Westinghouse Astronuclear Laboratory

NUCLEAR-POWERED ARTIFICIAL HEART PROTOTYPE SYSTEM DEVELOPMENT PROGRAM

QUARTERLY PROGRESS
REPORT

OCTOBER-DECEMBER 1973

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NUCLEAR-POWERED ARTIFICIAL HEART
PROTOTYPE SYSTEM DEVELOPMENT PROGRAM

PHASE III
QUARTERLY PROGRESS REPORT

October 1, 1973 through December 31, 1973

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INTRODUCTION AND SUMMARY

The Nuclear-Powered Artificial Heart Prototype System Development Program is being conducted by the Westinghouse Astronuclear Laboratory and its major subcontractor, the Philips Laboratories division of the North American Philips Corporation as Phase III of the USAEC's Artificial Heart Program under Contract No. AT(11-1)-3043. This report presents the technical progress and accomplishments on the active program tasks for the period from October 1, 1973 through December 31, 1973.

Thermal insulation component development continued during the reporting period. Outgassing tests were performed and analyzed and outgassing procedures developed. Work on spacer and cup technology development continued. The outgassing evaluation test apparatus was modified and design of the thermal evaluation test apparatus was initiated. Life test data from the 20-cup pack were analyzed and tests were initiated with the 40-cup pack, using the guarded orientation test apparatus. Technical progress on each of these efforts is reported under Task 1.

In Task 2, thermal converter component development efforts continued on five of the components originally chosen for further development: the hot finger, the stationary regenerator, the control system, the bellows seals, and the oil pump. The status and the detail of efforts on each of these components is presented under Task 2.

Blood pump component development efforts continued in Task 3. During the quarter, analyses of planetary gear test and the liquid filled drive test from the previous reporting period were completed. Similar tests using the Evoloid gear reducer were performed and are being analyzed. Initial results from the soaking of silicone rubber in oil were obtained; these tests are continuing.

Auxiliary component development, Task 4, had emphasis during the quarter on the waste heat coolant loop and coolant pump. For the Intermediate System, this function is expected to be

performed by a flexible water heat pipe. Because of the shorter time available for development of components for the IVBM, it was decided to use a pumped loop system for the IVBM. The prime circulating pump candidate for the IVBM is a bellows type pump located inside the blood pump mechanism.

Design of the thermal insulation for the electrically heated IVBM was initiated under Task 5. A vendor for the nickel foil cups was selected.

The conceptual and the preliminary designs of the IVBM thermal converter were completed. Detailing was initiated on the hot finger/displacer/heat exchanger assembly, as well as on the drive, enclosure, and several of the accessories. The detailing of the first item was completed, and is discussed in Task 6; the other three items are still in the detailing stage.

Design work on the IVBM blood pump drive mechanism was initiated. The design status of the mechanism per se is discussed under Task 7 along with design considerations for the bellows coolant pump that is planned for the IVBM.

Under Task 8, the conceptual design of the Intermediate System was completed during the reporting period and a topical report was prepared describing this design along with the factors that led to selection of design features. System integration on the IVBM was initiated during the quarter and a status review is presented under this task.

Reliability activities during the quarter centered about design reviews for the IVBM and the Intermediate System, upgrading the Failure Mode Effects Analysis for the Intermediate System, component testing and component failure analysis. These efforts are presented in Task 9.

The Phase II bench model Thermal Converter No. 2 was placed on life test, Task 15, on October 22. The converter operated initially without incident. During early November, several of the life test support equipment components failed. During these equipment failures, the thermal converter continued to operate satisfactorily. On November 21, the converter ceased operation with 729 hours cumulative time at power. Test instrumentation indi-

cated the failure was caused by a loss of engine compression. Later teardown inspection of the thermal converter revealed a perforation in the inner roll sock, confirming the test instrument assessment.

Under Task 16, support was provided for another in-vivo blood pump experiment at the University of Utah. The blood pump was implanted in a calf and driven by an external electric motor through a 36 inch long flexible drive shaft. The experiment was terminated 25 hours after implantation when the flexible shaft core broke. The core failure occurred near the end of the casing fitting connected to the blood pump. No permanent deformation of the casing near the core failure was observed. A failure analysis is being performed to determine the cause of the core failure.

TASK 1 - THERMAL INSULATION DEVELOPMENT

During the quarter, outgassing tests were performed and analyzed and outgassing procedures for insulation packages were developed. Work on spacer and cup technology development continued. The outgassing evaluation test apparatus was modified and design of the thermal evaluation test apparatus was initiated. Life test data from the 20-cup pack were analyzed and tests were initiated with the 40-cup pack using the guarded orientation test apparatus (GOTA). Technical progress on each of these efforts is reported below.

STATIC VACUUM TECHNOLOGY DEVELOPMENT

Outgas Identification

During this quarter, nickel foil materials were tested to determine their outgassing characteristics under various conditions of surface cleanliness. A list of the materials tested and their history is reported in Table 1-1. Prior to testing the samples, 25 mass scans were made on the empty system with the furnace tube at various temperatures to obtain background data and identify gaseous species characteristic of the empty system. Table 1-2 lists the major species monitored for the empty system at room temperature and at 1600°F (scans 1 and 23-24). The base pressure in the system at 1600°F after pumping for 2 hours was 8×10^{-7} torr. Data are given as relative intensity, i.e., ion current of a species divided by the largest measured ion current. After the samples of 0.0005 in. nickel foil were inserted into the crucible in the quartz furnace tube, the system was pumped down and baked out at 250°C for 24 hours. Mass scans were made at room temperature, ~ 500°F, ~ 800°F, ~ 1050°F, ~ 1200°F, and ~ 1570°F. In all, a total of 70 mass scans were made on these three samples. The resulting data are too voluminous to reproduce in this report, but it can be summarized by referring to Table 1-2.

Sample 1 - At temperatures up to 500°F, the normal outgassing products of H_2 (AMU 2)*, H_2O (AMU 18), CO and N_2 (AMU 28), and CO_2 (AMU 44) are measured with the relative

*AMU = Atomic Mass Units = mass of atom/electron charge

TABLE 1-1

MATERIALS EVALUATED FOR OUTGASSING CHARACTERISTICS
AND THEIR PROCESSING HISTORY

<u>Sample No.</u>	<u>Sample Description</u>	<u>Sample History</u>
1	Plasma Sprayed Nickel Foil Cup 2.032 grams, 172 cm ² Surface Area	Part of 0.100 in. dia. polka dot cup set. Run in vacuum at high temperature.
2	Nickel Foil Cup 0.292 grams, 44.5 cm ² Surface Area	As-received nickel foil with no pre-test cleaning.
3	Nickel Foil Cup 0.332 grams, 42 cm ² Surface Area	As-received nickel foil with pre-test cleaning by ultrasonic degreasing in M-6, followed by an ethyl alcohol rinse. Handled with tweezers.

TABLE 1-2

RELATIVE INTENSITIES OF SELECTED MASS SCANS

Atomic Mass Unit	Possible Ions																Total Pressure (torr)
	H ⁺ 1	H ₂ ⁺ 2	C ⁺ 12	N ₂ ⁺⁺ 14	CO ⁺⁺ 15	CH ₃ ⁺ 16	O ₂ ⁺⁺ 17	OH ⁺ 18	H ₂ O ⁺ 27	C ₂ H ₃ ⁺ 28	N ₂ CO ⁺ 32	O ₂ ⁺ 34	Ar ⁺ 40	CO ₂ ⁺ 44	SO ⁺ 48	SO ₂ ⁺ 64	
Room Temp. after bake- out (Scan 1)	7.69	100.0	1.10	2.09	-	3.58	8.28	31.14	0.95	16.63	0.15	-	1.54	1.07	1.1 × 10 ⁻²	3 × 10 ⁻³	6 × 10 ⁻⁸
Empty System 1600°F (Scan 23-24)	15.65	100.0	16.70	3.26	5.43	15.22	10.43	32.61	21.74	60.0	3.03	4.90	3.97	13.30	0.26	0.26	1 × 10 ^{-5*}
Sample No. 1 1600°F Scan 52	11.14	75.00	38.86	8.18	8.18	25.45	23.41	50.09	36.36	100.00	2.05	4.00	2.73	32.41	0.08	0.07	1.7 × 10 ^{-5**}
Sample No. 2 1600°F Scan 72	11.15	59.32	14.80	3.04	2.57	10.81	18.18	62.84	10.47	100.00	9.32	18.04	0.88	17.50	0.66	0.81	1 × 10 ^{-5***}
Sample No. 2 1200°F Scan 67	34.41	87.65	22.06	5.59	10.00	32.82	75.29	100.00	13.53	72.35	7.76	12.06	1.94	52.35	5.12	not measured	1.45 × 10 ^{-5***}
Sample No. 3 1600°F Scan 92	10.55	45.86	12.90	3.93	1.57	8.37	17.79	64.97	8.28	100.00	4.23	8.83	0.62	11.31	0.214	0.234	7 × 10 ^{-6*+}
Sample No. 3 900°F Scan 82	16.72	44.46	8.69	1.50	2.30	9.99	27.77	99.20	3.60	100.00	1.50	2.50	0.60	15.98	0.70	0.794	3.75 × 10 ^{-6*+}

* Gas load when temperature was initially reached - pumped at temperature to 8 × 10⁻⁷ after 2 hours.** Pumped down at temperature to 6 × 10⁻⁷ torr after 2 hours.*** Pumped to 4.2 × 10⁻⁶ in 1/2 hour.*+ Pumped to 4.3 × 10⁻⁶ in 1/2 hour.

intensity decreasing with increasing AMU similar to Scan No. 1, Table 1-2. At temperatures above 1200°F, CO-N₂ (AMU 28) and CO₂ (AMU 44) intensities increase with AMU 28 becoming the most abundant gas species (Scan 52, Table 1-2). After holding at temperature for 2-3 hours, these gases are pumped away and the pressure returned to the 10⁻⁷ torr range.

Samples 2 and 3 - Initially these samples behaved as did sample 1. However, at temperatures above 500°F, sample 2, which was not surface cleaned prior to testing, showed significantly high offgassing rates for H₂O (AMU 18), CO + N₂ (AMU 28), and CO₂ (AMU 44) as well as mass 34, 48, and 64 (scan 67). Very few inorganic compounds and standard organic cracking patterns show fragments falling on mass 34, H₂S, H₃P, and H₂O₂ are the inorganic compounds which can give a mass peak at 34. However, because peaks were also seen at 48 and 64, which are SO⁺ and SO₂⁺ ions, respectively. H₂S is the most likely species at AMU 34. The isotopic abundance of S³²/S³⁴ is 22.74. The isotopic abundance ratio of H₂S³²/H₂S³⁴ is also close to 22 indicating that a large portion of the mass 34 is due to the reaction of hydrogen gas with sulphur found in the surface of the nickel. The copper precipitate deposited on the nickel cups during etching from the mandrel is cleaned away by dipping in H₂SO₄. However, only sample 2 has excessively high H₂S⁺, SO⁺, and SO₂⁺ peaks. Sample 3, cleaned as described, had about 1/2 the H₂S relative intensity as shown in Table 1-2. The empty systems showed ~ 4% relative intensity of H₂S⁺, and sample 1, exposed to vacuum as a part of an insulating test package, was ~ 4%. The conclusion to be reached is that the sulphur contamination is primarily a surface effect and can be eliminated by first cleaning the cups as described and then subsequently heating the cups in a vacuum system at temperature between 500 - 800°F. Other than the sulphur containing species, no significant outgassing products, which would be detrimental to the static vacuum system integrity, have been discovered.

Other materials to be tested include U-700 (both the wrought alloy and powdered alloy), the U-700/braze/Ni-200 joint, the outer can/U-700 joint (to be defined), and the various feedthrough attachment schemes currently being contemplated.

Outgassing Procedure Development

In the interest of obtaining optimum static vacuum performance, WANL will use as a guide material selections, design, fabrication, and cleaning procedures developed for constructing ultra-high vacuum systems^(1, 2). It is recommended that all welded feedthrough systems be utilized. Soldered feedthroughs and the problems associated with the flux removal from crevices, etc. will compromise static vacuum performance.

The outgassing data being reported thus far are preliminary in nature. For instance, the time and temperature conditions reported on Table 1-2 should not be construed as proposed recommendations for outgassing procedures. When quantitative outgassing data as a function of time and temperature are measured for the particular materials in the vacuum foil insulation system, then valid extrapolations for predicting static vacuum performance will be possible and outgassing procedures will be specifically outlined.

The literature^(1, 2, 3) contains many references to accepted cleaning and outgassing procedures suitable for ultra high practice. Any standard practices which are selected will, of course, be verified prior to incorporation into the processing sequence; however, thermal converter considerations such as temperature limitations, surface effects, joint, and material heat treating schedules may require development of alternate cleaning and outgassing procedures.

The handling and cleaning principals listed in Table 1-3 are generally recommended for all components which communicate with the vacuum volume.

-
1. Strausser, Yale, "Review of Outgassing Results", Varian Associates, Palo Alto, California, VR-51.
 2. Keeton, A. R. and Stemann, L. G., "Fabrication of Fuel Pin Assemblies - Phase III - Appendix E, WANL-M-FR-72-004, NASA CR-120956, December 1972.
 3. Singleton, J. H., "Gas Sources and Pumping Processes in Electronic Power Tubes", Sci. Paper 71-1C8-FIVAC-P-1, March 11, 1971, Westinghouse Research Lab., Pittsburgh.

TABLE 1-3

GENERAL RECOMMENDED HANDLING AND CLEANING
PRINCIPALS FOR COMPONENTS WHICH COMMUNICATE
WITH THE THERMAL INSULATION VACUUM SPACE*

1. All surfaces must be degreased by ultrasonic agitation in M-6 or trichlorethylene.
2. This solvent must be removed by a detergent or suitable ethyl alcohol rinse.
3. All surfaces should be as smooth as possible.
4. All surfaces should be subjected to bakeout at temperatures no lower than 150°C. (Higher temperatures preferable.)
5. Once baked out, samples should be stored in Ar or N₂ filled polyethylene bags.
6. Handling should be done only with tools or instruments similarly cleaned or with gloved hands. Gloves should be clean and fresh.

*These principals subject to modification as particular materials and specific outgassing information is generated.

SPACER TECHNOLOGY DEVELOPMENT

The possibility of plating alloy cups or composite cups is being investigated. An electroplated nickel cup with a thin chromium layer over-plating has been made. From this point, several alternatives are possible including; 1) alloying the cup by interdiffusing the Ni-Cr layer, and 2) oxidizing the chromium layer to form a thin Cr_2O_3 coating which, if it did not compromise the emissivity, could be utilized as a spacer between adjacent cups. Inquiries are being made to determine if data describing the effect of a Cr_2O_3 layer have been published. The infra-red scattering investigation and improved plasma spraying patterns and techniques are scheduled to begin during January.

CUP TECHNOLOGY DEVELOPMENT

A set of 5 Ni-B electroless deposited foil cups were ordered from DuPont. Electroless deposition is an attractive process for forming cups. One feature is the possibility of utilizing reusable mandrels. Also, the Ni-B alloy is a wear resistant alloy which is harder than electro-deposited nickel. This property could indicate the possibility of providing a stiffer cup with a higher modulus resulting in less tendency to sag and sinter together. The first step in the evaluation process has been to attempt to etch the cups from the mandrels at WANL. Two attempts at etching proved unsuccessful. The Ni-B cups cracked and broke apart. A Ni-B coated mandrel has been sent to the R&D Center for etching using their etching procedure employed for the electroplated nickel cups. Results are expected during January. Two more mandrels will be delivered with alternate heat treatments to be described based on the results of the etching. If it is possible to achieve a free standing cup, the cup material will be subjected to heat treatments, hardness measurements, and sag tests to determine its merits relative to electroplated Ni.

Several nickel cups have been plated to various thickness (of 0.0005, 0.0010, and 0.0015 inches) at the R&D Center. These cups or rings cut from the cups will be tested for creep or sag behavior as a function of thickness. In order to best utilize a spacer, the foil cup material

should not sag between high areas of the spacer particles and should resist creeping to conform to the surface contours of the heat source or adjacent inner cups. Resistance to sag should decrease foil to foil conduction loss. For this reason, utilization of thicker or more sag resistant foil layers in the hottest areas of the insulation may be advantageous.

TEST APPARATUS

The OETA (Outgassing Evaluation Test Apparatus) has been modified. As previously configured, the apparatus was suited to a qualitative analysis of outgassing from sample materials. To provide precise, time dependent data for outgassing rate measurements. the system was modified, and this modification is illustrated in Figure 1-1. One 15 l/sec ion pump was removed, and a 150 l/sec ion pump was added in place of the turbomolecular pump. Quantitative measurements of outgassing rates will be done as follows.

Referring to Figure 1-1, valves V4, V6, V5, and V2 will be closed. Valves V1 and V3 will be open. The sample will be heated, and the system will be pumped through a 5/16 inch diameter orifice (C₁). The expression which relates the outgassing rate with the pumping speed and system pressures and volume is:

$$V \frac{dP}{dt} = Q - SP \quad (\text{Eq. 1-1})$$

where:

V = system volume - liters

P = system pressure - torr

t = time - sec.

Q = outgas rate of all surfaces - torr-liters/sec.

S = pumping speed of all pumps - liters/sec.

Figure 1-2. Schematic of the Gas Calibration Apparatus

In the case where $V \, dP/dt = 0$ as happens when an equilibrium steady state condition is achieved, a trivial solution

$$Q = SP \text{ (torr-liters/sec.)} \quad (\text{Eq. 1-2})$$

describes the system.

In the system shown in Figure 1-1, which is flow rate limited by a conductance C_1 (liters/sec.), the pumping speed of the system S' is given by

$$S' = C_1 P_1 \text{ (torr-liters/sec.)} \quad (\text{Eq. 1-3})$$

if $P_1 \gg P_0$, where P_0 = pressure in the ion pump. By making the conductance C_1 small, the condition $P_1 \gg P_0$ is realized. Various ranges of outgassing rates can be measured by varying the size of C_1 .

To eliminate errors due to gas reactions and gaseous products produced at the mass spectrometer filament, a partial pressure of each gas species will be measured before and after closing valve V3. The pressure differences will then be only the gases coming from the heated sample.

This system will be calibrated and cracking patterns determined for each of the major gas species which are being studied during the outgassing investigation. This will be accomplished by leaking a pure gas at a known pressure through a calibrated leak into the sample chamber.

If P_2 is the pressure in the reservoir, Figure 1-2, then by solving the differential equation

$$V_1 \frac{dP_1}{dt} = Q_2 - Q_1 \quad (\text{Eq. 1-4})$$

where Q_1 is the quantity of gas flowing through C_1 with the pressure gradient $P_1 - P_0$ and Q_2

is the quantity of gas flowing through C_2 with the pressure gradient $P_2 - P_1$, one obtains a solution at equilibrium (steady state) when $\frac{C_1 + C_2}{V} \ll t$ of

$$P_1 = \frac{C_2 P_2 + C_1 P_0}{C_1 + C_2} \quad (\text{torr}) \quad (\text{Eq. 1-5})$$

Therefore, by knowing the pressure in the mass spectrometer chamber P_1 from equation (5) and the ion current measured by the mass spectrometer at that mass, the sensitivity of the mass spectrometer and the cracking pattern of a particular gas species and fragmented ion will be determined. By knowing this sensitivity in amps (ion current)/torr (pressure of the gas), the exact pressure of the gaseous species in the system can be directly calculated. By multiplying the known pressure by the conductance C_2 , the outgassing rate of a particular gas can be measured in torr liter/sec - area of sample as a function of temperature and as a function of time at temperature.

THERMAL EVALUATION TEST APPARATUS

The design of the thermal evaluation test apparatus was started after the IVBM hot finger dimensions had been received from Philips. This apparatus, to be completed by the end of February, will be used to evaluate all aspects of the insulation test package and to test both IVBM thermal insulation test packages prior to their being assembled on the Phillips hot finger for both the electrical and isotope powered systems. The thermal evaluation apparatus will be a portable multi-compartment vacuum chamber capable of being attached to the precision gas analyzer (PGA) for outgassing evaluation, placed in a dynamic vacuum system for long term thermal testing or being sealed with metal gaskets for long time static vacuum evaluation. An electrical heater identical to that used in the IVBM will be inserted inside the dummy hot finger. Heat loss down the cylinder wall will be measured with a water calorimeter.

TEST INSULATION PACKAGE

Detailed drawings for the test insulation system are currently being made. The drawings will be signed off during the first week in January. Based on a set of sketches and some preliminary specifications, electroplating and other manufacturing sources of foil cups were sought. A bid package was sent to six possible suppliers. Servometer Corporation of Clifton, New Jersey responded with a price and delivery schedule which would satisfy the program requirements. Servometer Corporation supplied the first set of Ni cups evaluated during the high performance thermal insulation test program. Orders for a set of cups will be placed in early January.

CUP PACKAGE INVESTIGATIONS

20-Cup Pack Test (Post-Life Test Evaluation)

After four months of life testing, the 20-cup pack test was disassembled. Thermal data for this test was presented in a previous quarterly progress report*. Some thermal degradation was observed during the four months exposure. The cause of the degradation was discovered during disassembly. Several points on the heater surface have sintered to the first cup. The points are shown in Figure 1-3. The first and second cups were also sintered together at these points. In foils 3 through 6, there was some resistance to separating cups. A slight pressure accomplished the separation.

To summarize the data for the 20 cup test, the heat loss for the 56 cm^2 test system was 2.4 watts and after the total exposure, the heat loss through the insulation was ~ 2.8 watts, an increase in heat loss of 0.4 watts or about 15% thermal performance degradation. The life testing on the 20-cup pack was done under a dynamic vacuum. The heat loss was determined after the life test by measuring the power required to achieve a thermal equilibrium between the guard heater and the working heater. This permitted separation of the effects of guard heater degradation and thermal insulation performance.

*Astronuclear Laboratory, Westinghouse Electric Corp., "Nuclear-Powered Artificial Heart Prototype System Development Program", Phase III Quarterly Progress Report, July 1 through September 30, 1973, WANL-3043-12, USAEC Contract AT(11-1)-3043.

Figure 1-3. Guard Heater Hot Finger After Removal of the 20 Cup Pack
Showing the Points Where Sintering Occurred

The heat loss resulting from this sintering between foil layers is equivalent to a loss of 3 to 4 foils or ~ 0.9 watts based on an 84 cm^2 hot finger. An equivalent amount of sintering in a 40 or 60 layer package, i. e., ≈ 4 foils being lost would amount to only ~ 0.25 watts for a 40 layer package and ≈ 0.17 watts for a 60 layer package. As the number of foil layers increase, the effect of sintering 3 or 4 layers becomes less significant. However, to prevent sintering, a new technique for applying ZrO_2 spacers was developed and applied to the 40 cup pack test. For the 20 cup pack test the nickel cups were coated with ZrO_2 on only the outside by a brushing technique. The coverage was only ~ 3.2 mg of ZrO_2 per cup. The technique used for the 40 cup pack gave an average coverage of 18 mg per cup and deposited ZrO_2 particles on both the inside and outside of the cups.

40-Cup Pack Assembly

Individual cups for the 40 cup pack were first weighed and the thickness determined by a weight-area-density relationship. The majority of the cups were from 0.0006 to 0.0007 inches thick, with several cups being over 0.0008 inches thick and only one cup under 0.0005 inches thick. The total cup weight for this 40 cup package was 63.566 grams.

An alcohol- ZrO_2 powder (~ 325 mesh) slurry was made by mixing 24 grams of ZrO_2 per 1000 ml of ethyl alcohol. This slurry was then mixed and placed into a glass tray. Periodically, the concentration of the slurry was checked by sampling the slurry and weighing the slurry before and after the alcohol has evaporated. Concentrations remained with ranges between 12-18 mg/ml. The difference between the as-mixed concentrations and that measured is due to the instant settling of the larger ZrO_2 particles. Figure 1-4 shows a cup dipped into a slurry filled tray. The cup was rolled in slurry and allowed to drip for 15 to 30 seconds before drying in a hot air stream. The cup was handled only at the closed end. After drying, the cup was carefully picked up by the open end skirt and the top dipped and dried twice to cover the area used to handle the cups after the initial dipping. The 40 cups, after coating, are shown in Figure 1-5. The cups were assembled by inserting the cups into the outer cup one at a time and weighing the package after each additional cup was added. The weight of the powdered spacer was then calculated by subtraction. Figure 1-6 shows the 40 cup pack fully assembled.

Figure I-4. Dipping Technique Used to Coat Nickel Foil Cups for 40 Cup Pack

Cup Nos.

1 - 6

14 - 20

27 - 35

7 - 13

21 - 26

36 - 40

Figure I-5. As-coated Nickel Foil Cups Prior to Assembly of the 40 Cup Pack Test

Figure I-6. Bottom View of 40 Cup Pack Test After Assembly

The average weight of ZrO_2 on a cup was ~ 18 mg. Air films generated by air displacement when a cup was inserted, closed end down, were utilized to guide the cups into place. This minimized the probability of scratching or removing the spacer. Cups 1, 39 and 40 are shown in Figure 1-7 after assembly and disassembly. These photographs indicate no wiping away of the ZrO_2 spacer material. Cup 40 is uncoated on the exterior. The cups were assembled onto the guarded orientation test apparatus (GOTA) in the vertical position. The system was pumped down in late December. At the end of this reporting period an equilibrium data point had been reached, and the position of that point (0.7 watts heat loss through the 40 layer package at 780.7°C) indicates that the performance will be as predicted by the extrapolation of the experimental data previously acquired.

5-Cup 0.020 in. Diameter Plasma Sprayed Test

The 5-cup 0.020 inch ZrO_2 plasma sprayed cups were assembled onto the guarded heater, and the temperature was raised to 1600°F . After 10 days, degradation was noted in the thermal performance of the 5-cup package. After 24 days, the test was terminated after aging had caused a 12-14% heat loss. Figure 1-8 shows the 5 cups after disassembly. Cup No. 1, which had sintered to the electrical heater assembly at one location was crushed as it was lifted off against the force of the sintered bond. Cup No. 2 was wrinkled during removal. It is apparent from the wrinkling which occurred that sintering occurred only at several small points similar to that shown in Figure 1-3. The percent of area coverage of the 0.020 inch ZrO_2 dots was only about 2.5%. These tests indicate that a larger area should be covered by the ZrO_2 . Plasma sprayed ZrO_2 is very adherent and still appears to be the most promising technique for applying standoff material.

No. 1

No. 40

No. 39

Figure 1-7. Several Cups after Disassembly of a Trial Assembly Showing that ZrO_2 was not Scratched or Removed during Assembly. (Cup 40 was not coated externally.)

Figure 1-8. .020 in. ZrO_2 Flame Sprayed Cups After Disassembly.
Cup 1 is deformed due to forcing the cup up to disengage it
from the sintered spot. Cup 2 is wrinkled from handling.

Guarded Orientation Test Apparatus (GOTA)

The GOTA is pictured in Figure 1-9 with a Min-K-2002 standardization and calibration sample in the horizontal position. The GOTA guarded heater design is basically the same as has been previously described. Chromel-Alumel bare wire thermocouples, a 1/8 inch layer of Min-K-2002 between the guard and measuring heater, and a slit in the stainless steel hot finger at the equilibrium heat flow plane are the only modifications made in this system. No difference in the heat loss for the Min-K-2002 test piece when tested either horizontally or vertically was found. This test system has been evaluated and checked out. Currently, the 40 cup pack test is being evaluated on the GOTA in the vertical position.

Figure 1-9. Min-K-2002 Being Tested in Horizontal Position on Guarded Orientation Test Apparatus

TASK 2 - THERMAL CONVERTER DEVELOPMENT

Efforts continued on five of the components originally chosen for further development: the hot finger, the stationary regenerator, the control system, the bellows seals, and the oil pump. The status and the detail of efforts on each of these components is discussed.

HOT FINGER MATERIALS INVESTIGATION

The aim of this subtask is to arrive at a hot finger metallurgical structure which will meet the long term program goals. The discussions which follow summarize recent efforts in this direction.

A metallographic examination of the thin wall section of the hot finger revealed very large grains. After operating in the Stirling engine at high temperature, offsetting of the grains occurred as shown in Figure 2-1 due to grain boundary sliding. Excessive sliding can lead to the separation of grains, and in turn, leakage, which is intolerable for the performance of the Stirling engine.

Literature search and consultation with the supplier of Udimet-700, Special Metals Corp., led to a heat treatment schedule designed to maintain small grains in the heat treated material. The heat treatment schedule is as follows:

(1) 1095°C for 8 hours

Overaging of the γ' phase, which leads to a pinning down of the grain boundaries.

(2) $1130^{\circ}\text{C} \pm 7^{\circ}\text{C}$ for 4 hours Rapid Cool

Only a partial solutioning of γ' is accomplished. However, this is sufficient for high strength properties.

Figure 2-1. Ruptured 0.004 Inch Hot Finger Wall (70 X)

Figure 2-2. Udimet Bar No. 1 As Received (100 X)

- (3) 870°C for 8 hours
- (4) 983°C for 4 hours
- (5) 650°C for 24 hours
- (6) 760°C for 8 hours

Two different bars of cast and swaged Udimet-700 from Special Metals have been investigated. The two bars show different grain structures in the as received state. (Figure 2-2 and Figure 2-3). The No. 1 bar exhibits a heavily deformed structure whereas the No. 2 bar shows a re-crystallized structure.

After being subjected to the steps (1) and (2) in the heat treatment, the No. 1 bar displayed a somewhat smaller grain size than the one observed with the previously used standard heat treatment. (Figure 2-4). However, it is felt that the grain size is still not small enough. The No. 2 bar showed the same large grain size as obtained after the previous standard heat treatment. (Figure 2-5).

To overcome these problems, it was decided to investigate the possible use of powder metallurgically (PM) prepared bars of Udimet-700 from Federal Mogul Corp. These bars are much more uniform in their composition than the cast bars and are also fully dense. The PM bar exhibits a grain size (in the as received condition) that is much smaller than the cast bars. (Figure 2-6). Even after subjecting the PM bar to the first two steps of the above described heat treatment, the grain size remains small. (Figure 2-7). It is felt that the structure of the PM bar, after being subject to the above described heat treatment, has a very suitable grain structure and will exhibit a high creep resistance. An alternate heat treatment which maintains acceptable grain sizes while developing acceptable bulk properties with the swaged material has been identified. This treatment is expected to be used for the IVBM.

STATIONARY REGENERATOR

Analysis and design work on the stationary regenerator configuration was completed. Its main features, and the reasons for its incorporation into the IVBM units are discussed below.

Figure 2-3. Udimet Bar No. 2 As Received (100 X)

Figure 2-4. Bar No. 1 After Heat Treat (100 X)

Figure 2-5. Bar No. 2 After Heat Treat (100 X)

Figure 2-6. Powdered Udimet as Received (100 X)

Figure 2-7. Powdered Udimet as Received (100 X)

The regenerator of the Phase II bench model design is an integral part of the displacer, i. e., it reciprocates in the internal cavity of the hot finger. Its configuration is depicted in the cross-sectional sketch of Figure 2-8. The performance tests conducted with the engine at the conclusion of Phase II of the program indicated a design deficiency: There was rubbing contact between the top of the displacer and the adjoining surface of the heater, indicated by A and B respectively (see Figure 2-8). Also, for inertial balance, the reciprocating regenerator matrix had to be compensated by counterweight masses, in addition to the weights normally required.

For adequate heat transfer from the source (i. e., from the radioisotope capsule) to the working medium, the annular gap width between the inner heater wall and the upper circumference was set at 0.006 inch. To avoid the need for guiding the displacer in the hot region in order to maintain the required gap geometry, provisions were made to insure straight displacer motion with the aid of mechanical constraints incorporated into the engine drive. It should be noted that guides in the hot region would have resulted in significant frictional losses. During the engine performance tests, however, it became evident that rubbing occurred between the displacer and the adjoining wall. Although the rubbing was significantly reduced by improving the precision of the mechanical constraints, there is evidence that it was not completely eliminated.

The rubbing contact, however slight, is undesirable for three major reasons: it is associated with a power loss, it gives rise to a life problem, and it possibly results in a heat transfer loss because of poor annular gap geometry. It should be noted that it is very difficult to determine the magnitude of the overall loss caused by this condition.

The total weight of the displacer assembly, i. e., shell, regenerator matrix, and upper cap is 137 grams. Of the latter, 74 grams can be ascribed to the regenerator matrix and that part of the displacer which is required to retain the mesh in place. The counterweight required to balance the moving mass is 40 grams.

Figure 2-8. Phase II Moving Regenerator Design

As a means of eliminating the rubbing contact problem, an analysis was made of a stationary regenerator design. In a configuration of this nature, the gap between the displacer and the hot finger is no longer critical. The geometry arrived at is shown in Figure 2-9 (for an electrically heated engine). As indicated, a representative annular gap width is 0.016 inches.

It should be noted that in the stationary regenerator design, as opposed to the design shown in Figure 2-8, an additional wall connects the hot to the cold region. Although this improves the rigidity of the structure, it also presents an additional heat leak path. Specifically, the total conduction losses of the stationary regenerator design total 4.8 watts as opposed to 4.0 watts conduction losses incurred in the previous configuration. The other thermodynamic characteristics of the two designs, i.e., flow losses and regenerator effectiveness, are the same. An energy balance for the thermal converter using the moving regenerator is shown in Figure 2-10. This balance is for an ideal heater geometry, and does not include the loss of heater performance due to the heater gap not being concentric, nor does it include an estimate of possible frictional losses due to rubbing. It can be compared with Figure 8-4 to assess the theoretical differences between the two regenerators.

There are several other features of the two designs which should be noted:

- The stationary regenerator configuration is more complex, and therefore more expensive to fabricate.
- Because of the larger overall diameter of the stationary design, the ultimate capsule can be made shorter, saving approximately 0.7 inches in the overall length of the engine.
- The counterweight size can be reduced in the stationary configuration, with an attendant weight saving of 40 grams.

The details of the stationary regenerator configuration will be discussed in conjunction with the IVBM design.

Figure 2-9. Stationary Regenerator Configuration

Figure 2-10. Energy Balance with Moving Regenerator

CONTROL SYSTEM

To improve and possibly simplify the design of the control system, two new regulating schemes were investigated. The control system designed and built during Phase II was subjected to a preliminary test to determine its capability of controlling the power output with constant engine heat input. Additional tests have been performed on the control system with the control valve operated manually. Power reductions up to 2 watts were obtained with the engine operating at constant speed. The only definite conclusion which can be made concerning the effect of the control valve on heat input is that it is small. The experimental results indicated that the hot side temperature went down by about 10°F . This indicates a slight increase in heat input to the engine for constant temperature operation rather than the decrease which was expected. This change in temperature approaches the resolution limit of the temperature measuring equipment, and when other effects, such as the accuracy of speed measurement as well as random variations in heater power and coolant temperature are taken into account a 10°F temperature change is smaller than the expected accuracy of the experiment. The new systems which will be described, however, especially the second one, appear to be more attractive. They satisfy the same requirement while their operational characteristics offer simplicity.

The control system schematically depicted in Figure 2-11 makes use of flow losses (oil flow is supplied by a pump) in a valve to control the output of the engine. By increasing the system losses, the control will reduce output power and engine speed. When the engine is operating at design speed, the output pressure of pump I (4) is not sufficiently high to compress spring (2). If, however, a speed increase occurs, the output pressure from pump I will increase, pushing piston from position I to position II. In position I, output from pump II (5) is able to flow freely (see arrow) to the sump (6), but when a speed increase moves the piston, the flow from pump II becomes more restricted in the small clearance space (1) of the piston. By restricting the flow, losses due to viscous friction go up and the result is both pumps slow down and with them the engine speed falls to its design value.

Figure 2-11. Control Schematic - Flow Losses

The other control system shown in Figure 2-12, uses the flywheel motion to dissipate excess energy. When the shaft speed increases, it activates control (5) which opens valve I (6). Oil from the pump (7) flows into groove (2) and between the flywheel (4) and the housing wall, producing viscous losses. As viscous losses increase, the engine speed drops. When the speed has dropped to its control point, the control shuts valve (6), diverting the excess flow back into the bearing lines.

As the speed drops, valve II (8) changes pump input from sump (9) to input from wicking (1 and 3). This drains the oil from the flywheel housing and reduces viscous losses to their normal level. Then the valve (8) returns the pump input to the sump so the wicking lines remain primed with oil. Valve II has a dual function in that it would also permit removal of overflow oil from the flywheel housing if the engine is oriented such that oil leaks in along the shaft.

Plans are being made to prototype the second, or possibly both systems, and thus be able to determine their operational characteristics experimentally.

BELLOWS SEALS

The apparatus designed to test the performance and the life of the bellows seals was fabricated. Its design was shown in the previous quarterly report; the fabricated unit (disassembled) is shown in Figure 2-13.

As indicated in previous reports, the apparatus was to be driven by an electric motor via a magnetic coupling. The use of such a coupling was to provide design data and operating experience with a configuration which is being considered as a potential engine/pump interface in future models. However, during discussions with the Indiana General Corp., the manufacturer of magnetic materials for couplings, it was found that when driving a pulsating load, such as that of the engine, magnetic couplings may be subject to "hunting". In order to pro-

- 1 FLYWHEEL WICKING
- 2 OIL INLET GROOVE
- 3 SHAFT WICKING
- 4 FLYWHEEL
- 5 CONTROL
- 6 DISSIPATION CONTROL VALVE
- 7 PUMP
- 8 FLOW CONTROL VALVE
- 9 SUMP

Figure 2-12. Control Schematic - Flywheel Use

Figure 2-13. Bellows Test Apparatus

ceed with the bellows tests while investigating the hunting problem in more detail, it was decided to drive the apparatus directly by an electric motor, i. e., have the latter integral with the crankcase (buried in it).

Parts are presently being fabricated to allow for the new coupling method. Provisions are being left in the apparatus for possible interface with a magnetic coupling at a later date

OIL PUMP

The final design of the pump gears was completed. The tolerances on several dimensions are very high, in order to attain the desired pump efficiency. The gears will be manufactured by the Moore Precision Co., of Bridgeport, Conn. A purchase order for this work was issued in late December. It was decided not to proceed with the planned life test of the bench model oil pump. The pump detail design is sufficiently different from the bench model pump. It was felt that this test would not be representative because of the differences between the Phase II and Phase III designs.

OTHER COMPONENTS

In addition to the five components discussed above, plans were originally made to investigate three others: the connection between the engine and the buffer space, the drive bearings, and a "gap" type heat exchanger. As reported previously, work on the first item, i. e., buffer space connection, was completed. The power increase ascribable to increasing the buffer volume connector to 0.46 cm from 0.23 cm inside diameter, has been estimated at 0.5 watts. This was determined by comparing the performance of the Phase II bench model engine which has the small connector with the Philips version of this engine which has the larger connector. The first phase of the drive bearing investigation was also completed. The bearings are being integrated into the IVBM design, as well as into the bellows test apparatus. The third item, the "gap" heat exchanger is no longer needed, since the stationary regenerator design is being integrated into the IVBM unit.

TASK 3 - BLOOD PUMP DRIVE DEVELOPMENT

During the quarter, analyses of the planetary gear test and the liquid filled drive test from the previous reporting period were completed. Similar tests using the Evoloid gear reducer were performed and are being analyzed. Initial results from the soaking of silicone rubber in oil were obtained; these tests are continuing.

SPEED REDUCER INVESTIGATIONS

Data analysis from the planetary gear train test performed in the previous quarter was completed. The purpose of this test was to determine the power loss of the pump drive speed reducer over the range of input powers required during the bench model pumping cycle. The tests were performed at the bench model design point input speed, 900 rpm, and at 720 and 1080 rpm. The results obtained are presented in the graph shown in Figure 3-1. The efficiency of the speed reducer at the bench model average input power 6.0 watts, calculated from the measured power loss at this input power 0.45 ± 0.14 watts, was about 92.5 percent. The differences in the power losses measured for the three input speeds were not sufficiently well defined to permit conclusions to be drawn regarding the variation of power loss with speed.

A test similar to the one described above was performed with an Evoloid gear speed reducer that is interchangeable with the bench model planetary gear speed reducer. The Evoloid gear reducer uses gears with a special tooth form that permits a high speed reduction with high efficiency in a single step. This feature allows the envelope of the speed reducer to be somewhat smaller than it could be with a planetary gear arrangement. A careful review of the data obtained is in progress.

LIQUID FILLED DRIVE EVALUATIONS

The data obtained in the blood pump drive mechanism test performed in the previous quarter

Figure 3-1. Power Loss vs. Input Power

were analyzed. This test was performed to determine the power loss introduced by filling the bench model drive mechanism with sperm oil. The test results are presented in the graph shown in Figure 3-2. The lines through the data points in the graph show the no-load power required by the drive for flexible shaft speeds from 100 to 1000 rpm with three oil fill conditions of the drive. These conditions are no oil in the drive, oil in the mechanism and speed reducer, and oil in the entire drive. The power required values measured at the flexible shaft bench model design speed, 900 rpm, are as follows:

<u>Condition</u>	<u>Power Required (watts)</u>
No oil in drive	0.25 $\pm$ 0.03
Oil in mechanism and speed reducer	0.35 $\pm$ 0.03
Oil in entire drive	0.45 $\pm$ 0.03

Since filling the pump drive with oil increased the required drive power only 0.20 ± 0.04 watts, about 3.3 percent of the average total input power specified for the bench model pumping cycle, the efficiency goals established for the IVBM and subsequent model blood pump drives are reasonable for drives filled with a light oil. The difference between the power requirements of the two conditions where oil was used indicates that the oil in the flexible shaft accounted for about half of the total power increase that resulted from filling the drive with oil.

A test similar to the one described above was performed with an Evoloid gear speed reducer attached to the blood pump. Analysis of the data obtained is in progress.

DRIVE SHAFT SEAL STUDY

A survey was made to determine whether commercially available low leakage, low friction shaft seals could satisfy the requirements for the seal between the flexible shaft and the thermal converter. Because no suitable seals could be found, it was decided to use the bench model thermal converter type shaft seal in the IVBM.

Figure 3-2. Effect of Oil in Pump Drive on No-Load Input Power

SILICONE RUBBER - OIL COMPATIBILITY TEST

Soaking tests of silicone rubber in sperm oil and in Vitrea 21 oil (engine oil) were started at the Westinghouse Research Laboratories to confirm the results obtained in a simple experiment performed at WANL to determine whether the oil that will be used in the IVBM drive will react with the pump ventricles. In the WANL test, no significant deterioration of the silicone rubber was observed after soaking in sperm oil at 117°F for a period of 1.5 months.

TASK 4 - AUXILIARY COMPONENTS DEVELOPMENT

Because the primary path for heat rejection is to the blood through the walls of the blood pump pumping roll socks, a thermal flow path must be provided between the engine cooler in the thermal converter and the blood pump. For the Intermediate System, this function is expected to be performed by a flexible heat pipe, using water as a working fluid. Due to the much shorter time available for development of components for the IVBM, it was decided to use a pumped loop system for the IVBM. The two prime circulating pump candidates for this system are a bellows type pump located inside the blood pump mechanism and a gear pump.

COOLANT PUMP AND COOLANT LOOPS

The bellows^{*} pump can be expected to be fairly efficient. The major loss mechanism will be the pressure drops through the check valves required for proper operation. Although it is expected that an efficient gear pump could be made for the IVBM application, the efficiency is a complicated function of the geometry of the pump body and gearing. In addition, geometry of a gear pump is such that no convenient location for the pump is available inside the thermal converter or the blood pump mechanism. Therefore, a gear pump must be located external to these components, increasing the total volume of the system. In addition, a bellows, suitable for use as a pump, is a necessary part of the blood pump mechanism to maintain the proper location of the blood pump components. For these reasons, the bellows type positive displacement pump was selected for use in the IVBM.

With the coolant pump located inside the blood pump mechanism, severe restrictions are placed on the size of the lines leading to and from the pump.** As the detailed design of the

*Formerly referenced as Piston Pump No. 5A, Table 4-1, Nuclear-Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report, July-September, 1973, USAEC Contract AT(11-1)-3043.

**Physical integration of the IVBM coolant pump into the blood pump mechanism is covered in more detail under Task 7.

pump proceeded, it was found that the power losses in this area would be larger than anticipated. To reduce these losses, it was decided to use water instead of lubricating oil as the coolant. The lower viscosity of water is expected to reduce the pumping power required by about 1/2. As the blood pump body must be filled with oil, this introduces an additional thermal interface to the system, raising the coolant temperature somewhat. The area available through the bellows walls amounts to approximately 300 cm^2 (0.3 Ft^2), and the mean overall heat transfer coefficient is approximately $0.03 \text{ w/cm}^2\text{-K}$ ($60 \text{ Btu/Hr-Ft}^2\text{-}^\circ\text{F}$), and the temperature increase caused by this interface is 3K (6°F). The difference in operation of the thermal converter caused by raising the sink temperature 3K will be minute. The impact on the insulation surrounding the converter crankcase and the coolant lines will also be small.

HEAT PIPES

For the Intermediate System, the heat transport function is expected to be performed by means of a flexible heat pipe. Flexible heat pipes with dimensions and heat carrying capacity approaching the requirements for this application are available as off the shelf items. Two of these have been obtained. A radiographic examination of these pipes shows that they are of the annular wick type, with a bellows forming the outer tube. A woven metal sheath is provided to prevent the bellows from squirming. Performance tests will be carried out to provide a data base for extrapolation to the requirements for the Intermediate System.

TASK 5 - THERMAL INSULATION DESIGN

A vendor for the nickel foil cups has been selected. Delivery has been promised eight weeks after receipt of order. This meets schedular demands. Layout and detailed design work for the IVBM electrically heated system was started.

TASK 6 - THERMAL CONVERTER DESIGN

The conceptual and the preliminary designs of the IVBM thermal converter have been completed. Detailing was initiated on the hot finger/displacer/heat exchanger assembly, as well as on the drive, enclosure, and several of the accessories. The detailing of the first item was completed, and will be discussed below; the other three items are still in the detailing stage.

HOT FINGER

Two hot finger configurations are being developed: an electrically heated one, and one to be interfaced with the radioisotope capsule. Work on the latter has not been started.

The electrically heated hot finger configuration can be seen in Figure 6-1. As indicated, the stationary regenerator is an integral part of the finger assembly. The latter consists of five major elements: the heater (electrical) housing, a cylindrical structure about 1.5 in. O.D. x 0.9 in. I.D. x 2.0 in. long, made of nickel 200; the stationary regenerator matrix, made of stainless steel wire mesh; the working gas heater, consisting of 33 radial slits 0.010 in. wide x 0.157 in. deep x 0.280 in. long; two thin walls, 0.004 in. thick, made of Udimet-700; and the flange, made of Udimet-700, which will incorporate the feed-throughs, and which will couple the hot finger structure to the heat exchanger.

DISPLACER

The displacer assembly is depicted in Figure 6-2. Instead of the regenerator matrix, it now contains an insulating material, Min-K. Its wall is 0.004 in. thick, TIG welded at its lower extremity to a structure which serves several purposes: it couples the displacer to its drive rod, it contains the ring seal, and it provides means of pressurizing the internal cavity of the displacer. By pressurizing the displacer to the average working pressure, the walls can be made thin without danger of deforming the structure under the pressure fluctuations of the working space. The displacer will be made of Udimet-700.

Figure 6-2. IVBM Displacer

HEAT EXCHANGER

The heat exchanger, depicted in Figure 6-3, was designed for a water mean flow of $0.25 \text{ in}^3/\text{sec}$, at an inlet temperature of 120°F and with a pressure drop of 0.1 psi or less. To attain these conditions, the gas side of the exchanger will have 50 slits $0.008 \text{ in. thick} \times 0.157 \text{ in. deep} \times 0.520 \text{ in. long}$; its water side will have 6 slots 0.040 in. wide and 0.250 in. deep .

The heat exchanger structure will consist of a copper inner element, i.e., the part separating the working gas from the coolant, and a stainless steel envelope.

Figure 6-3. Heat Exchanger on IVBM Hot Finger

TASK 7 - BLOOD PUMP DRIVE MECHANISM DESIGN

During the reporting period, design work on the IVBM blood pump drive mechanism was initiated. The design status of the mechanism per se is discussed under this task along with design considerations for the bellows coolant pump that is planned for the IVBM.

IVBM DRIVE MECHANISM

A layout drawing defining the design of the pump housing and reciprocating mechanism has been completed. The mechanism is the same as the one used in the Phase II bench model, but many of the IVBM mechanism components as well as the pump housing will be made of low density metals such as titanium, aluminum or magnesium instead of stainless steel to minimize the weight of the pump. The basic external envelope of the pump body and the pumping sacks are the same as those of the bench model.

The speed reducer will be built into the wall of the pump housing with its input shaft perpendicular to the longitudinal axis of the pump as it was in the bench model. It had been planned earlier to incorporate a bevel gear set in the IVBM speed reducer to make the input shaft parallel to the longitudinal axis of the pump. However, information recently supplied by the University of Utah indicated that a perpendicular input shaft was preferred because this arrangement permits the blood pump to fit better in the calf's thoracic cavity. The speed reducer will step down the pump drive speed from 900 rpm at the flexible shaft to 120 rpm at the pump reciprocating mechanism as it did in the bench model, but a single Evoloid gear set instead of a planetary gear arrangement will be used to obtain the desired 7.5 to 1 speed reduction. Evoloid gears will be used because they have a special tooth form that permits a large speed reduction with high efficiency in a single step. This feature allows the speed reducer envelope to be made somewhat smaller than it could be with a planetary gear arrangement. The shaft speed of the bench model has been retained to avoid increasing the shaft torque loading and to allow an Evoloid gear set design that has been tested in the development program to be used in the IVBM.

The drive mechanism of the bench model blood pump was lubricated by packing the ball bearings with grease and using dry lubricants in the linear bushing ball retainers, speed reducer gears and flexible shaft guide tube. The IVBM pump mechanism will be lubricated by filling the mechanism cavity with oil. In addition to lubricating the moving parts of the mechanism, the oil will make the blood pump virtually insensitive to changes in barometer pressure. After the drive mechanism cavity is filled with oil and bled, only an extremely small amount of air can remain trapped in it. Any change in ambient pressure after the drive mechanism is sealed will cause a change in the volume of the trapped air, but this volume change will be very small and will not perceptibly affect the pump ventricles.

BELLOWS COOLANT PUMP

The bellows arrangement shown in Figure 7-1 has been incorporated in the blood pump to circulate water that will transport the thermal converter waste heat to the pump. The heated water from the converter will reject heat across the walls of the bellows to the oil in the pump mechanism chamber. Then, the heat is transferred from the oil to the blood in the ventricles by conduction through the pump components. Because the travel of the pump displacement cup yoke during the pumping cycle is twice as large as that of the yoke housing, a relative motion of one half the yoke travel occurs between the ends of yoke drive rods and the top and bottom walls of the yoke housing. This motion is used to actuate the bellows, which are installed around the yoke housing bosses for the linear bushings that guide the yoke.

During half of the pumping cycle, water is pumped by one of the bellows, A, through a discharge check valve, C, into the other bellows, B. In the other half of the cycle, water is pumped by bellows B through the engine heat exchanger and suction check valve, D, into bellows A. Check valve C prevents the water discharged by bellows B from returning to Bellows A and bypassing the converter heat exchanger. Check valve D prevents the water discharged by bellows A from entering the return line of the converter heat exchanger.

Figure 7-1. Converter Coolant System Schematic Diagram

Since both the check valves are located outside the blood pump, line E functions both as a fill and a discharge line. The heated water left in this line after bellows A is fully extended will not enter this bellows and will reject heat mainly through the wall of bellows B, but the balance of the water from the converter heat exchanger will reject heat through both bellows. However, since the volume of water in line E will be less than 25 percent of the stroke volume of one of the bellows, the maximum water temperature will not be significantly higher than it would be if all of the heated water from the converter entered bellows A.

Because the same dimensions will be specified for manufacturing both bellows, only a very small compliance volume is required in the water loop to accommodate any difference in the stroke volumes of the two bellows. A small diaphragm has been incorporated in one of the end closures of one of the bellows to provide the compliance volume required. This diaphragm will bulge either inwardly or outwardly as required to provide a coolant loop volume that will remain constant during the pumping cycle.

The lines that carry the water from the bellows to the blood pump housing must be either flexible or flexibly connected to the housing wall to accommodate the relative motion between the bellows and pump housing during the pumping cycle. Because lines that will fit in the blood pump and satisfy the coolant pumping power loss requirements are quite stiff, the lines are flexibly secured to the housing wall with beam flexures. Since the flexures will permit the ends of the lines to rotate slightly, small bellows are used to connect the ends of the lines to ports built into the pump housing wall.

TASK 8 – SYSTEM DESIGN & INTEGRATION

The conceptual design of the Intermediate System was completed during the reporting period and a topical report has been prepared describing this design along with the factors that led to selection of design features. Design work on the IVBM was initiated during the quarter and a status review is presented under this task.

INTERMEDIATE SYSTEM CONCEPTUAL DESIGN

The Intermediate System conceptual design is illustrated in Figures 8-1 and 8-2. There are two major subassemblies: a thermal converter and a blood pump. The two are interconnected by: (1) a flexible mechanical shaft to transmit drive power, and (2) a coolant line to carry the waste heat from the thermal converter to a heat exchanger located in the blood pump assembly. A summary of system characteristics is presented in Table 8-1. The Intermediate System, sized to meet its performance goals, has approached but not attained all of its weight and volume objectives. This is indicative that component development work is needed prior to the design maturing via the preliminary and detailed design stages.

Estimates of the thermal converter dimensions are as follows: 17.3 cm high, 9.6 cm long, and 7 cm wide. These dimensions are to be compared with the goals projected for the Intermediate System and Prototype Systems at the conclusion of the Phase II program.

	<u>Intermediate</u>	<u>Prototype</u>
● Weight	1.7 kg	1.5 kg
● Volume	0.8 liter	0.7 liter
● Envelope	17.0 cm x 9.5 cm x 8.5 cm	16.0 cm x 8.7 cm x 7.7 cm

*Values given in Nuclear-Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report, July-September, 1973, USAEC Contract AT(11-1)-3043 are in error.

Figure 8-1. Intermediate System Thermal Converter

Figure 8-2. Intermediate System Blood Pump

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TABLE 8-1
SUMMARY OF INTERMEDIATE SYSTEM CHARACTERISTICS

Maximum Power Output - Watts

●	Blood pump left ventricle	2.7
●	Blood pump right ventricle	0.4
●	Power pack	5.0

Isotope Inventory - Watts (t)

●	Initially	27
●	After ten years	25

Temperature - K (°F)

●	Peak material	1190	(1700)
●	Peak gas	1000	(1340)
●	Minimum gas	335	(144)

Working Gas

Krypton

Weight & Volume

	<u>Gm</u>	<u>CC</u>
● Thermal converter	1764	740
● Blood pump	791	725
● Converter and Pump	2555	1465

The principal components of the thermal converter, shown in Figure 8-1, consist of:

- A rhombic drive Stirling-cycle heat engine connected to the flexible drive shaft through step-up gearing and a flywheel.
- A plutonium-238 heat source to provide thermal energy to the engine.
- Vacuum foil thermal insulation to minimize heat losses from the heat source and the high temperature portions of the engine.
- A control system that maintains nearly constant engine speed.
- A heat remover (cooler) that delivers engine waste heat to the coolant line connecting to the blood pump.

An operating mode was selected for the Intermediate System which provides a nearly constant blood pump beat rate of 120 beats per minute by controlling the thermal converter piston speed to an average value of 600 revolutions per minute (rpm). Starling's Law of Blood Flow is satisfied by designing the blood pump to accept whatever amount of blood is delivered to the ventricles during diastole, and letting the stroke volume vary up to the maximum stroke volume which will provide 12 liters/min. This operating mode takes advantage of the inherent adjustment of the body's circulatory system via variable venous pressure.

The radioisotope heat source and Stirling engine are sized to meet the maximum design power demand of the blood pump, 3.1 watts (hydraulic). This is done so that the thermal converter can use a simple controller that regulates the converter's power output to follow the power demanded by the blood pump. The controller consists of a hydraulic pump within the converter that serves as a sensing element and a control valve. Flow losses caused by the valve regulate the net converter output.

The heat source is a vented plutonium-238 fuel capsule. The Stirling engine uses a single cylinder, stationary regenerator and rhombic drive. Varying gas pressure on the power piston

provides power input to the rhombic drive, which converts the oscillating motion of the power piston to rotary motion. The rhombic drive, in turn, drives the displacer. The function of the displacer is to move the working gas from the hot end of the working volume to the cold end and back again to provide the pressure pulses which drive the power piston. The design of the engine is such that the gas is forced to flow through the cooler, the regenerator, and heater as it is transferred between the two ends of the engine.

The flywheel operates at high speed to minimize its weight and size. Since the blood pump delivers a pulsatile flow of blood, the input power to the pump is pulsatile also. The flywheel smooths this pulsatile requirement so that the load on the engine is nearly uniform. To reduce friction losses, the flywheel runs in a dry, gas filled cavity. A magnetic coupling closes one end of this cavity; engine oil is kept out of the cavity by means of a screw seal on the flywheel shaft.

The crankcase is pressurized to the mean operating gas pressure. Pressurizing the crankcase also minimizes the pressure loads on the displacer and power piston bellows,^{*} decreasing their stress levels and increasing their working life. To allow the crankcase to be pressurized, a hermetic seal is provided by the magnetic coupling.

The vacuum foil insulation consists of nested nickel cups enclosed in a titanium vacuum vessel. A copper pinch off tube is provided to allow evacuation of the insulation after assembly and fueling. The length of each cup is adjusted so that the temperature of the cup matches the temperature of the hot finger at the point of contact. Therefore, each cup is maintained in an isothermal condition, and no heat is conducted along the cup surface.

The heat not converted into mechanical work by the engine is extracted from the gas by the cooler, and transported through the cooler walls into a vapor chamber that forms the evapora-

* Previously shown as roll sock seals in Figure 8-1, Nuclear-Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report, July-September, 1973, USAEC Contract AT(11-1)-3043.

tor end of the flexible heat pipe. A mesh screen maintains a thin water film on the outer surface of the cooler. Vapor evaporated off of this surface is transported through the heat pipe to the blood pump, where it is condensed, giving up heat to the blood pump. The condensate is returned to the thermal converter through the heat pipe annulus where pumping action is provided by capillary forces.

The entire thermal converter is enclosed in a titanium housing to provide a smooth outside contour. Because the cool portion of the converter is approximately 11°C hotter than the surrounding body tissue, a layer of polystyrene foam insulation is provided to protect the tissue. The entire converter is sealed with a thin layer of silastic. The helium vent tube from the fuel capsule cavity terminates under this silastic layer, and the helium diffuses through the silastic into the body tissues and is transported to the atmosphere through the lungs.

The principal components of the blood pump, shown in Figure 8-2 are:

- Blood pump right and left ventricles
- A drive mechanism, which converts the rotary output of the flexible shaft to motion of pusher cups that drive the ventricles.

The outside diameter of the blood pump body is 9.3 cm and the overall length is 13 cm. A heat transfer jacket and a layer of thermal insulation are wrapped around the outside surface of the pump body. The jacket provides a condensation chamber for the vapor in the heat pipe, and the insulation reduces heat rejection rates to surrounding tissue to tolerable levels. The waste heat in the jacket is conducted through the pump housing to the lubricating oil in the drive mechanism chamber. From there the heat is transferred by conduction through the pump components and ventricles to the blood.

The speed reducer built into the wall of the blood pump reduces the drive speed from 1800 rpm at the flexible shaft to 120 rpm at the shaft that powers the pump drive mechanism. In addition, the speed reducer incorporates a set of right angle bevel gears at the input side of a set

of EvoloidTM gears. These gears permit the input shaft of the speed reducer to be parallel to the axis of the pump and allow a compact coupling of the flexible shaft to the pump.

The output shaft of the speed reducer is connected through a square spline to a crankshaft with three throws. A roller on the center throw engages a Scotch yoke that converts the rotary motion of the crankshaft into a reciprocating motion with one cycle per revolution of the crankshaft. Drive rods on each side of the Scotch yoke are connected to displacement cups which alternately compress and expand the two dacron reinforced silastic pumping sacks (blood pump ventricles).

The offset of the end throws of the crankshaft allow the crankshaft ends to move past the mid-plane of the pump and permit a displacer cup to occupy space required by the opposite displacer cup in the return stroke. This arrangement minimizes the overall pump length. Rollers on the end throws of the crankshaft engage Scotch yokes built into the ends of a housing and cause it to reciprocate as the crankcase rotates. This housing supports linear ball bushings that guide the displacer cup yoke. A bellows secured to the yoke housing and the displacer cup yoke serves as a torsion resisting element that prevents relative motion between these components.

The entire blood drive mechanism and flexible shaft casing is filled with oil. The oil provides lubrication for all drive components and avoids the need for a compliance bag to accommodate pump volume change that could result from variations in ambient pressure.

The flexible drive shaft assembly, as presently conceived but which is under extensive analysis, consists of a very flexible core, a special wear resistant teflon guide tube for the core, and a convoluted metal tube around the guide tube. The convoluted metal tube permits flexibility of the shaft assembly to be maintaining. It also prevents squeezing of the guide tube on the core and seals off the shaft assembly from body fluids. End fittings are welded to the ends of the convoluted metal tube and connect the shaft assembly to the thermal converter and the blood pump. These fittings are sealed with silicone "O" rings.

IVBM DESIGN STATUS

Figure 8-3 is a preliminary layout of the IVBM converter. Changes in this design are to be expected as detail design proceeds. The IVBM converter, like the Intermediate System, will have a stationary regenerator with a radial slit heater and cooler. Unlike the Intermediate System, the crankcase will be at ambient pressure, and roll sock seals will be utilized, with a piston ring on the power piston. An integral oil pump will be included to provide back-up oil for the roll socks and lubrication for the bearings. An energy balance for the thermal converter operating under full load at the beginning of use is shown in Figure 8-4. This balance has been somewhat revised since the previous report.

Two versions of the hot finger assembly will be constructed, an electrically heated version for in vitro testing and an isotopically heated version using the presently available 33 watt source for in vivo testing. To allow for disassembly and repair of the electrical heater without disturbing the working components of the engine, the electric heater is located outside of the engine working volume. This also permits the electrical leads to be located outside of the engine working volume, where there is no interference problem with the engine working components, without requiring high temperature (about 1200 K (1700°F)) vacuum tight seals. Because the in vivo model will have a sealed insulation system, the isotope cannot be located in the insulation chamber. The helium evolved by the isotope would seriously reduce the effectiveness of the insulation over a period of time if it were allowed to vent into this region. Providing a helium tight seal in this region appears to be impractical. One approach involves transporting the isotope to the field in the presently available shipping container. This would involve the seal being made as a field operation during assembly. For these reasons, the isotope heated version will have the heat source located inside the engine working volume. The helium vented from the capsule will be released inside the engine volume. Diffusion through the roll socks and, eventually, through the blood pump diaphragm to the blood will provide a path for the helium to escape to the atmosphere.

Figure 8-3. Preliminary Layout of IVBM Converter

Figure 8-4. IVBM Energy Balance with Stationary Regenerator

The flexible shaft will be connected to the engine crankshaft through a pair of bevel gears providing a 1.5 to 1 step up to a flexible shaft speed of 900 rpm. The right angle drive formed by the bevel gears is required to obtain the optimum shaft orientation for implantation. It was decided to set the shaft speed at 900 rpm to reduce the torque load on the shaft. This permits the use of presently available shafting and blood pump speed reducer.

The flywheel will be driven at 1800 rpm, and located in a separate cavity at the opposite end of the crankcase from the power take off. Provision for starting the system will be made by providing an access path into the flywheel cavity. As all components of the system outside the roll socks are at roughly ambient pressure, screw seals will be utilized throughout. Two control concepts, utilizing the oil pump as a power source and sensor, are being investigated and are discussed under Task 6. The heat rejection loop will be a pumped loop using water as the heat transport system. This loop is discussed in more detail under Task 4 and Task 7.

SPEED CONTROL REQUIREMENTS

Work carried out last quarter indicated that variable speed operations of the blood pump would not appreciably improve the blood pump response to variations in venous return pressure. The addition of the temperature control system necessary to allow variable speed operation of the engine, therefore, is not warranted, and the work performed in the control area has been concentrated on control system designed to hold the converter speed more or less constant over a range of loads. Present efforts are directed toward defining criteria for adequate speed control performance and identifying control systems to meet these criteria. The control criteria fall generally into two categories; transient and steady state. The transient response of the system must be such that flow variations initiated by changes in venous pressure are accomplished in 5 to 10 beats (2.5-5 seconds), with no danger of stalling under conditions of rapidly increasing the load. This requirement is fairly clear. The required steady state characteristics are presently being examined. In general with no temperature control, the amount of heat extracted from the isotope increases with engine speed and isotope temperature. Since the isotope power level is constant, this implies that an increase in engine speed produces a

decrease in isotope temperature and vice versa. With the isotope power levels contemplated for any of the systems, and the expected mass of the isotope capsule, the time constant for this temperature variation can be expected to be at least 1/4 hour, and probably considerably greater. As a result, if the engine is allowed to speed up too much during periods of light load, the hot finger might cool down to the point where the system would be unable to produce its full power output for a much longer period than permitted under the transient criteria. The use of a throttle valve between the working volume and a control volume tends to minimize this effect. The action of the controller changes the phase relationship between the gas volume and temperature. With the valve open, this effect is expected to reduce the heat input to the system somewhat more or less compensating for the increased heat input due to the increased speed. A study of engine performance and instantaneous heat input for a range of speeds and hot finger temperatures is being pursued for the IVBM design. The results of this study will be used to establish criteria for acceptable speed variations with load.

TASK 9 - RELIABILITY ENGINEERING

Reliability activities during this quarter centered about design review for the IVBM and the Intermediate System, upgrading the failure mode effects analysis (FMEA) for the Intermediate System, component testing and component failure analysis.

DESIGN ANALYSIS

Design analyses associated with the Intermediate System resulted in finalization of power train configurations for the converter as well as definition of a magnetic coupling concept. In addition, the conceptual design uses bellows seals in the engine as a replacement for roll sock seals. Testing experience with the bench model engine has verified that this is probably the most critical component in terms of both likelihood of failure and its failure effect.

A proposal by Philips to (1) use the lube pump only for speed control, and (2) utilize a bearing design which obviates the need for forced lubrication and reduces bearing area requirements offers both advantages and disadvantages from the reliability viewpoint. Using the lube pump only for speed control reduces the failure effects of the pump to a less critical category, but the reliability potential of the bearings may be reduced because of increased accuracy requirements, contamination and air entrapment and inability to verify the quality of the assembled shaft-bearing configurations.

RELIABILITY TESTING

The Phase II Bench Model Thermal Converter, No. 2, was put on endurance test during this quarter. The test setup and events during the testing are covered under Task 15. This unit had been repaired by replacement of a damaged inner roll sock seal*. Failure of the engine occurred on 11-21-73 after 729 hours of operation and the engine was returned to Philips for

* Nuclear-Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report, July-September, 1973, pp 9-2, 9-7.

failure analysis. The analysis showed the cause of failure was again the inner roll sock seal. The nature of the failure was a small perforation in the roll sock. The perforation had the appearance of a cut caused by foreign object damage. However, no foreign object was found so that a failure because of basic material characteristics cannot be positively ruled out.

In vivo testing of the blood pump and flexible shaft is part of current efforts at the University of Utah. During testing there, failures of the long flexible shaft have been experienced. Analysis of early failures pointed to the need for a more rugged device which was used on later tests. A failure in the latest test, discussed under Task 16, has raised the question about application environment, since this shaft is mounted externally and is subject to handling and movement and is considerably longer than an implantable shaft with the thermal converter. A detailed failure analysis of the shaft from the latest test is planned for early in the next quarter, so that mechanisms can be determined and margins provided for other field test applications and for the IVBM design.

FAILURE MODES EFFECTS AND CRITICALITY ANALYSES (FMEA)

The FMEA for the Intermediate System has been updated and derived comments and recommendations were used in the design reviews. Changes of significance included finalization of the magnetic coupling, new bearing design for the engine, incorporation of the heat pipe, and using the lube pump only for speed control.

TASK 15 - INTEGRITY AND LIFE TESTING

The Phase II Bench Model Thermal Converter No. 2 was placed on life test on October 22. Initial converter operation was without incident and produced a torque of ~ 3.6 inch-oz. at ~ 930 RPM using a 33 watt Pu-238 heat source. The life test setup is shown in Figure 15-1.

During early November, several of the life test support equipment components failed. On November 1, 1973, an underspeed-overspeed alarm triggered because the dynamometer used to maintain thermal converter load had failed. During the time the repair took place, the load was held manually. Also, on November 7, 1973, the air regulator used to supply air to operate the vacuum system failed. The regulator from Bench No. 2 was used as the replacement. An N_2 bottle and regulator were added to the system as a backup in case of another failure. During these equipment failures, the bench model converter continued to operate satisfactorily.

On November 21, 1973, the converter ceased operation with 729 hours cumulative time at power. Test instrumentation indicated the failure was caused by a loss of engine compression. Later teardown inspection of the thermal converter revealed a perforation in the inner roll sock, confirming the test instrument assessment. The nature of the perforation has been covered previously under Task 9.

Figure 15-1. Thermal Converter No. 2 on Life Test

TASK 16 - FIELD SUPPORT ACTIVITIES

Support was provided for another in-vivo blood pump experiment at the University of Utah. The blood pump was implanted in a 105 Kg calf and driven by an external electric motor through a 36 inch long flexible drive shaft. The experiment was started on December 11, 1973 and terminated 25 hours after implantation when the flexible shaft core broke. Both the blood pump drive arrangement and flexible shaft that were used were similar to those used in the previous in-vivo experiment started on September 27, 1973. In that experiment, the drive system operated satisfactorily for 69 hours, and the poor condition of the animal after that period of time caused the experiment to be discontinued.

For the shaft assembly used in the latest experiment, the core failure occurred near the end of the casing fitting connected to the blood pump. No permanent deformation of the casing near the core failure was observed. A failure analysis is being performed to determine the cause of the core failure.

For the shaft assembly used in the previous experiment, the shaft core exhibited no evidence of damage or wear, but the casing was permanently bent where it was attached to the end fitting connected to the blood pump. It is believed that this permanent bend resulted from severe bending of the casing near the blood pump end fitting either during the violent movements of the calf in its cage or when the calf rested on the floor of the cage. The bend did not noticeably affect the measured motor output torque required to drive the blood pump throughout the experiment. However, since the bend appeared to be sharp enough to significantly affect the fatigue life of the shaft, a measurement was made in the post test examination to estimate the bend radius that could have been developed during the experiment. This radius was found to be 1-5/8 inches. The shaft supplier, S. S. White, has estimated that the shaft should have a life of at least 30 days with a minimum operating bend radius of 3-3/4 inches, but the life can drop as low as 3 to 6 days when the bend radius is reduced to 1-5/8 inches.

An 8 inch long flexible shaft was procured from S. S. White for in-vivo experiments where the blood pump will be driven by an electric motor implanted inside the animal. This drive arrangement should avoid the severe flexible shaft bending due to external loading that can occur when the blood pump is driven by an external electric motor. The first experiment with the internal drive arrangement will be started early in January.