating for the reduction in elastic modulus at the liner's operating temperature, then the liner thickness can be increased slightly.

Using the previous equation as a guide, a 0.2 mil increase in thickness will increase the pressure capability 230 percent, while only a 20 percent heat leak increase will result. It is estimated that an additional $3/8$ watt of engine output power will result using this 0.5 mil foil instead of the 2 mils of Udimet.

Test samples of ultrasonically welded overlap joints have been received from Sonobond Corp. The bonds appeared to be as strong as the base metal since upon being pulled apart, the metal tore before joint failure occurred. However, due to frequent sticking of the ultrasonic welding tips to the samples, it was difficult to get an acceptable reject rate. The contemplated weld pattern is a series of 40 overlapping welds (0.062 inch in diameter) along the length of the joint. To minimize tip sticking, special sapphire tips are being procured in place of the current Udimet tips, a technique previously successful at Sonobond.

STRUCTURAL INVESTIGATIONS

Techniques have been developed to clad titanium (Ti-5 Al-2.5 Sn) with aluminum (1100) by vacuum melting. Hermetic joints have been made. Aluminum to aluminum joints are subsequently made by vacuum brazing. Careful control of time and temperature during the brazing operations is necessary for successful joints. Current efforts are aimed at simulating the actual heat exchanger configuration in more representative test specimens in order that processing and brazing procedures can be determined.

ROTARY SHAFT SEALS AND STARTER COUPLING

Life testing of the flywheel shaft magnetic fluid seals continued through the quarter. The first seal fixture has an accumulated history as follows:

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1000 hours @ 7600 rpm

2638 hours @ 2400 rpm

3638 total hours

A drive motor failure in Fixture No. 2 after 3192 hours has resulted in a temporary shutdown of the fixture. Nevertheless, the seals in both fixtures remained hermetic under an imposed pressure differential of approximately 10 psi. There has been no increase in power consumption nor any evidence of ferrofluid breakdown.

A screw seal was designed for sealing Jojoba blood pump oil out of the magnetic coupling armature housing. It was decided to optimize a design based on a 0.250 inch diameter screw that is 0.250 inch long. This larger size (compared to the previous 0.188 inch dia. x 0.125 inch long seal) will allow good sealing capability with larger radial clearances, larger eccentricities, and lower oil viscosities (a consequence of the higher oil temperature when used as the system coolant). Figure 50-1 presents the theoretical performance characteristics of the seal showing in particular that an optimal thread depth exists for maximizing the pressure capability. The design point chosen is indicated and meets the 4 psi requirement.

The following are optimal thread proportions:

Maximum diametral clearance	0.0016 inch
Maximum eccentricity ϵ_o	0.5
Thread length	0.250 inch
Major diameter	0.250 inch
Number of threads	3.0
Threads/inch	14
Thread depth	0.005 inch
Thread width	0.0314 inch
Groove width	0.040 inch
Helix angle	5.19°

Figure 50-1. Gear Box Screw Seal Pressure Capability

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Performance data:

Pressure capability
@ 2400 rpm $\epsilon_o = 0.5$

6.2 psi
at 120°F (normal)

5.6 psi
at 128°F (fever)

Power consumption

14 mw

Figure 50-2 is a photograph of the starter coupling. The male portion is made from 4340 steel, has a diameter of 0.150 inch and measures 0.112 inch across the flats. The engagement is 0.10 inch. In the Mark I prototype engine, this male piston will engage a hole in the end of one of the drive crankshafts and the unit will be started by an external motor.

A test was performed to demonstrate the structural integrity of the coupling. The female portion was rigidly attached to an absorption dynamometer, while the male portion was coupled to a starter motor. Over thirty start/stop cycles against a torque of 15.6 in-lbs were performed. No evidence of distortion to either component could be seen.

CONTROL SYSTEM DEVELOPMENT

Ambient temperature changes affect oil viscosity and therefore oil pump output pressure. Via the control valve, these changes would cause a speed change of the engine even at constant load. Test data was obtained on IVBM thermal converter No. 2 during a five day continuous run period. In summary, a 2.7°F temperature increase resulted in a 15 rpm change in engine speed. An analysis of the speed control system was made and the following equation resulted, which accurately predicts the above result.

$$\Delta N = \frac{\epsilon P}{\frac{1}{K} + C} \Delta t \quad (2)$$

Δt = temperature change, °F

ΔN = speed change, rpm

P = 70; oil pressure, psi

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Figure 50-2. Prototype Starter Coupling

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- K = 10; valve gain, rpm/psi
- C = 0.019; oil pump gain, psi/rpm
- ϵ = 0.0143; oil viscosity sensitivity, $\%/100^{\circ}\text{F}$

With a calf fever of 8°F , the IVBM engine would be expected to increase in speed approximately 45 rpm.

In performing the same type of analysis for the Mark I system, it became apparent that the speed control valve could be designed to completely compensate for temperature induced changes in oil pressure since the buffer space pressure changes as well. In the original, both effects adversely affect the speed band. A schematic of the new concept is shown in Figure 50-3.

Referring to the figure, a decrease in oil pressure (due to increasing temperature) forces the oil bellows to move to the left. The same occurs when the buffer space pressure increases with temperature. By placing a vacuum in the larger diameter bellows, the change in buffer space pressure alone acting over this larger area can be made to completely balance the former forces. This results in zero net motion of the bellows. The area ratio of the two bellows to accomplish this depends on the temperature coefficient of viscosity change of the oil, ϵ , at the operating temperature, as well as the absolute buffer space pressure and oil pressure. Since the oil pump is located in the pressurized crankcase, oil pressure depends on the absolute crankcase pressure as well. It turns out that by changing the pressure (rather than using a pure vacuum) in the large bellows, temperature compensation can be effected for off-nominal engine pressures. Thus, this new valve design gives complete flexibility of engine charge pressure and oil pressure, while eliminating set point drift due to ambient temperature changes.

Every effort has been made to hold down the size of the control valve. To this end, the set point spring was folded back through the use of a seesaw pivot which should be nearly friction free. This is because it transmits axial motion to the oil bellows and spring while undergoing pure rolling at its point of contact with the support flat. This pivot area, as well as the spring

Figure 50-3. Temperature Compensated Speed Control Valve

housing, is completely filled with oil. The overall control valve length is 3.75 inches, including manifold, and fits within the axial length of the hot finger. A short speed adjusting screw is provided with access from the hot finger end.

It is presently felt that the potential improvement in overall system performance for other than just "fever conditions" would justify the use of this larger valve. The valve is roughly 1 inch longer than the non-temperature compensated version discussed in the previous quarterly report*. Since the valve location was chosen parallel to the hot finger, the added length still fits within the overall engine envelope. Simplification of implantation procedures is an example of a potential benefit, since engine speed could be set without having to first reach a long duration thermal equilibrium in the calf.

FERROFLUID PISTON SEALS DEVELOPMENT

A new concept was demonstrated this quarter for sealing a reciprocating shaft (or piston) using magnetic fluid. It is known that ferrofluid will be attracted into an increasing magnetic field gradient. Thus, by placing an axial field gradient between a reciprocating shaft to be sealed and its housing, any ferrofluid within this gap will be forced back by the gradient. A seal incorporating this effect is shown in Figure 50-4. The tapered pole face creates an axial magnetic field gradient being strongest near the magnet, where the pole face gap is the smallest. Ferrofluid at the top of the shaft is sealed under static conditions in the same manner as in the rotary shaft seal. When the shaft is moved downward, the shaft motion out of the ferrofluid reservoir produces a thin film of ferrofluid on the shaft. But the magnetic field gradient force exceeds the downward gravity force, the shaft acceleration forces, and any viscous forces, thereby not permitting the ferrofluid to move down the shaft. By making the tapered pole face length greater than the shaft stroke, the ferrofluid film is trapped.

*Westinghouse Advanced Energy Systems Division", Nuclear Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report COO-3043-26, July-September 1976", ERDA Contract EY-76-C-02-3043.

Figure 50-4. Linear Ferrofluid Seal Concept

This concept was tested by modifying one of the rotary shaft seal fixtures as shown in Figure 50-5. The fixture ball bearings were used as linear bushings to guide a reciprocating 0.160 inch dia. rod. The rod was driven by a string attached to an eccentric. The fixture could operate up to a stroke of 0.093 inch at 1800 rpm. The seal used was the same geometry rotary shaft seal as the one presently under test. Even though this type of seal is designed to hold back pressure for a rotating shaft, the pole face air gap does possess a small axial magnetic field gradient, but only over a short distance. By keeping the stroke small, less than 0.06 inch, the seal was run several times for 24 hours against 5 psi. There was no ferrofluid leakage past the seal.

In order for this seal to work properly, the gas pressure on one side must exceed the oil pressure on the other. This, too, was tested and was shown to be the case. By staging several of these seals in series, large static pressure differentials may be held; however, it is not believed that these seals are suited for sealing large fluctuating pressures as each stage must then blow by until the instantaneous pressure is sustained. This causes the loss of ferrofluid in the stages. It is precisely this blowby and fluid loss phenomena which occurred in the fifty stage "magnetic piston" concept tested earlier, and which cast doubt on the validity of that approach.

It is felt that a future prototype system engine could be built making use of this new principle to replace the rollsock seals. It would be possible to dampen out the pressure wave fluctuation of the engine directly over the seal so that the piston and displacer rod magnetic seals would see essentially a small static pressure, duplicating the conditions just described.

LUBRICATION SYSTEM DEVELOPMENT

The ferrofluid filter/oil pump test fixture had accumulated 2026 hours when it was decided to stop the test in order to determine pump wear. Pump flow rate versus pressure drop was determined under identical conditions to those used before the life test. The data reproduced within 1 percent (within the measurement accuracy), indicating that no pump wear or any increase in internal leakage had occurred. This is encouraging information, confirming the assumption that ferrofluid particles do not cause wear of hardened gear surfaces or of soft pump housing surfaces.

Figure 50-5. Linear Ferrofluid Seal Feasibility Test Fixture

The filtering test was then resumed using as much of the original ferrofluid as possible. To date, 2386 hours have been accumulated. There has been no increase in pressure drop across the 2 micron filter.

In an effort to see whether any of the ferrofluid particles had been damaged by the oil pump, a sample of oil was withdrawn and prepared for TEM photographs. A special effort was made to use the same carbon substrate preparation technique as was used when the particle coatings could be seen; as shown in Figure 36-7 of the January-March 1976 Quarterly Progress Report*. As was the case during the previous quarter, we have been unsuccessful in observing the coating although the particles can be seen. It was also desired to accurately measure the coating thickness as this information can be used to predict ferrofluid stability. We do not presently know whether the sample and/or substrate preparation is at fault or whether the electron beam is vaporizing the coating. The next approach will be to change the solvent used for washing away the base oil since it may also have been dissolving the coating.

Figures 50-6, 50-7 and 50-8 are photographs illustrating the first of two lubrication system test fixtures. Fabrication of all parts (including some not shown in the photographs) was complete by the end of this quarter. The assembly phase will commence the first week in January.

*Westinghouse Astronuclear Laboratory, "Nuclear Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report COO-3043-24, January-March 1976", ERDA Contract AT(11-1)-3043.

Figure 50-6. Mark I Prototype Lubrication System Test Fixture Components

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Figure 50-7. Lubrication Test Fixture Crankcase

Figure 50-8. Mark I Prototype Rhombic Drive

TASK 51 - BLOOD PUMP AND COUPLING MECHANISM RESEARCH AND DEVELOPMENT

FLEXIBLE SHAFTS

The examinations of the three development shafts that failed in testing described in the previous Quarterly Progress Report* were completed. The cause of failure of both the 1/4 inch diameter core shaft and the IVBM type shaft with the very high strength music wire core was fretting corrosion of the core wires resulting from loss of lubricant between the core and casing midway between the ends of the shaft assembly. The increased core bending stiffness of these shafts forced the grease lubricant from the center of the shaft toward the ends at a faster rate than expected and caused loss of lubricant at the center of the shaft before the next scheduled regreasing. It is believed that low fatigue strength of the core wires was responsible for the early failure of the shaft with the Elgiloy wire core.

Three new flexible shafts were put on life test. One, a 3/16 inch diameter core shaft of the same type as the development shaft that has been operating since February of 1975, is being tested to establish the reproducibility of the life characteristics of the original shaft. The second, an IVBM type shaft with an Elgiloy wire core similar to the one that failed after 519 hours of operation, is being evaluated to determine whether the low fatigue life of the first Elgiloy core is characteristic of shaft cores made from this wire. The third, an IVBM type shaft with a casing incorporating a teflon liner, is being tested to investigate the influence of the teflon casing liner on the life of an IVBM shaft.

Other shaft life tests in progress were continued. The flexible shaft test hours accumulated and shaft stiffness changes observed during this reporting period, as well as the total test hours to date, are shown in Table 51-1.

*Westinghouse Advanced Energy Systems Division, "Nuclear Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report, COO-3043-26, July-September 1976", ERDA Contract EY-76-C-02-3043.

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TABLE 51-1
FLEXIBLE SHAFT LIFE TEST DATA

Type Shaft	Torque Loading	Test Hours Oct. thru Dec.	Total Test Hours To Date	Stiffness Change (in-oz/deg) Oct. thru Dec.
S.S. White - 3/16 Inch Dia. Core (Shaft No. 35)	Normal ⁽¹⁾ 2.75 x Normal	----- 2297	2088 12,265	----- Decreased from 0.27 to 0.18.
S.S. White - 3/16 Inch Dia. Core (Shaft No. 156)	Normal 2.75 x Normal	2088 258	2088 258	Decreased from 2.35 to 0.25. Not Measured.
S.S. White - IVBM Config- uration (3/16 Inch Dia. Core), Grease Lubricated	Normal 2.75 x Normal	----- 2294	964 7067	----- Decreased from 0.26 to 0.20.
S.S. White - IVBM Config- uration (3/16 Inch Dia. Core), Oil Lubricated	2.75 x Normal	2295	7630	Decreased from 0.19 to 0.18.
S.S. White - IVBM Config- uration, Grease Lubricated, Elgiloy Wire Core	2.75 x Normal	2366	2366	Decreased from 2.9 to 1.1.
S.S. White - IVBM Config- uration, Grease Lubricated, Teflon Lined Casing	2.75 x Normal	417	417	Not Measured.
S.S. White - 1/4 Inch Dia. Core	Normal 1.5 x Normal 2.0 x Normal 2.75 x Normal	----- ----- ----- 1839	4966 481 2013 2635 ⁽²⁾	----- ----- ----- Decreased from 0.33 to 0.30.

(1) Pulsatile with amplitude of 0 to 24 to 0 in-oz and frequency of 4 pulses per second.

(2) Testing discontinued.

Testing of the 1/4 inch diameter core shaft was discontinued after 2635 hours of operation at 175 percent overload conditions so that the test station for this shaft could be used to evaluate the IVBM type shaft with the teflon lined casing.

BLOOD PUMP BEARINGS AND LUBRICANTS

Life testing of standard and special blood pump bearings operating in jojoba and Vitrea 21 oils was continued. The bearing test hours accumulated during this reporting period and the total test hours to date are as follows:

<u>Bearing MPB No.</u>	<u>Bearing Type</u>	<u>Oil</u>	<u>Test Hours Oct. thru Dec.</u>	<u>Total Test Hours To Date</u>
SR4	Ball - Single Row	Vitrea 21	2184	15,720
F155	Ball - Double Row	Vitrea 21	2184	3,504
F155	Ball - Double Row	Jojoba	2184	10,368 ⁽¹⁾
F160	Ball - Double Row	Jojoba	2184	9,336 ⁽²⁾
F338	Roller	Jojoba	2184	5,088

(1) Oil changed from sperm to jojoba after 6,048 test hours.

(2) Oil changed from sperm to jojoba after 5,000 test hours.

No frictional torque change or evidence of significant wear of any of the bearings was observed during this testing.

BLOOD PUMP PERFORMANCE EVALUATIONS

A test procedure has been established to measure the performance of the IVBM Blood Pump that will be implanted with the electrically heated thermal converter. This pump will have

a reduced right ventricle stroke volume and a compliance chamber. The test objective in particular will be to determine pump power requirements at different pump speeds for a range of systemic arterial pressures and cardiac output flows.

Performance characteristic curves for IVBM Blood Pump No. 5 were determined using the mock loop circulation system. A picture of the IVBM blood pump connected to the mock circulation loop is shown in Figure 51-1. The pump ventricles used were supplied by the University of Utah and were tested as schematically shown in Figure 51-2. Fixed systemic valve position and constant arterial pressure cases were considered during the testing for different pulmonary valve positions from wide open to approximately 10 percent open. Also, the transient function performance responses of the Bjork-Shiley (B-S) valves were evaluated for the left ventricle. Location of the B-S valves and the pressure transducers are schematically shown in Figure 51-3. The performance data collected for Blood Pump No. 5 shows good reproducibility.

Figure 51-1. Picture of IVBM Blood Pump/Mock Circulation Loop (Series Connection).

Figure 51-2. Schematic of Mock Circulation Loop/Pump Arrangement
(Dots Indicate Pressure Transducer Locations) Series Connection

Figure 51-3. Schematic of Left Ventricle Showing Transducer and Bjork-Shiley Valve Locations

TASK 52 - WASTE HEAT REJECTION RESEARCH AND DEVELOPMENT

A pumped cooling system, using circulating blood pump lubricating oil as a coolant has been identified as a prime candidate for use in the Mark I system. Because of potential advantages of TOPS (no mechanical power required; smaller coolant lines through the diaphragm) development will continue on it as an alternative approach.

PUMPED OIL COOLANT LOOP

Design specifications for the oil coolant pump were prepared and issued. Highlights of the specification include:

- The pump should be capable of delivering $0.45 \text{ in}^3/\text{sec}$ of oil against a pressure drop of 1.0 psi while the blood pump is operating at 120 bpm.
- The pump shall consume less than 0.20 watts of mechanical power while operating under the conditions described above.
- The pump shall be incorporated into the right angle gear box housing, and shall not increase the envelope dimensions of the converter - gear box assembly.
- The weight of the pump shall be less than 20 grams.

The specification includes a 0.4 psi allowance for pressure drop through the converter cooler. Using this value, a parametric study of the oil side cold end heat exchanger for the Mark I converter was carried out. The heat exchanger consists of two semi-circular sets of cooling fins wrapped around the outside of the converter cooler. Fourteen and sixteen fins, with 0.030 inch wide flow passages and varying heights were examined, with the results shown in Figures 52-1 and 52-2. The design point selected (fin height = 0.145 inches, 14 fins) results in a satisfactorily compact heat exchanger, with acceptable hydraulic and thermal performance. The fin thickness for this design is 0.020 inches, and the design can be fabricated readily.

Figure 52-1. Film Temperature Drop as a Function of Fin Height

Figure 52-2. Pressure Drop as a Function of Fin Height

Increasing the number of fins to 16 would allow a more compact design for the same performance, or allow improvements in hydraulic and thermal performance. The fin thickness would be reduced to 0.014 inch, and it was felt this was too thin to manufacture conveniently. Increasing the passage width from 0.030 inch to 0.032 inches significantly improves the pressure drop characteristics of the cooler, but decreases the thermal performance, as shown by the single points shown in the figures for 14 fins.

Two preliminary gear pump designs were completed. The gears are supported by ball bearings in one design and journal bearings in the other.

THERMAL OSCILLATION PUMP (TOP)

All parts and materials for planned TOP experiments have been received. Thermal performance characteristics for the new design configuration will be determined and the test set-up for the new configuration, which does not use a pressurizer, is currently being assembled. The following working fluids have been tentatively identified for testing: trichlorotrifluoroethane, acetone and propionaldehyde.

A procedure has been established for determining an upper limit liquid volume for the new TOP configuration. This procedure ensures sufficient liquid for the TOP system to be operable under any system orientation and/or mode of operation. The amount of liquid volume for the TOP system is determined based upon the assumption that some liquid is always necessary in the vaporizer for the system to function reliably. Experiments are planned to verify the analytical estimate of the upper limit liquid volume.

TASK 53 - SYSTEM STUDIES

MARK I INTERFACES

Detailed examination of the Mark I Prototype system interfaces is continuing. Several meetings were held between Westinghouse Advanced Energy Systems Division and the Philips Laboratories to arrive at mutually satisfactory designs for the interfaces. Interface areas include:

- Review of the System and Converter Specification.
- Insulation Package/Thermal Converter Interface as described in Task 61.
- Magnetic Coupling/Gear Box Interface as described hereafter.
- Circulating oil coolant system and alternate coolant system as described in Task 52 and below.
- Hot Finger/Heat Source Interface as described hereafter.

To facilitate design of the system, mock-ups of the thermal converter are being prepared. It is planned that these will be kept up to date and will incorporate all external attachments to the converter as designs of the system components are completed.

As a preliminary step in the examination of the hot finger/heat source interface, a study was undertaken to estimate the requirements for the joint between the Mark I converter hot finger and the heat source. The contact heat transfer coefficient of this joint must be sufficiently large to enable the heat released by the source to be transported across the joint without overheating the heat source. Estimates of the surface finish required on the mating surfaces as a function of the contact pressure at the joint have been prepared for various temperature conditions at the joint. In order to maximize the compliance of the interface,

a thin (10 - 20 mil) layer of pure nickel was assumed to be metallurgically bonded to the top of the hot finger. In the absence of test data for the materials and conditions of interest, a correlation developed by F. Nejat Veziroglu* was used as the basis for the analysis.

Under normal operation, the hot finger temperature is not expected to exceed 1250°F. Figure 53-1 shows the required surface finish as a function of contact pressure for various heat source clad temperatures at the interface, with the hot finger at 1250°F.

A preliminary analysis of the heat source indicates that bottom clad temperature should be slightly in excess of 1450°F with the hot spot clad temperature held to its specified operating temperature of 1565°F. Under these conditions, surface finish of 8μ inches on each surface, combined with a contact pressure of 12 psi is expected to provide adequate heat transfer.

Under extreme overload conditions at the beginning of use, the hot finger temperature can be expected to rise as high as 1330°F. This condition is not expected to last for more than 1% of the total lifetime of the system. If short periods of 80°F over temperature can be tolerated by the heat source, the curves given in Figure 53-1 can be used as a design basis. If the peak heat source cannot ever exceed 1565°F, the curves shown in Figures 53-2, for the hot finger at 1339°F must be used. In this case, the design condition for the interface can be expected to be about 20 psi with an 8μ-inch finish on the mating parts.

The values of surface finish must include waviness and other microscopic irregularities as well as the microscopic roughness of the surfaces. In addition, it must be pointed out that contact coefficients, particularly under vacuum conditions, are difficult to predict, and the conclusions drawn from this analysis must be verified by an experimental program prior to incorporation into a final design.

* Paper 67-317 Presented at the AIAA Thermo-Physics Specialist Conference New Orleans, La., April 17-20, 1967.

Figure 53-1. Required Surface Finish as a Function of Contact Pressure and Heat Source Bottom Clad Temperature with Hot Finger at 1250°F

Figure 53-2. Required Surface Finish as a Function of Contact Pressure and Heat Source Bottom Clad Temperature with Hot Finger at 1330°F

With respect to the waste heat rejection system, a decision for the Mark I prototype was made to proceed with a circulating oil loop as the prime heat rejection approach. Design provisions are currently being made to incorporate a gear type pump onto the right angle gearbox. The mechanically driven pump will circulate blood pump oil as a coolant for the thermal converter. Work on the thermal oscillating pump heat rejection system is being continued as a back-up for Mark I. Figure 53-3 shows the latest Heat Exchanger design at the thermal converter/TOP interface.

A preliminary design study of the converter gear box was performed to establish the converter gear box/magnetic coupling interface shown in Figure 53-4. In this study, a set of miter gears is used to obtain a 90° change in the direction of the axis of rotation, and a set of spur gears is used to reduce the speed of rotation from 2540 rpm at the input shaft to 900 rpm at the output shaft. The gear box oil is excluded from the magnetic coupling cavity by a screw type seal. Because a gear pump driven coolant loop was selected as the primary waste heat transport system for the thermal converter, provisions were incorporated in the gear box for driving a gear pump mounted on one of the gear box walls.

The insulation package/thermal converter interface is shown in Figure 53-5. A discussion of the design features of this interface is presented under Task 61 of this report.

Figure 53-3. Thermal Converter/TOP Interface for the Mark I Prototype

Figure 53-4. Mark I Prototype Converter Gear Box/Magnetic Coupling Interface

Figure 53-5. Mark I Prototype Insulation Package/Thermal Converter Interface

TASK 54 - IVBM MODIFICATION DESIGN

BLOOD PUMP

A compliance chamber is required for the IVBM blood pump with a reduced right ventricle stroke volume when the pump is operated with an oil filled mechanism cavity. Because the volume of oil enclosed in the mechanism cavity decreases during half of the pumping cycle and increases during the other half of the cycle, a compliance chamber must be added to the mechanism cavity to provide a total oil reservoir volume that remains unchanged throughout the entire pumping cycle.

The compliance chamber shown in Figure 54-1 was designed to accommodate a right ventricle stroke volume reduction of 9 cc. The chamber consists of a silastic diaphragm that provides the required volume compliance, a component which serves as the chamber housing and connects the chamber to the pump housing, and a cover that protects the diaphragm from handling damage or contact loading in the thorax. The diaphragm is made from silastic sheet and is molded in the distended shape that it will assume when the chamber is filled with oil. When oil flows from the chamber to the blood pump, the diaphragm collapses to reduce the chamber volume. Pressure communication between the thorax and diaphragm is accomplished through several holes in the cover.

All the detail drawings required to manufacture the chamber components were completed.

614393-1C

Figure 54-1. IVBM Blood Pump Compliance Chamber

TASK 55 - IVBM FABRICATION

The electrically heated IVBM insulation package was fabricated, outgassed, and pinched off during this quarter. The heater, described in the previous quarterly progress report*, was installed in the IVBM hot finger.

Figures 55-1 to 55-3 show several steps during the fabrication sequence. The heater was held in place by an Inconel 625 retaining ring. The high purity Al_2O_3 disc was fabricated to accept the two heater leads and the two wires which tied down the insulator for the electrical leads. The heater leads were brazed to a sheathed twin conductor nickel electrical lead.

Figure 55-4 shows a closeup of the connection between the heater lead and the sheathed electrical lead at the heater. Figure 55-5 shows a closeup of the Al_2O_3 electrical insulator in place over the heater lead-electrical lead joints.

A closeup of thermocouple-feedthrough connections is shown in Figure 55-6. The 0.006 inch diameter thermocouple leads are exposed by stripping away the thermocouple sheath. The Al_2O_3 double bore ceramic is used to prevent the thermocouple leads from shorting to the sheath. These 0.006 inch diameter leads are then brazed to ceramic-metal feedthroughs as shown. The brazed electrical lead-feedthrough connection is shown in Figure 55-7.

The electrically heated IVBM was mounted on a test stand which was designed to permit pressurizing the internal engine cylinder while the system was hot. By monitoring the insulation vacuum, as is normally done during bakeout, the vacuum integrity of the thin wall cylinder was

*Westinghouse Advanced Energy Systems Division, "Nuclear Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report COO-3043-26, July-September 1976", ERDA Contract EY-76-C-02-3043.

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Figure 55-1. IVBM Electrical Heater Installed in Hot Finger and Contained by a Retainer Ring

Figure 55-2. Al_2O_3 Disc Insulation Showing Heater Leads and Wires to Hold Al_2O_3 Electrical Insulation to Heater Leads

Figure 55-3. Finished Heater Installation

Figure 55-4. Closeup of
Heater Leads-Electrical
Lead Joint at Heater

Al_2O_3 Sleeve
Sheathed Electrical Lead

Figure 55-5. Closeup of
Finished Heater Lead-
Electrical Lead Joint with
 Al_2O_3 Insulation in Place

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Figure 55-6. Closeup of the Thermocouple-Feedthrough Connection

Feedthroughs

Figure 55-7. Closeup of the Electrical Lead-Feedthrough Connection

verified under simulated operating conditions. These internal pressurization tests were performed on the electrically heated IVBM and no helium leaks were detected.

The time-temperature bakeout history of the electrically heated IVBM insulation package is shown in Figure 55-8 and is compared with the bakeout histories of the previous IVBM system insulation packages. Table 55-1 lists the bakeout time in days, the days under static vacuum prior to pinchoff, and the pinchoff date. As of the latest testing, both the IVBM No. 2 rebuild and the electrically heated IVBM are still operable insulation systems.

After all bakeout was completed, a final leak check just prior to pinchoff revealed a very small leak in the area of a ceramic metal thermocouple feedthrough. This leak was only 8×10^{-12} cc(STP)/sec for helium. The value was determined by measuring the mass spectrometer sensitivity to helium using a calibrated helium leak. This is equivalent to a 3×10^{-12} cc/sec air leak. A leak of this magnitude would not be detectable for 3.1 years based on thermal performance since this is the time required to reach 10^{-3} mm Hg pressure in the insulation cavity.

The leak was repaired using an ultrahigh vacuum epoxy sealing compound. After sealing, no detectable leaks were found. The system was successfully pinched off on December 8, 1976.

Finished external electrical and instrumentation connections to the hot finger are shown in Figure 55-9. The final configuration of the electrical hot finger after the lead connections were potted is shown in Figure 55-10. The thermal performance of this system is very close to that for IVBM No. 1 previously reported*.

*Westinghouse Astronuclear Laboratory, "Nuclear Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report COO-3043-23, October-December 1975," (p. 41-6), ERDA Contract E(11-1)-3043.

55-6

Figure 55-8. Bakeout History for the IVBM System Insulation Packages

TABLE 55-1
IVBM PACKAGE PROCESSING HISTORIES

Insulation Package	Bakeout Time (days)	Static Vacuum Time Prior to Pinchoff (days)	Pinchoff Date
IVBM No. 1	20	-	3/24/75
IVBM No. 2	29	2	12/10/75
IVBM No. 2 Rebuild	21	7	2/16/76
IVBM Electrical	19	4	12/8/76

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Figure 55-9. Bottom View of Electrical IVBM Hot Finger Showing Thermocouple and Electrical Heater Connections Prior to Epoxy Potting

Pinchoff Cover

Thermocouple Lead

Electrical Lead

Figure 55-10. Bottom View After Potting

TASK 56 - IVBM PERFORMANCE TESTING

The rollsock failures on IVBM thermal converter No. 2, reported in the previous quarterly report*, were attributed to burrs on the small rollsock clamping rings. These burrs were locally abrading the surface of the seal. The rings were repolished and no recurrence of this problem appeared thereafter. The converter was run continuously for 146 hours during which the output power was 4.9 to 5.0 watts at 600 rpm, 170 psig charge pressure. Following this run, the unit was delivered to Westinghouse for further testing.

The implantable electrically heated IVBM converter was assembled late in the reporting period and testing of the converter was initiated. Tests performed to date indicate that the converter is capable of producing 4.7 watts of mechanical power at the output of the right angle gear box. During this run, 33.3 watts of heat was supplied to the heater and the temperature of the thin wall portion of the hot finger was 1460°F. These results are comparable with the results obtained with this converter at Philips. The power output is somewhat lower than that reported in the January-March 1976 Progress Report** for similar operating conditions, and the hot finger temperature is slightly higher. The higher operating temperature is tentatively ascribed to improved performance of the static insulation package installed on the converter.

The most likely cause of the reduced power output is increased losses in the rhombic drive, which had been damaged and had to be rebuilt in the interim. The rebuilt drive train performance may improve as the components run in with additional testing.

*Westinghouse Advanced Energy Systems Division, "Nuclear Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report, COO-3043-26, July-September 1976", ERDA Contract EY-76-C-02-3043.

**Westinghouse Astronuclear Laboratory, "Nuclear Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report, COO-3043-24, January-March 1976", ERDA Contract E(11-1)-3043.

A significant number of problems have hampered checkout of IVBM thermal converter No. 1. While the converter was in storage this summer swelling of the piston ring occurred, apparently due to oil contamination. A new piston ring was installed. Before testing could begin, the control valve oil line broke at the braze joint where it enters the valve. This break was successfully repaired. After reassembly of the converter, argon leakage was found at the fitting which joins the working space connection on the control valve to the engine. The entire tube from the engine cooler to the control valve was replaced, and the fitting was upgraded to a standard 1/8 inch Swagelock.

IVBM thermal converter No. 1 is presently undergoing run-in of the new piston ring. When thoroughly checked out, the unit will become a spare for system implantation experiments.

The IVBM system will undergo a sterilization gas treatment prior to animal implantation. This will involve a 12 hour exposure to Oxyfume-12, a commercial gas sterilant*, carried out in a closed laboratory glove box. Ethylene oxide, the active and highly toxic component of the sterilant gas, is readily absorbed by many polymer materials; as such, its permeation through the gloves and gaskets of the glove box is to be expected to some degree. Data were gathered to estimate the permeation rate of ethylene oxide through neoprene rubber gloves.

Permeation was determined by measuring the rate of pressure decay in a chamber fitted with a neoprene rubber diaphragm and filled with Oxyfume-12. The experimental apparatus is schematically described in Figure 56-1. The test chamber was purged with sterilant gas to displace air then sealed at a pressure slightly above atmospheric. (A pressure of Oxyfume-12 1 to 10 mm (Hg) above atmospheric will be maintained for a period of 12 hours to accomplish IVBM sterilization). The decay of chamber pressure was then recorded as a function of time to yield a measurement of the rate of Oxyfume loss occurring through the neoprene diaphragm. The test chamber was held at room temperature in air.

*A Union Carbide Corp. product containing 12 percent ethylene oxide and 88 percent Freon 12.

Figure 56-1. Schematic of Oxyfume-12 Neoprene Permeation Test System

Pressure was measured using a capacitance gage. Movements of the sensing disc in the pressure gage and that of the neoprene rubber test diaphragm were so small over the pressure ranges examined that the chamber volume could be assumed constant. The volume of the chamber, including connections to valves and the pressure gage, was 246 cm^3 .

An ultra-high vacuum stainless steel pipe tee was used for the test chamber. The neoprene rubber test diaphragm was attached to one leg of the tee. Valving and pressure sensing connections were made at the other tee locations. Leakage of gas other than that occurring through the neoprene diaphragm was eliminated by using copper gaskets to seal flanges, brazed tube-to-flange connections, and metal seat valves. No pressure loss was observed in a leak check of the test system made by filling it with air to a 50 mm (Hg) gage pressure and holding for 16 hours.

Test material was cut from a neoprene glove identical to those on the IVBM sterilization box. A 12 cm^2 area of surface was exposed to the Oxyfume gas during testing. The material was 0.046 cm thick. The test diaphragms were inspected for presence of holes with a high voltage arc probe (results negative) prior to use.

Two permeation tests were run, results of which are described in Figure 56-2. A different neoprene diaphragm was used on each test. The first test was started at an Oxyfume-12 gage pressure of 44 mm (Hg) and run for a period of 22 hours; the second was started at 18 mm (Hg) pressure and run for 12 hours. In both cases, pressure losses occurred initially at a rate of $\sim 6 \text{ mm/hour}$ and decreased to a steady state rate of $\sim 1 \text{ mm/hour}$ after passage of 2 to 4 hours. Pressure losses averaged 1.5 mm/hour over a 12 hour period.

Estimates of the ethylene oxide flux passing through the neoprene diaphragm were made using the data presented in Figure 56-2 and the ideal gas equation. It was assumed that only the

Figure 56-2. Pressure-Time Curves for Oxyfume-12 Permeation through Neoprene Rubber

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ethylene oxide component of the Oxyfume-12 sterilant diffused through the rubber. Initial, steady state, and average ethylene oxide flux levels are:

$$J \text{ (initial)} = 0.3 \text{ mgs (eto)/cm}^2 \text{ hr}$$

$$J \text{ (steady state)} = 0.05 \text{ mgs (eto)/cm}^2 \text{ hr}$$

$$J \text{ (12 hr. avg.)} = 0.07 \text{ mgs (eto)/cm}^2 \text{ hr}$$

The surface area of a neoprene glove used on the IVBM sterilization box is 3310 cm^2 .

Estimates of amounts of ethylene oxide, which could leak through a glove, are:

$$q \text{ (initial)} = .99 \text{ gms (eto)/hr/glove}$$

$$q \text{ (steady state)} = .17 \text{ gms (eto)/hr/glove}$$

$$q \text{ (12 hr. avg.)} = .23 \text{ gms (eto)/hr/glove}$$

These quantities convert to the following gas volumes for STP conditions:

$$V \text{ (initial)} = 548 \text{ cm}^3 \text{ (eto)/hr/glove}$$

$$V \text{ (steady state)} = 94 \text{ cm}^3 \text{ (eto)/hr/glove}$$

$$V \text{ (12 hr. avg.)} = 127 \text{ cm}^3 \text{ (eto)/hr/glove}$$

TASK 57 - IVBM SYSTEM LIFE TESTING

BLOOD PUMP

When the bellows pump coolant circulating sytem development effort was terminated, Blood Pump No. 6, which had been used to test the system, was put on life test. This pump has operated 2332 hours to date. No evidence of ventricle leakage or change in pump performance was observed during this testing.

IMPLANTABLE BLOOD PUMP MOTOR DRIVE

A 2,000 hour endurance test of the first Westinghouse Advanced Energy Systems Division implantable blood pump electric drive was performed. No change in motor performance was observed throughout this test.

TASK 58 - FIELD SUPPORT

BLOOD PUMP IMPLANT EXPERIMENTS

Since balanced and relatively low atrial pressures were achieved throughout ERDA Implant Experiment No. 33 by using a modified IVBM blood pump with a reduced right ventricle stroke volume, two additional IVBM blood pumps were modified in the same manner and shipped to the University of Utah for in vivo testing. All of these pumps were cleaned and rebuilt one or more times after blood leaked into the mechanism cavity during implant experiments. The leakage occurred through ventricle diaphragm perforations caused by improperly installed pusher cup skirt rollpins or improper seating of the diaphragm on the pusher cup and subsequent pinching of the diaphragm by a yoke housing roller. In the cases where the diaphragm was damaged because it was not properly seated on the pusher cup, a modified diaphragm secured only to the ventricle dome was used. Because it was concluded that the procedure used at Utah to seat ventricle diaphragms of this type would have to be developed further to avoid improper seating of the diaphragm, only ventricles with diaphragms secured to both the ventricle dome and pusher cup are now being used at Utah.

Two Westinghouse Advanced Energy Systems Division implantable blood pump electric motor drives and several spare flexible shafts were shipped to the University of Utah for use in blood pump implant experiments.

TASK 59 - RELIABILITY AND QUALITY ASSURANCE

Quality Engineering and Quality Control activities continued at a level consistent with the program procurement and fabrication activities. Routine receiving and in-house fabrication inspections were conducted as required.

TASK 61 - MARK I THERMAL INSULATION DESIGN

A source for the titanium alloy feedthroughs has been found. Based on the available feedthrough geometry and size, a layout of feedthrough locations for an electrically heated MARK I hot finger was prepared and is currently being reviewed.

The MK-I thermal converter-thermal insulation interface is shown schematically in Figure 53-5. The feedthroughs were routed radially from the base flange to ensure clearance for unobstructed installation of one of several proposed cooling systems below the flange. The four circumferential locations for each pair of feedthroughs and the pinchoff port are designed to extend over the corners of the thermal converter crank case. In this position, the dimensioned envelope established for the thermal converter will not be violated. The dimension of the feedthrough ports and their location are dictated by the size of the commercially available titanium feedthroughs. Routing and fastening of thermocouples and heater leads to these feedthroughs will be those successfully utilized for the electrical IVBM system. The isotope vent connection will replace the electrical power feedthroughs when an isotope powered system is built.

TASK 62 - MARK I THERMAL CONVERTER DESIGN

THERMAL CONVERTER PERFORMANCE

A reoptimization of the Mark I thermal converter thermodynamic design was completed. The discussion below presents the steady-state and transient performance characteristics expected from this new design.

Figure 62-1a presents the steady-state shaft power output from the thermal converter versus crankshaft speed for both the original and new designs. The extreme cases of 33.0 W, Beginning of Life (BOL), and 30.5 W, End of Life (EOL), heat source inventory are shown. The significant improvement in output power is attributable to two design changes. The first involves the allowable thin wall operating temperature. Previously, based on BOL performance, the thin wall temperature had been limited to 925°K at 600 rpm, and creep rupture characteristics had been evaluated assuming ten years at that temperature. In reality, a less conservative approach is justified since the temperature of the hot side steadily decreases as the heat source is depleted, (see Figure 62-1b). So, where previously a steady temperature of 925°K at 600 rpm had been assumed, now 950°K at 630 rpm, BOL is assumed. The creep calculated over ten years for the latter assumption is no worse than previously calculated with the former assumption. This design change has the effect of shifting the output curves "up"; the gain is most dramatic at the EOL condition.

The second change was to redesign the regenerator filling to reduce its flow resistance. While a small net increase in output power at 600 rpm is realized, the most significant result of this change is reduction of the rate at which the output power falls off with increasing speed. This is responsible for the shallower negative slope of the Case 2 curves in Figure 62-1a. The net efficiency of this engine at 600 rpm, BOL, is calculated to be better than 22 percent.

Figure 62-2 is a plot summarizing the transient performance characteristics of the Mark I thermal converter. Output power is plotted versus crankshaft speed for given thin wall temperatures.

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a. Shaft Output Power vs. Crankshaft Speed

b. Thin Wall Temperature vs. Crankshaft Speed

Figure 62-1. Steady-State Performance Characteristics of Mark I Prototype Thermal Converter

SHAFT POWER VS. THIN-WALL TEMP. AND CRANKSHAFT SPEED FOR REOPTIMIZED DESIGN

Figure 62-2. Transient Performance Characteristics of Mark I Prototype Thermal Converter

Figure 62-3 superimposes the new steady state output versus speed characteristics for the thermal converter at BOL and EOL, and the blood pump load versus speed characteristics with an engine crankshaft to blood pump crankshaft speed ratio of 5.3. Note that at 120 beats per minute (bpm), the engine crankshaft speed is 636 rpm. The power output required at 100 mm Hg, 120 bpm is 5.54 watts. At BOL, the power available from the thermal converter at 636 rpm is 6.88 watts. This is more than required by the blood pump; therefore, the control system must dissipate 1.34 watts. At EOL, the power available from the thermal converter is 5.47 watts. This is close enough to satisfy the blood pump load (in reality, the blood pump would slow from 120 bpm to 119.5 bpm). Thus, under these conditions, the system can deliver 12 ℓ /min at 100 mm Hg over the full ten year design life.

In the following discussion, it is assumed that the equilibrium thin wall temperature either remains constant or increases slightly as power is dissipated by the short circuit control system. The overload discussed is a sudden rise in mean aortic pressure (PA) to 125 mm Hg.

At BOL and 636 rpm, the steady state temperature of the thin wall is 942°K (from Figure 62-1b). The power available from the thermal converter is 6.88 watts, while the overload demands only 6.60 watts at 636 rpm. Therefore, the overload can be met instantly without any reduction in system speed.

After two years, the thin-wall temperature would have decreased to 922°K (interpolating Figure 62-1b), and the output power available at 636 rpm would be 6.6 watts. However, since the same overload at 636 rpm only requires 6.6 watts, the overload can still be met instantly without a decrease in speed.

As time progresses, the thin wall temperature will continually decrease and the available power at 636 rpm will not be sufficient to meet the overload. The system will equilibrate at a slower speed, and the delivery will fall below 12 ℓ /min. This is acceptable, as stated in the thermal converter specification; however, no speed slower than 600 rpm is permissible for the thermal converter.

ENGINE SHAFT POWER & BLOOD PUMP SHAFT LOAD VS. SPEED AND MEAN AORTIC PRESSURE

Figure 62-3. Mark I Prototype System Performance - Nominal Mean Aortic Pressures

We can calculate how long the overload can be met without slowing the converter to less than 600 rpm. Figure 62-3 shows that a PA of 125 mm requires an output power of 5.95 watts at 600 rpm. Figure 62-2 demonstrates that a minimum thin wall temperature of 890°K is required to produce 5.95 watts at 600 rpm. Finally, interpolation of Figure 62-1b at 636 rpm shows that the thermal converter would have cooled down to this temperature in 5.5 years.

After 5.5 years, if while operating under nominal load at 636 rpm a sudden overload of 125 mm Hg were to occur, there would be insufficient power available to prevent slowing the converter to below its minimum allowable speed. However, a gradual increase in PA (hours or days) to 125 mm Hg can be met since the temperature of the thermal converter would have time to re-equilibrate as the speed decreases. This would be a quasi-steady state, rather than a transient situation. As can be seen in Figure 62-3, sufficient output exists, even at EOL, to meet the 125 mm Hg overload.

Figure 62-4 illustrates that the system can be set up to satisfy far higher mean aortic pressures by adjusting the ratio of thermal converter to blood pump speed to a higher value (via the gear box). However, this change will always result in a reduction of maximum available blood flow to a value less than 12 l/min.

HEAT REJECTION INTERFACE

Effort in this area at Philips Laboratories concentrated on arriving at a design for an oil cooler and circulating pump compatible with the design specification.

A heat transfer/pressure drop analysis resulted in the selection of an oil side heat exchanger configuration consisting of 14 parallel, semi-circular flow channels (see Task 52).

Design layouts of this cooler were prepared. The method of integration of the coolant tubes and working space pressure taps have been worked out, and the structural design of the cooler and its maintainable interface with the crankcase have been completed.

Figure 62-4. Mark I Prototype System Performance - High Mean Aortic Pressures

Alternative designs for three types of coolant pumps were prepared and submitted to Westinghouse. These included 15 hz gear pumps, 40 hz gear pumps, and 40 hz screw pumps. All were sized to deliver $0.45 \text{ in}^3/\text{sec}$ of oil at 1.0 psi. As a class, the 15 hz gear pumps were the most efficient, and were selected to be the pump of choice on that basis. Table 62-1 shows the selection of 15 hz pumps presented to Westinghouse. The 10 tooth, 32 diametral pitch pump was chosen for further development.

HEAT SOURCE/HOT FINGER INTERFACE

Philips Laboratories investigated methods of mechanically securing the Mark I PuO heat source to the hot finger. It was assumed that LASL would prefer to retain the basic bench model containment system and not complicate the design by incorporating load carrying fastening contrivances into the heat source and that the final Mark I heat source would be a simple right circular cylinder.

The type of restraint system recommended was one wherein a third member (in addition to the heat source and hot finger) is used to exert an axial force on the top of the heat source, pressing it against the top of the hot finger. The reaction force would be carried by the hot finger. The heat source would be piloted into the end of the hot finger to constrain radial movement.

The key to the success of this approach is the selection of a material for the restraining member which has a coefficient of thermal expansion lower than that of the heat source. In addition, this material must have good yield and creep strength at nominal operating temperatures, and sufficient strength at temperatures up to 2500°F to ensure that the heat source will not fall off before the Udimet thin wall failed. The requirement involving the coefficient of thermal expansion is needed to ensure that as the temperature of the assembly increases from room temperature to normal operating temperature, the restraint member is placed in tension and interfacial pressure between the heat source and hot finger is maintained for good heat transfer.

TABLE 62-1

ROUGH SIZING OF GEARS AND PUMP HOUSINGS FOR 15 Hz PUMP SPEED

N = number of teeth per gear
 DP = diametral pitch
 PD = pitch diameter, inch
 L = face width, inch

N	DP	PD	L	Inches			Power Input, mw
				X	Y	Z	
8	16	.5000	.165	1.379	.879	.365	469.
8	20	.4000	.260	1.154	.754	.460	289.
8	24	.3333	.370	1.004	.671	.570	214.
8	32	.2500	.660	.817	.567	.860	159.
9	16	.5625	.150	1.505	.942	.350	619.
9	20	.4500	.230	1.254	.804	.430	358.
9	24	.375	.330	1.087	.712	.530	254.
9	32	.2813	.585	.879	.598	.785	177.
10	16	.6250	.135	1.629	1.004	.335	806.
10	20	.5000	.210	1.354	.854	.410	446.
10	24	.4167	.300	1.171	.754	.500	304.
10	32	.3125	.530	.942	.630	.730	200.
12	16	.750	.110	1.879	1.129	.310	1,327.
12	20	.600	.170	1.554	.954	.370	678.
12	24	.500	.250	1.337	.837	.450	436.
12	32	.375	.440	1.067	.692	.640	258.

The columbium alloys are attractive candidates for the restraint member since they satisfy the requirement that their coefficient of thermal expansion is less than that of platinum-rhodium, they have good yield and creep strength at normal hot finger temperatures, and would retain their strength for a reasonable time at temperatures up to 2500°F. Any tendency to oxidation attack should not be detrimental since the member will be in a vacuum. The material need not bear directly on the heat source or hot finger (an intermediate washer could be used) if there is fear that the columbium/platinum-rhodium or columbium/Udimet combinations may result in the formation of undesirable intermetallic compounds.

Sketches were prepared showing various methods of securing such a restraint member to the tip of the hot finger. These methods include a quick turn threaded interface, a swaged design, and one using finger springs.

These suggestions are presently under consideration.

INSTRUMENTATION FEEDTHROUGHS

A study was performed to determine those areas of the thermal converter outer envelope available for the routing of instrumentation lines and the static vacuum pinchoff port. A full scale wooden mockup of the thermal converter was prepared for this purpose. As a result of this study, it was decided to restrict the area between the top of the crankcase and base flange of the hot finger for coolant lines, access to the four cooler mounting bolts, and control valve connections. The "lower" end of the outer cylindrical surface of the hot finger/vacuum insulation assembly (an integral portion of the hot finger flange) will be used for instrumentation feedthroughs (see Task 61).

MAGNETIC COUPLING/GEAR BOX INTERFACE

Alternative designs for a screw viscosity oil exclusion seal were prepared by Philips Laboratories. The seal is intended to prevent gear box oil from entering the magnetic coupling armature cavity. Details of the selected seal design are presented under Task 50.

STARTER COUPLING INTERFACE

While the starter coupling itself has been designed and preliminarily tested (see Task 50), the interface between any needed alignment devices and the engine crankcase has not yet been defined. This area is presently under study.

THERMAL CONVERTER STRUCTURAL DESIGN

The baseline engine structural configuration was selected this quarter. The crankcase is to be EDM roughed in one piece instead of fabricating a multi-piece brazement. With this design, only the crankcase top plate (buffer space cover) need be brazed which should result in improved leakage reliability and a lower final weight. The magnetic coupling membrane will be welded in place at the output shaft end, and the crankcase oil cover will be secured with screws rather than welded. It was decided to use small external bosses for the crankcase cover screws rather than increase the overall dimensions of the crankcase. The control valve will be mounted to a manifold using two screws. This eliminates the large, cumbersome fittings which exist on the IVBM while improving reliability. Manifold reliability using O-ring seals should exceed that for the IVBM where oil lines occasionally broke, and fittings occasionally leaked. The valve has been located above the flywheel housing and parallel to the hot finger axis. A layout drawing of the control valve was made in order to firm up the manifold area. The crankcase EDM drawing is in process.

The cooler drawing has been completed and a final layout of the lower hot finger section is in process. Details of the rhombic drive parts are nearly completed.