Westinghouse Astronuclear Laboratory

NUCLEAR-POWERED ARTIFICIAL HEART PROTOTYPE SYSTEM DEVELOPMENT PROGRAM

QUARTERLY PROGRESS REPORT

January 1, 1974
Through
March 31, 1974

MASTER

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NUCLEAR-POWERED ARTIFICIAL HEART
PROTOTYPE SYSTEM DEVELOPMENT PROGRAM

PHASE III

QUARTERLY PROGRESS REPORT

January 1, 1974 through March 31, 1974

Westinghouse Astronuclear Laboratory
P. O. Box 10864
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Prepared for
UNITED STATES ATOMIC ENERGY COMMISSION

Under Contract No. AT(11-1)-3043

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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 January 1, 1974 through March 31, 1974.

Thermal insulation component development continued during the reporting period under Task 1. The thermal evaluation of the 40 nickel cup pack in the vertical orientation was completed. Five nickel cups with plasma sprayed zirconia were received, and thermal evaluation was initiated. Additional calibration testing on the guarded heater systems was completed, and these results were analyzed. The outgassing evaluation continued and calibration of the mass spectrometer system is being completed.

Thermal converter component development efforts continued on four of the components originally chosen for further development: the hot finger, the control system, the bellows seals, and the oil pump. In addition, efforts were initiated on the development of shaft seals. 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. Development efforts were pursued on the flexible shaft including analysis of calf implantation experiments at Utah. The final results of the speed reducer and liquid filled drive tests were analyzed. A pump housing deflection test was conducted and bearing and lubricant investigations continued. A materials evaluation for the IVBM drive mechanism component was completed.

Auxiliary components development had continued emphasis during the quarter on the waste heat coolant loop and coolant pump. Design analyses were completed for the coolant loop being developed in preparation for the IVBM. Results of these analyses are presented under Task 4.

Design of the IVBM thermal insulation packages continued under Task 5. Drawings were issued for machining mandrels and electroplating nickel cups on the mandrels. Layout drawings were prepared for the 60 cup package and the outer vacuum container.

Under Task 6, the design of the IVBM thermal converter is nearing completion. Items remaining to be done include: the detailing of the nuclear hot finger, the final lay-out of the oil distribution circuit, and the detailing of the new throttling valve for the control system.

Design of the IVBM blood pump drive mechanism continued under Task 7. Layout drawings of many of the components were completed. The bellows coolant pump configuration was modified and materials were selected for the blood pump drive mechanism components.

An engineering evaluation of two stage implantation was initiated during the reporting period. As part of this evaluation, a tentative implantation protocol was prepared for both stages and is discussed under Task 8.

During the quarter, a design review of the IVBM was held on the thermal converter and blood pump configurations. The results of the design review have been documented and action items are in the process of being closed out. Failure analysis of the flexible shaft was conducted. These efforts are discussed under Task 9.

Fabrication activities for the IVBM thermal insulation packages were initiated under Task 10. Under Task 11, significant progress was made on fabrication of the thermal converter components.

Under Task 14, two mock circulation units were completed and checked out.

TASK 1 - THERMAL INSULATION DEVELOPMENT

During this quarter the thermal evaluation of the 40 nickel cup pack in the vertical orientation was completed. Five nickel cups with a 50% surface area coverage of plasma sprayed ZrO_2 were received, and thermal evaluation of this 5 cup pack was initiated. Additional calibration testing on the guarded heater systems was completed, and these results were analyzed. The outgassing evaluation has continued after a system modification described in the last report, and calibration of the mass spectrometer system is being completed. Technical progress on each of these efforts is reported below.

STATIC VACUUM TECHNOLOGY DEVELOPMENT

Outgas Identification

Investigation of the outgassing characteristics of materials that see vacuum in the thermal converter was continued. The materials tested and/or conditions evaluated during this quarter are listed in Table 1-1. A total of 185 different mass scans was made, and the data have been recorded. Final reduction of the data will, however, not be accomplished until completion of the calibration runs on the gas analyzer system and finalization of a computer computation program.

The outgassing behavior of the as-plated nickel foil (Sample No. 4, Table 1-1) did not differ significantly from the vacuum treated nickel foil previously reported*. The outgassing behavior of the Udimet 700 (Sample No. 6, Table 1-1) was also similar to Sample No. 4. The Udimet 700 was double vacuum melted and was given prior vacuum heat treatments at 1400 - 2100°F. Thus, nothing but surface absorbed gases were anticipated and this was experimentally confirmed.

*Nuclear Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report, October-December, 1973, COO-3043-13.

TABLE 1-1
MATERIALS EVALUATED FOR OUTGASSING CHARACTERISTICS
AND THEIR PROCESSING HISTORY

<u>Sample No.</u>	<u>Sample or Condition</u>	<u>Scan No.</u>	<u>Temp. (°C)</u>	<u>Cleaning and Processing History</u>
4	Ni foil - as-plated Area = ~12.3 cm ² Wt. = 0.07605g	104 - 117	RT*	Ultrasonic degrease - M-6 Oxylene Rinsed - ethyl alcohol Handled with tweezers
		118 - 125	193	
		126 - 129	RT	
		130 - 135	427	
		136 - 142	RT	
		143 - 146	649	
		147 - 151	871	
		152 - 154	RT	
5	Empty System	155 - 162	RT	No sample
		163 - 172	204	
		173 - 180	427	
		181 - 186	638	
		187 - 194	868	
		195 - 214	816	
6	Udimet 700 Area = 7.54 cm ² Wt. = 8.719 g	215 - 221	RT	Same as Sample No. 4
		222 - 228	193	
		229 - 235	199	
		236 - 241	421	
		242 - 247	649	
		248 - 254	871	
7	N ₂ Calibration	255 - 289	RT	Capillary gas inlet system used in place of furnace.

*Room Temperature

The calibration procedure has been started, and all outgassing evaluation test system components are operating satisfactorily. However, sensitivity for N_2 is lower than expected by a factor of 6-8. Because of good agreement between the various pressure sensors in the system, it is suspected that the actual conductance of the 30 cm long by 0.03 cm diameter precision bore capillary used as the calibrated leak is lower than that calculated from the manufacturer's specifications. To eliminate any uncertainties, a set of calibrated leaks (calibrated at 1.5×10^{-5} , 1.5×10^{-4} , and 1.5×10^{-3} liters/sec for N_2) have been procured and will be used for system calibration.

From the outgassing data obtained thus far, there does not appear to be any problem in achieving the IVBM static vacuum objectives or in providing and verifying the vacuum processing procedures for the IVBM system. The major thrust of the work described above is to define long term static vacuum behavior which must be documented by the time the prototype system is being designed.

SPACER AND CUP TECHNOLOGY DEVELOPMENT

Several electroformed nickel cups have been electroplated with chromium. The resistance to elevated temperature deformation of these cups and nickel cups of varying thicknesses will be investigated in addition to the Cr_2O_3 emissivity properties described previously.*

ZrO_2 spacer material was applied by plasma spraying to the outer surfaces of a set of 5 nickel cups. To achieve an approximate 50% surface area coverage, a wire mesh screen was used as the plasma spraying mask. The as-plasma sprayed cups and ZrO_2 spacer patterns are shown in Figure 1-1. Since plasma spraying had to be performed with the cups in place on the mandrel, it was not possible to accurately determine the weight of the applied ZrO_2 spacer material. Prior to plasma spraying, the surface of the nickel foil cups was grit

**ibid.*, Pg. 1-1.

(A) |←1 inch→|

|←1 inch→|

(B)

Figure 1-1. Photographs of Electroplated Ni Cups Plasma Sprayed with ZrO_2 Through a Wire Mesh Screen

blasted using the same wire mesh screen as a mask that was to be used for plasma spraying. Adherence of the plasma sprayed ZrO_2 was excellent. Thermal insulation of the 5-cup pack was started, and the results are discussed later.

Two additional electroplated Ni-B cups with alternate heat treatments designed to relieve plating stresses are expected from DuPont, and an attempt will be made to etch these from the mandrel. To date, all attempts to etch the Ni-B cups from the mandrels have proved unsuccessful.

The infrared scattering investigation has resulted in a definition of some areas where further improvement in insulation performance could be realized. First, the emissivity of a metal cup surface is directly related to the thermal performance; the lower the emissivity, the lower the heat loss. Secondly, the spacer material could be contributing to heat loss either by conduction between foil layers or by altering the emissivity of the cup surface. ZrO_2 , the current spacer material, was chosen because of its low thermal conductivity. However, if the spacer is affecting the emittance, which is the most important heat loss mechanism, then one should choose a spacer material which has the lowest emissivity in the infrared radiation wave lengths which operate in a thermal system. In this case, the thermal conductivity of the spacer would be a secondary consideration. Measurement of total hemispherical emissivity of materials at 870°C is a major problem since no standardized test procedure exists. (Thus, the published data are not totally reliable.) However, an attempt will be made to collect the emissivities of alternate metals which could be used as cup materials. In addition, in an attempt to define spacer materials which do not behave as a black body at certain infrared wave lengths, the absorption characteristics of possible spacer materials will be compiled from the literature. From these data, it might be possible to design an optimized cup pack test system which would embody the principles mentioned.

THERMAL EVALUATION TEST APPARATUS

The thermal evaluation test apparatus components are shown in Figure 1-2. After assembly, this system will be used to evaluate the test insulation package and the IVBM insulation systems. The thermal evaluation test apparatus will be an all welded system using type 304 stainless steel as the simulated hot finger. The wall thickness in the section simulating the cylinder wall has been measured to within ± 0.0001 inch. A measurement of this precision reduces uncertainties in heat loss calculations down the thin wall to about 3.5%. Sheathed chromel-alumel thermocouples will be used to instrument the system. A duplicate hot finger is planned to permit assembly of the first IVBM package while the test insulation package is being evaluated.

TEST INSULATION PACKAGE

The test insulation pack has been designed as a 30 cup system with integral bottom radiation shields. This concept holds the promise of providing the maximum weight reduction and minimum insulation system surface area. This concept, however, cannot be used on the IVBM system because of disassembly difficulties resulting from the dual diameters on the outside of the hot finger. The bottom radiation shields on disassembly after performance testing would not readily be removable from the hot finger. Also, to compensate for the possibility of replacing the heater in the electrically heated system and the requirement to thermal test the insulation systems before final assembly onto the hot finger, the IVBM insulation packages were designed with straight-sided cups and built up "fluff-foil" at the bottom. This built up insulation at the bottom is the same as that designed and utilized satisfactorily in the bottom area of the bench model system. This now defines the test insulation package as a demonstration of the feasibility of utilizing the integral bottom radiation shields as the insulation approach for the intermediate and prototype system. The test insulation package, however, regardless of insulation configuration will provide the first total insulation system outgassing

Figure 1-2. Thermal Evaluation Test Apparatus Showing Disassembled Components

test and will serve to evaluate the thermal measurement techniques prior to evaluation of the IVBM insulation system. The cups for the test insulation system will be made in-house because of a delay in a commitment from a prospective vendor.

CUP PACK INVESTIGATIONS

After 47 days of testing with no measurable thermal degradation, the 40-cup pack test in the vertical position was terminated on February 16, 1974 by a vacuum failure. The test results verify that spacer application by dipping in a slurry of ZrO_2 and ethyl alcohol will be satisfactory for the IVBM. As a result, this coating procedure* will be used to coat the IVBM insulation cups.

After the vacuum system failure, the 40-cup pack was removed from the guarded heater hot finger. The first two internal cups were removed from the guarded heater hot finger for inspection. They were discolored as a result of the exposure to air and pump oil vapors when the vacuum system malfunctioned. The discolored film was thin enough to refract various colors. The two inner cups were easily removed and exhibited no evidence of sintering or bonding. The 40-cup system was then reassembled and 26 additional days of thermal testing in the vertical position have been accumulated.

Test results after the vacuum system malfunction showed a 0.3 W increase in heat loss, apparently due to discoloration of the foil cups and the resultant increase in emissivity. Table 1-2 lists the equilibrium heat loss values for the 40-cup pack tests before and after the vacuum system malfunction. These results are plotted in Figure 1-3. Also included in Table 1-2 are the first two equilibrium data points for the 5 nickel cup pack test which is

*Ibid., pg. 1.

TABLE 1-2
EQUILIBRIUM THERMAL DATA FOR THE 40 NICKEL CUP PACK
AND THE 5 NICKEL CUP PACK*

<u>Insulation Configuration</u>	<u>Test System</u>	<u>Equilibrium Temperature (°C)</u>	<u>Heat Loss (watts)</u>
40 Cup	No. 2 Vertical	780.7	0.86
Before Vacuum		797.0	0.98
System Failure		861.3	1.52
40 Cup	No. 2 Vertical	685.1	0.40
After Vacuum		774.4	0.98
System Failure		826.2	1.52
		835.0	1.52*
		865.5	1.83
5 Cup	No. 1 Vertical	693.0	5.36
Plasma Sprayed		822.0	10.25

* Two weeks at same power setting indicating some cleanup of system.

Figure 1-3. Total Heat Loss as a Function of Temperature for the Nickel Foil Cup Pack Tests

evaluating the plasma flame sprayed ZrO_2 applied through the wire mesh. The heat loss through this 5-cup system is 2.3 watts greater than the average heat loss as determined by the least square equation described later in this report. The results are plotted in Figure 1-3.

At this point in time, it is not clear if the increased heat loss is the result of increased thermal conductivity between layers because of the spacers, or a change in emissivity. When assembling this 5-cup pack, more force was required to fit the cups together. This indicates less clearance between adjacent cups and could lead to larger thermal heat losses by conduction.

An additional 5 cups are being plasma sprayed with ZrO_2 through the wire mesh. These cups are designed to be added to the 5-cup set now on test to provide a 10-cup pack test. Because it appears that there is insufficient clearance between cups in the first 5 cups now on test as evidenced by the tight fit, alternate cups from the 10-cup pack will be assembled to form a 5-cup pack. This will minimize the thermal conductance contribution to the heat loss and enable an evaluation of the effect of the ZrO_2 sprayed mesh pattern on the effective emissivity of the nickel cups.

GUARDED HEATER TEST SYSTEM CALIBRATION PROCEDURES

Because of difficulties inherent in thermal measurements at these low heat loss levels, two standard insulation calibration systems were machined from Min-K-2002 to be used to evaluate the guarded test heaters and to have some criteria against which to evaluate individual heater performance. As guard heater designs were modified or heaters repaired, a set of heat loss measurements were made on the Min-K-2002 calibration packages. The heat loss through each Min-K-2002 calibration package was identical for a given heater configuration. Figure 1-4 shows the heat losses through Min-K-2002 for the heater designs utilized in the program. These values are presented in tabular form in Table 1-3. The first two heater designs have been previously analyzed*.

*R. C. Svedberg, An Investigation of High Performance Thermal Insulation Systems, Final Report, April 1974, WANL-M-FR-73-001.

Figure 1-4. Total Heat Loss vs. Temperature—Min-K 2002 Calibration Test Runs

TABLE 1-3
CALIBRATION DATA FROM GUARDED HEATER TEST

<u>Test</u>	<u>System</u>	<u>Equil. Temp. (°C)</u>	<u>Heat Loss (watts)</u>
MIN-K	No. 1 Vertical	855.0	12.55
		886.0	13.30
MIN-K	No. 2 Vertical	716.3	11.04
		803.4	13.35
		819.0	13.88
		871.0	15.50
MIN-K	No. 2 Horizontal	693.5	10.48
		780.5	12.85
		841.7	12.85
		869.2	15.55

Table 1-4 lists the changes in heater designs incorporated during the thermal insulation test program. The two major heater changes, which caused the largest difference in heat loss, were: Mark II change, where the guard heater was extended and Mark IV change where the stainless steel heater leveler was grooved. With the Mark IV configuration, the heat loss through the Min-K was about 2 watts greater than the calculated heat loss value at 870°C. The other modifications to the heaters resulted in a heat loss through the Min-K of about 0.5 watt below the calculated heat loss value. The spread in this thermal data implies an uncertainty in a 40 nickel cup insulation package of only ± 0.27 watt or 0.0048 watt/cm^2 . To more precisely define the limits of accuracy of the thermal test data, several more adjustments in the heat leveler configuration are required. Additional modifications to the heater design are planned and will be incorporated when the systems are down for required maintenance.

The reference heat loss vs. temperature curve for normalizing heat loss data has been previously the same as reported* and is shown on Figure 1-4. Based on the reference heat loss vs. temperature calibration line, the heat loss per cm^2 at 870°C vs. the number of foil layers is presented in Figure 1-5. The points plotted include data from a previous program* for the 4, 5, 9, 10, and 20 layer cup packs.

The data used to calculate the least squares fit include data measured for the 5-cup packs with different spacer materials, i.e., -325 mesh ZrO_2 powder, ZrO_2 fiber spacers, and the 0.100 inch plasma sprayed ZrO_2 dots, prior to degradation due to sintering.

A straight line through the points as determined by a least squares analysis of the data plotted results in the equation

$$\ln(W) = -(0.204 + 0.976 \ln N)$$

where W = heat loss in watts/cm^2

N = number of cups.

*Ibid., pg. 1-11.

TABLE 1-4
DESCRIPTION OF GUARDED INSULATION TEST SYSTEM HEATERS

1. Mark I Heater - System #1	A. 1.5 in. long guard heater B. Both heaters wound on the same tube
2. Mark II Heater - System #1	A. 2.5 in. long guard heater B. Guard and working heater wound on separate tubes. C. 0.010 in. thick Al_2O_3 oxide tube used as heater protection
3. Mark III Heater - System #1	A. A 1/8 in. thick Min-K-2002 insulation layer placed between the measurement and guard heater.
4. Mark IV Heater - System #2	A. The stainless steel heat leveler has two grooves machined 1.0 in. and 1.5 in. from the bottom.

Figure 1-5. Heat Loss per cm² as a Function of the Number of Ni Foil Cups

This will enable one to calculate the heat loss per cm^2 for a given number of foil cups. It was from the data presented in this plot that the decision to use a 60-cup insulation package on the IVBM was made. The lines at the lower left hand corner of Figure 1-5 show the heat loss range anticipated for the IVBM hot finger geometry. Bear in mind that the IVBM insulation is covering the entire hot finger area where the heat source temperatures range from 870°C to 40°C . In addition, it is difficult to analyze the effects of the fluff-foil or multi-length cup geometry on the heat losses. If one considers the heater area as that at 870°C , then a 40-cup system could be utilized. If one considers the entire hot finger area as the effective heat loss area, a 75-cup system would have to be chosen. The effects of the fluff-foil bottom and the multi-length cup geometry will be evaluated when thermal measurements are made on the insulation test package and the two IVBM insulation packages.

TASK 2 - THERMAL CONVERTER DEVELOPMENT

Efforts continued on four of the components originally chosen for further development: the hot finger, the control system, the bellows seals, and the oil pump. In addition, efforts were initiated on the development of shaft seals.

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.

The hot finger of the engine has a nickel 200 to Udimet 700 joint which is microbrazed. A metallurgical investigation was conducted to determine to what extent the brazing filler metal diffused into the parent metals. A sample joint, shown in Figure 2-1, was prepared under conditions equivalent to those used on the actual hot finger assembly. Following the completion of the braze, the joint was metallurgically investigated with the aid of the photomicrographs shown in Figures 2-2 and 2-3. As these indicate, the microbrazed diffusion into the nickel 200 is pronounced, but the part is not critical in terms of stress. The diffusion into the Udimet 700 side is minimal, and would probably pose no long range problems.

During this report period, a test facility for additional hot finger metallurgical studies was set up. The purpose of the tests being planned is the determination of the stress-rupture time values of the thin wall cylinders. The test facility consists of a tubular furnace through which dried argon is passed as a protective atmosphere; a dew point of -50°F has been attained. The inside of hot finger samples will be pressurized directly from a tank containing very high purity argon. The initial test samples, made from cast Udimet 700, will be pressurized to give a stress-rupture life of 10 hours. These results will subsequently be compared to the stress-rupture life of hot fingers machined from a powder metallurgical bar of Udimet 700. As indicated in the last report, powder metallurgically prepared Udimet 700 is envisioned

Figure 2-1. Sample Braze Joint

Figure 2-2. 35 Power Micrograph of Nickel-Udimet

Figure 2-3. 270 Power Micrograph of Nickel-Udimet

as the material best suited for the hot fingers of the intermediate engine. The bulk properties of powder metallurgical and cast Udimet 700 are:

1400°F	85,000 Psi	21 hours	Homogeneous Metal
1400°F	85,000 Psi	37 hours	Fed. Mogul
1400°F	85,000 Psi	45 hours	Special Metals

It should be emphasized that the above results are bulk properties. As pointed out by E. G. Richards (J. Inst. Met. 96, 365 (68)), the ratio of specimen diameter to grain size diameter has to be ten or better in order for bulk properties to apply. This is best fulfilled for powder metallurgical hot fingers.

CONTROL SYSTEM

The major characteristic of the engine control is that it be capable of maintaining the engine speed nearly constant, and through it the speed of the blood pump, as the torque output varies between its maximum and minimum values. This is to be accomplished while the heat input to the engine remains constant, meaning that its hot side temperature does not change, or that the change, if any, is negligible. The conditions outlined above can be met with the control illustrated in Figure 2-4.

The control scheme is essentially similar to that of the Phase II program. The discharge pressure of an engine driven oil pump actuates a short circuit valve which connects the working space of the engine with the buffer space. Opening of the valve results in an increased, but controlled internal leakage loss, which effectively reduces the output shaft torque at a slightly lower heat consumption per revolution. (See Figure 2-5.)

Thus, if the speed control system is designed with the proper gain, the heat input will remain constant, since the off-speed conditions then exactly compensate for the off-consumption per revolution.

Figure 2-4. Control System Schematic

Figure 2-5. Short Circuit Control Characteristics

As indicated in the previous quarterly report, the original throttling valve design was re-designed, as shown in Figure 2-6.

Figure 2-6. Throttling Valve Design

The new design offers the following advantages:

- It is completely pressure balanced. This means that the mean level of the cyclic fluctuations of either the working space pressure or the buffer space pressure have no influence, or do not affect the valve opening. Only the speed dependent oil pump pressure compared to a reference pressure level (crankcase) determine the size of the valve opening. Thus, pressure balance is desirable in order to avoid a frequent valve stem movement caused by the working space variation.
- Low friction due to proper shaft guidance and metal bellows sealing.
- Low dead volume at the working space connection.
- High frequency response due to the low masses of the moving elements.

The block diagram shown in Figure 2-7 represents the control circuit, in which the signals are normalized by the full load angular speed, ω_o , the full load torque T_o , the full load oil pressure p_o , and the maximum useful valve opening A_o .

Figure 2-7. Control Circuit Block Diagram

The control system is characterized by:

- The valve gain $K = \frac{\frac{A}{A_o}}{\frac{\omega}{\omega_o} - 1}$
- The influence factor K_n of the speed effects

$$K_n = a + b - \frac{A}{A_o} \quad (\text{See Figure 2-8})$$
- The response time τ_p for the oil pressure.
- The inertia integration time $\tau_i = \frac{\tau \omega_o}{T_o}$ in which τ is the mass moment of inertia.

The Nyquist diagram of Figure 2-9 and other methods of analysis show that the proposed control system will be stable if:

1. $K_n \geq 0$, or
2. $|K_n| < K$, and $|K_n| < \frac{\tau_i}{\tau_p}$

As the diagram indicates, a stable control system is predicted.

BELLOWS SEAL

Late in January, two completed bellows assemblies were received from Servometer Corp. and preliminary testing was initiated. These tests uncovered two design deficiencies in the bellows tester which have since been rectified. These problems involved leakage from the externally mounted oil pump, and stalling of the drive motor. The first was solved by incorporating the oil pump into the crankcase of the tester; the second by increasing the size of the drive shaft flywheel.

Figure 2-8. Normalized Speed - Torque Curves

Figure 2-9. Nyquist Diagram

A more serious problem involved the bellows assemblies themselves. After an estimated total of twenty minutes of operation at 1200 RPM, a check was made for leakage across the bellows. A leak was evident, and upon disassembly, a large circumferential crack was evident along the inner diameter of a central convolution. A similar occurrence followed five minutes of testing on the second bellows assembly.

The rapidity of both failures suggested either a basic defect in, or misuse of, the bellows. The manufacturer was contacted for his opinion of the failure modes. Upon examination of the failed and spare bellows, the manufacturer discovered that their stiffness was significantly greater than intended; their stroke capability was greatly compromised. The manufacturer offered to replace the bellows at no cost and this is being done.

OIL PUMP

A pump suction system has been devised to provide uninterrupted oil flow with the thermal converter in any orientation and a partially filled crankcase. Several capillary wick structures (part of the suction side of the oil system) are located in the crankcase; sufficient oil is loaded to assure that at least one of the wick structures is submerged in the oil in any attitude. The basic principle of operation is that ingestion of air by the uncovered wicks will be prevented by the surface tension force between the oil and the wick's fine wire gauze structure. The surface tension force results in an equivalent "capillary head" which must exceed the pressure difference between the air and the oil in an uncovered wick to prevent air breakthrough.

Figure 2-10 illustrates the basic concept. One wick structure is shown submerged in oil; the other is uncovered. The entire suction side of the system is assumed to be primed with oil. Point 1 is defined to be inside the submerged wick, point 2 inside the "tee" connection, and point 3 at the highest point inside the uncovered wick. The pressures at these three points are, respectively:

$$P_1 = 14.7 + \rho h_1 - QR_{\text{Entrance}} \quad (\text{where } Q = \text{volumetric flow rate, and } R_{\text{Entrance}} = \text{effective flow resistance of entrance area})$$

Figure 2-10. Capillary Wicking Pump Suction System Schematic

$$P_2 = P_1 - \rho h_2 - QR_{1-2}, \quad (\text{where } R_{1-2} = \text{effective flow resistance from point 1 to point 2})$$

$$P_3 = P_2 - \rho h_3$$

Substituting for P_1 and P_2 one obtains

$$P_3 = 14.7 - \rho [H] - Q [R_{\text{Entrance}} + R_{1-2}]$$

For successful operation, the difference in pressure between the air and the oil at point 3 must not exceed the capillary head capability of the wick structure, i.e.:

$$H_{\text{wick}} > H + \frac{Q [R_{\text{entrance}} + R_{1-2}]}{\rho}$$

The capillary head can be calculated for a given wick structure as a function of the pore radius, r_h , and the surface tension coefficient, σ ; for example when using a 325 x 325 mesh gauge:

$$H_{\text{wick}} = \frac{2\sigma}{\rho r_h} = 8.2 \text{ inches}$$

The design of the wicking system has been completed. Its function, and the new gear pump, will be evaluated as part of the pump development task.

SHAFT SEALS

The IVBM thermal converter contains two shaft seals which have undergone design verification testing this past quarter. The first seal is incorporated on the thermal converter's output shaft at the interface between the thermal converter and the blood pump gear box, and serves to minimize mixing of the two mechanisms' lubricating fluids. The second seal is located on the converter's flywheel shaft and prevents leakage of crankcase oil into the flywheel housing.

Figure 2-11 is an illustration of the output shaft seal. Two screw thread viscosity pumping grooves (one a left hand and the other a right hand thread) are utilized to pump the two fluids inward towards each other. Theoretically, the fluids should meet at an interface across which no mixing should occur. In practice, some mixing due to diffusion effects and thread imperfections must be expected with time.

Figure 2-11. Thermal Converter/Blood Pump Shaft Seal

The thread angle (ϕ), the pitch (t), the land width (e), the clearance (h_0) and the thread depth ($h-h_0$) are the same for both halves of the seal. The differences in viscosity between the two fluids is compensated for by the number of threads provided in each half of the seal.

The pressure developed by each half of the seal can be obtained from:

$$\Delta P = \frac{\mu a}{\beta + \gamma n}$$

where,

$$a = \frac{\pi D h (t-e) \cos^2 \phi}{2}$$

$$\beta = \frac{h^3 (t-e) \sin \phi \cos \phi}{12L}$$

$$\gamma = \frac{\pi^2 D^2 h_o^3 \tan \phi}{12eL}$$

$$n = \text{shaft speed, 10 rev/sec}$$

Since the thread profile is identical for either half of the seal, no flow will cross the central boundary when

$$\frac{\mu_1}{\mu_2} = \frac{L_2}{L_1}$$

The power dissipated by the shaft seal can be calculated from:

$$P = 2\pi^3 D^3 \frac{e}{t} \frac{L_1 \mu_1}{h_o} n^2$$

The power loss predicted is equal to 0.04 watt.

A prototype of the seal has been undergoing design verification testing. Figure 2-12 is a photograph of the test assembly. The reservoir at the motor end is filled with Vitrea 21 oil while the reservoir at the far side is filled with sperm oil. The seal-shaft runs in a bore connecting the

two reservoirs. To date, testing has been successful; the first test run was terminated after 30 days of operation with no detectable mixing or oil transfer from one side to another.

Figure 2-12. Thermal Converter Output Shaft Seal Test Fixture

The flywheel shaft seal combines a screw thread pumping groove with a cylindrical clearance seal to prevent leakage of crankcase oil into the flywheel cavity. In theory, the pumping groove develops sufficient pressure head and pumping capacity to prevent leakage due to pressure differences between the crankcase and flywheel cavity.

The analysis of the pumping groove is similar to that for the grooves on the output shaft seal. One can show that:

$$Q_{\text{net}} = a n - (a + \gamma) \frac{\Delta P}{\mu}$$

where

a , a , and γ have been previously defined and $n = 30 \text{ rev/sec}$.

The present seal geometry was selected to provide a positive pumping action away from the flywheel with a back pressure of one atmosphere. The power dissipation due to the screw thread portion of the seal is 0.04 watt.

A cylindrical clearance seal is provided as a backup to the screw seal. In the event that some oil migrates past the screw seal, the clearance seal serves to slow its entry into the flywheel housing. For a cylindrical seal, including its eccentricity, one can show that:

$$Q_{\text{max}} = \frac{5 \pi \Delta D^3 D \Delta P}{192 \mu L}$$

where

ΔD is the diametral clearance.

For the clearance seal provided, assuming a pressure head of 1 psi, $Q_{\text{max}} = 0.003 \text{ in}^3/\text{hr}$. This would ensure several days of testing before significant amounts of oil could accumulate in the flywheel housing. The power dissipation due to the clearance seal should not exceed 0.07 watt; this brings the total dissipation due to the flywheel shaft seal to 0.11 watt.

Figure 2-13 is a photograph of a test version of the seal. To date, testing has been a success; the unit has accumulated thirty days of operation with an equivalent crankcase overpressure of 0.5 psi with no leakage whatsoever. Further testing at higher overpressures will be conducted during the next quarter.

Figure 2-13. Flywheel Shaft Seal Test Fixture

TASK 3 - BLOOD PUMP DRIVE DEVELOPMENT

FLEXIBLE SHAFT DEVELOPMENT

Developmental efforts have been undertaken to solve the implantation problems associated with the flexible shaft (power transmission link) between the thermal converter and the blood pump drive mechanism speed reducer. A first step in this effort is to better understand the implanted environment through analysis and experiment so that reliable implantation and operational design criteria can be established. In this connection an 8 inch long flexible shaft, a bench model blood pump, and a small electric drive motor were furnished the University of Utah. The University of Utah carried out an experiment in which the blood pump and flexible shaft were implanted in a calf along with the electric drive motor.

The flexible shaft failed 35 hours after implantation. The failure occurred shortly after the animal fell on the relatively sharp edge of an angle iron frame member as the animal lunged forward out of its confinement cage. This motion imparted a substantial impact below the animal's rib cage apparently in the region where the flexible shaft is protected by only about 1/2 inch of tissue. Twenty minutes after the accident, the drive motor controller overload tripped and efforts to continue pump action by alternating between two controllers were characterized by excessive motor current indicating heavy torque loading, eventually culminating in shaft breakage. The nature of the terminal failure was a fracture of the shaft core at the motor end of the shaft. In addition, the shaft was unwound slightly at the blood pump end. The characteristics of the failure were those of an overload condition.

The shaft used in the Utah experiment had a rubber casing reinforced with a metal spiral which also served as the bearing surface for the core. The core was multilayered. The layers were wound from 0.017" wire. The inner layers used high carbon steel wire. The outermost layer was of low carbon steel wire. The core was counter-rotation wound; the outer layer tends to unwind under load.

After autopsy, a bend of about 1.7 inch radius was observed in the case near the motor end transition between the rigid and flexible portions of the case. A review of the experiment with Mr. Lee Smith of the University of Utah indicated that:

- The shaft is not subject to bending moments during the implantation procedure; it is pulled through a surgical tunnel and thereafter connected to the pump and motor.
- The shaft assumes a mild 8 inch radius bend during surgery and when the calf is standing. It assumes a nearly straight geometry when the calf is lying in its confinement cage.
- High arterial pressures (220/160) of less than four minutes duration were experienced during surgery.
- If necessary, a rigid, curved shaft casing could be employed for implantation experiments.

The following development approach is being followed. First, based on preliminary analysis of the failure and consultations with two flexible shaft manufacturers, revised shaft specifications are being prepared. These specifications will be used to procure a shaft of revised design that could be used in an implantation experiment in the near future. Second, shafts of the previous design are being subjected to bench testing in an attempt to better define the failure cause or causes.

Bench tests were performed to determine whether the shaft loads developed under the reported pressure conditions during the experiment could have caused this failure. A shaft of the same design and from the same batch as the implanted shaft was endurance tested for 196 hours without loss of load carrying ability. Some no load windup of the core was observed after the test, but the torsional spring constant remained virtually unchanged. In addition to the normal blood pressure loading (120/80 arterial pressure), four minutes of overload pressures (220/160) were imposed during this test to simulate conditions encountered at the University

of Utah during transfer of the calf from heart and lung by-pass to the implanted blood pump. Another test is planned to reproduce the stress conditions caused by the observed permanent bend of the implanted shaft. Also, a test on a new shaft of the same design as the implanted shaft is being conducted to measure compliance, hysteresis yield, and breaking load. The implanted blood pump is also being returned to WANL from the University of Utah for examination.

SPEED REDUCER AND LIQUID FILLED DRIVE EVALUATIONS

The data obtained in the efficiency test of the Evoloid gear speed reducer and the power loss test of the oil filled pump mechanism with the Evoloid gear speed reducer were analyzed. The results of these tests as well as the results of similar tests with the planetary gear speed reducer, reported in the previous quarterly, are shown in Figures 3-1 and 3-2. These results are summarized in Table 3-1.

Qualitatively, the results were as expected. The Evoloid gears showed a lower no load loss but a higher loss under load. The lower no load loss is particularly noticeable in the oil filled condition. These effects tend to balance each other. The total power loss for each gear type driving an oil filled pump at design load is shown in the last line of Table 3-1. These two values are not considered to be significantly different for the test conditions.

PUMP HOUSING DEFLECTION TEST

A test was run to determine the deflection of the bench model stainless steel pump housing under load. This was done to determine whether the deflections are low enough so that additional housing flexibility resulting from additional cutouts in the housing or the use of a housing material with a lower modulus of elasticity than that of stainless steel can be used in the IVBM without affecting the action of the yoke guide housing rollers. Maximum load is applied to the rollers when the crank is 90° before top dead center on the left ventricle. The left ventricle was pressurized to 190 mm Hg with the input shaft clamped at 90° before top dead center. This was the pressure required to produce the previously measured peak torque of 24 in-oz during normal pump operation. The pump was mounted on a surface

Figure 3-1. Power Loss vs. Input Power

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

TABLE 3-1
AVERAGE POWER LOSS IN WATTS FOR VARIOUS
PUMP DRIVE MECHANISM CONFIGURATIONS AND CONDITIONS

	Speed Reducer	
	<u>Planetary Gear</u>	<u>Evoloid Gear</u>
Pump Drive Mechanism (Including Speed Reducer) Oil Filled, No Load, 900 RPM	.45	.26
Gear Box Only, 900 RPM, 6 W Input	.45	.67
Gear Box Only, 900 RPM, No Load	.17	.08
Loss in Gear Box due to Design Pump Load	.28	.59
(1) Total Loss Oil Filled Mechanism, 6 W Input	<u>.73</u>	<u>.85</u>

(1) This total loss does not include the loss in the reciprocating mechanism due to the pumping loads. However, this loss is not influenced by the type of speed reducer used.

plate with the roller wear inserts horizontal and the shaft tipped downward so that readings could be made over the thinnest section of the inserts. Height gage readings were taken at nine points along the inserts top and bottom. The pressure was then released and the shaft returned to the zero torque position. The inserts were surveyed over the same marked points previously read.

The results showed that the housing deflects quite uniformly 0.0004 in. on the diameter due to the spreading action of the rollers. In addition, the readings were accurate enough to detect a 0.0001 in. per in. tilt of the inserts relative to the gear housing due to the moment of the two rollers. Note that the 24 in.-oz. is input shaft torque, the moment tending to twist the housing is 7.5 times this or 15 in. lbs.

The deflections measured are considered to be quite small. It is felt that diametral deflections as high as 0.001 to 0.002 inches should not significantly affect the operation of the pump mechanism.

BALL BEARING AND LUBRICANT INVESTIGATIONS

Twelve hundred hour ball bearing friction and wear tests were performed using SR4 ball bearings, the largest bearing in the IVBM pump drive mechanism, and three candidate oils for the drive mechanism cavity. All three of the oils have a viscosity of about 20 centistokes at room temperature. All the tests were made with a radial load of 50.8 lb on a set of two bearings rotating at 120 rpm. This load is double the maximum design load of the SR4 bearings in the pump mechanism.

Torque values are noted below for each of the lubricants during test:

Transformer 55 Oil		Vitrea 21		Sperm Oil	
<u>Time</u>	<u>Torque (mg-mm)</u>	<u>Time</u>	<u>Torque (mg-mm)</u>	<u>Time</u>	<u>Torque (mg-mm)</u>
24 hr	22900	24 hr	21000	24 hr	22950
456 hr	23800	192 hr	21000	168 hr	20100
1008 hr	24650	648 hr	21250	672 hr	19600
1292	25650	1272 hr	21700	1220 hr	21000

After the tests, the bearings were visually inspected by Westinghouse and the bearing manufacturer and found to be in excellent condition.

Immersion tests were performed at the Westinghouse R & D center to evaluate the compatibility of both the current silastic material, MDX 4-4514 and a more oil resistant fluoro-silicone compound, MDX 4-4026, with candidate oils for the drive mechanism cavity. Tests of 10, 30 and 38 day durations were made in accordance with the ASTM D-471-72 test procedure for determining the change in properties of elastomeric vulcanizates resulting from immersion in fluids. The oil temperature in the tests was 120°F, and the specimen size was 2" x 1" x 0.08" thick. The results of volumetric swell measurements made after each test are shown in Table 3-2. These results indicate that the present ventricle material did not undergo a significant dimensional change when immersed in sperm oil, but did experience an objectionable change when immersed in the other oils. The fluoro-silicone compound did not undergo significant dimensional change in any of the oils tested. Tests are now in progress to evaluate the effects of immersion on the mechanical properties of both elastomers.

MATERIALS EVALUATIONS

The review of candidate low density metals for the IVBM blood pump mechanism components was completed. It was concluded that beryllium is the most desirable material from the standpoint of specific stiffness, modulus of elasticity/density. However, this material has a low ductility and must undergo special treatments prior to and between machining operations to avoid cracking during manufacturing. Also, components made of beryllium must be designed with generous corner radii to avoid cracking in service. The corrosion resistance of beryllium is believed to be adequate for the expected IVBM environment. Titanium is the best compromise material with very good corrosion resistance, good ductility, fair stiffness, and fair density. Magnesium is the best material from the standpoint of density, but although its ductility is very good, its stiffness is low and its corrosion resistance when properly protected should be adequate but slightly poorer than that of beryllium. None of the above materials appear to pose a toxicity problem for the IVBM application.

TABLE 3-2
SWELL OF VENTRICLE ELASTOMERS IN MECHANISM OILS

<u>Elastomer</u>	<u>Oil</u>	% Swell for Various Days			<u>Description</u>
		<u>10 days</u>	<u>30 days</u>	<u>38 days</u>	
MDX 4-4514	Sperm	3.8	5.4	3.3	No dimensional change.
MDX 4-4514	Vitrea 21	15.0	11.6	14.5	Material widened 1/32 inch. Color changed to light yellow.
MDX 4-4514	Santotrac 30	37.6	37.9	45.0	Unknown.
MDX 4-4514	Transformer 55 plus 20% additive	----	33.8	34.4	Material lengthened 1/16 inch and yellowed.
MDX 4-4026 (Fluro-Silicone)	Sperm	2.0	2.0	0.5	None.
MDX 4-4026	Vitrea 21	1.6	1.6	1.0	Light yellow.
MDX 4-4026	Transformer 55 plus 20% additive	2.0	3.4	2.9	None.

TASK 4 - AUXILIARY COMPONENTS DEVELOPMENT

Design analyses were completed for the coolant loop being developed in preparation for the IVBM. The loop consists of a bellows coolant pumping chamber and a bellows compliance chamber located inside the blood pump mechanism and resilient tubes connecting these chambers to the outside of the blood pump mechanism. Check valves and a cross-over tube are located outside the blood pump mechanism. Polypropylene tubing connect the check valves to the thermal converter cold side heat exchanger. In operation, the coolant pumping chamber compresses, forcing coolant through the polypropylene tubing and the thermal converter heat exchanger and into the coolant compliance chamber during one half of the stroke. In the remaining half of the stroke, the compliance chamber compresses, returning the fluid to the pumping chamber through the cross-over tube. This arrangement preserves the constant volume requirement for the fluid inside the blood pump mechanism. The check valves direct the flow in the proper direction for each half of the cycle.

The coolant compliance chamber and pumping chamber are identical, and are shown in Figure 4-1. The lower two bellows shown in the figure form the pumping or compliance chamber. The upper bellows serves as a spring which drives the lower pair of bellows. The assembly shown in Figure 4-1 serves several functions. First, it prevents the blood pump pusher cups from rotating relative to the rest of the mechanism. Without this provision, relative rotation between these components would be absorbed by the ventricle diaphragms, reducing their expected lifetimes. The lower chambers, in addition to forming the working components of the coolant pump, serve as a heat exchanger to transfer the heat from the coolant to the oil filling the blood pump mechanism. These functional requirements, combined with the geometry of the space available and the lifetime requirement, limit the effective area of the pump and compliance bellows assemblies. Using the entire stroke available, the mean coolant flow for the coolant pump design at 120 beats per minute would be about 1 in³/sec. This is far higher than required, and the pumping power required to

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Figure 4-1. Coolant Bellows Pump

circulate this amount of coolant is unacceptably high. Space limitations within the blood pump mechanism prevent using lines with inside diameters significantly larger than 0.11 inches, so the power required could not be significantly reduced by enlarging the coolant lines, and the coolant flow rate had to be reduced.

Since the effective area of the bellows assemblies forming the coolant pumping and compliance chamber cannot be decreased readily, a practical way to reduce the flow is to reduce the stroke seen by each of these components. To accomplish this without altering the stroke of the blood pump mechanism, the coolant pump and compliance bellows assemblies were shortened. An additional bellows was inserted between each of the shortened assemblies and the respective pusher cups. This bellows acts as a spring (drive spring) and the total stroke is divided between each drive spring and its associated pump or compliance bellows assembly. By properly selecting the spring constant of the drive spring and its associated pump or compliance bellows assembly, the desired stroke and therefore the desired flow rate can be obtained. An analysis was performed to estimate the spring rates required to provide adequate flow for cooling the thermal converter and to estimate the expected coolant temperature for a selected pump design.

For a coolant flow rate of $0.17 \text{ in}^3/\text{sec}$, the coolant water temperature at the inlet to the thermal converter is expected to be 116°F and the outlet temperature 119°F . The greatest uncertainty in heat transfer calculations occurs across the blood pump ventricles. Assuming that the estimates for oil and blood film coefficients are accurate within a factor of two, the coolant inlet temperature could be as high as 125°F and as low as 110°F . Conclusive information on this process can be obtained by instrumenting an oil filled blood pump and measuring the thermal performance of this component. The power required by the coolant pump has been estimated at 0.06 watts. The flow rate is quite insensitive to changes in the absolute magnitude of the overall spring rate of the bellows components. On the other hand, the flow is nearly proportional to the spring rate of the drive spring divided by the sum of the drive spring rate plus the combined spring rate of the two bellows forming the pumping chamber.

TASK 5 - THERMAL INSULATION DESIGN

Drawings have been issued for machining mandrels and electroplating nickel cups on these mandrels. The 60 cup insulation system was designed to meet the following objectives.

- An 0.004 inch annular spacing between cups at operating temperature.
- Radiation leakage at the bottom miter will be minimized by utilizing a non-linear miter joint.
- Disassembly and reassembly of insulation system without damage to the nickel cups.

To insure proper fit up and to prevent binding due to thermal expansion during heatup, a layout drawing of the 60 cup system was made at 20 times size in both the hot and cold configuration. This also facilitated making the fluff-foil profile template. This template will be used during the fluff-foil buildup to maintain the proper miter joint profile.

The drawing for the electrical model outer vacuum container is shown in Figure 5-1. This container was designed with a 0.018 inch side wall thickness and an end wall thickness of 0.035 inch. The outer container will be constructed from Type 304 stainless steel. The flange fitting welded into the outer container is for connection to the vacuum system. This flange can be sealed with either a polymer or a metal gasket.

A meeting was held with Philips to finalize the feedthrough and pinchoff port location for the nuclear heated IVBM thermal converter. The material for the outer flange, feedthrough, and outer container will be Monel 400.

Electrical IVBM Outer Insulation Vacuum Container

TASK 6 - THERMAL CONVERTER DESIGN

The design of the IVBM thermal converter is nearly complete. Specifically three items remain to be done: the detailing of the nuclear hot finger, the final lay-out of the oil distribution circuit, and the detailing of the new throttling valve for the control system.

The layout of the converter, with the electrically heated hot finger, is shown in Figure 6-1. The overall dimensions are 22.2 x 11.0 x 9.0 cm, as compared with 22.0 x 10.0 x 9.0 cm, the goals set originally. The oil distribution circuit will be incorporated in the lower portion of the crankcase.

The design of the heat exchanger is being revised to reflect bends in the coolant inlet and outlet lines which would insure that the parts are on the same side of the engine for ease of coolant loop integration.

Figure 6-1. IVBM Thermal Converter

TASK 7 - BLOOD PUMP DRIVE MECHANISM DESIGN

DETAILED DESIGN

The layout drafting for the blood pump drive mechanism has been essentially completed. Layout drawings defining in detail the design of the reciprocating mechanism, speed reducer and converter coolant circulating pump were completed. Many of the drawings for manufacturing the components of these subassemblies are ready for checking.

BELLOWS COOLANT PUMP

For the reasons discussed in Task 4, a new bellows coolant pump arrangement with a smaller stroke volume than the configuration described in the previous quarterly report was integrated into the blood pump drive mechanism. A trimetric of this pump is shown in Figure 4-1. This arrangement uses the original bellows convolution geometry, but reduces the stroke volume by reducing the length of the pumping bellows assembly. A third bellows attached to the free end of the pumping bellows assembly provides the additional flexibility required to accommodate the full travel of the bellows pump actuation system.

MATERIAL SELECTION

A material selection was made for each component of the drive mechanism. The components and component materials are listed below:

<u>Component</u>	<u>Material</u>
Pump Housing	Titanium
Yoke Housing	Titanium, Magnesium ⁽¹⁾
Displacer Cups	Titanium, Magnesium ⁽¹⁾
Speed Reducer Housing	Titanium
Speed Reducer Gear	Delrin AF
Speed Reducer Pinion	17-4 PH Stainless Steel
Crankshaft	303 Stainless Steel
Scotch Yoke	440C Stainless Steel

Rollers	440C Stainless Steel
Roller Tracks	440C Stainless Steel
Ball Bearings	440C Stainless Steel
Ball Bushings	440C Stainless Steel
Roller Shafts	303 Stainless Steel

(1) Two sets of components will be made of titanium alloy, and one will be made of magnesium alloy

Titanium was selected as the material for the blood pump housing because it has a good modulus of elasticity and it is highly corrosion resistant. A stress and deflection analysis of the titanium housing using a three-dimensional computer code is underway to confirm this selection. Beryllium appeared to be a very attractive material for the housing because it is very stiff and light, but since beryllium is brittle, it was felt that a component as highly loaded and complex as the pump housing should not be made of this material before a development component is evaluated.

Magnesium was the material originally selected for the yoke housing and displacer cups because the strength and stiffness properties of this material are adequate for these components and because magnesium is the lowest density material available with the required strength properties. However, it was felt that further investigations of surface treatments would be required to ensure that this material could be adequately protected against corrosion. Therefore, it was decided to make parts for two mechanisms out of titanium and parts for a third mechanism out of magnesium. The titanium parts will maximize the probability of achieving adequate corrosion resistance in the IVBM, and the magnesium parts will provide an opportunity for evaluating this material in the IVBM in-vivo experiments.

Although magnesium would provide adequate strength for the speed reducer housing, it was decided to make this component out of titanium to avoid potential corrosion problems. Since the housing is an external pump component, body fluids could come into contact with the housing if the silastic coating that will be applied over it is damaged. Corrosion problems could then result if the surface treatment of the magnesium did not provide adequate corrosion resistance.

Hardened stainless steel was selected for the Scotch yoke, ball bearings, ball bushings, rollers, roller tracks and speed reducer pinion gear to provide the hardness required to avoid wear. The crankshaft and roller shafts will be made of 303 Stainless Steel because this material has a high fatigue strength and can be machined readily. The large gear in the speed reducer will be made of Delrin AF to allow the gear set to run quietly.

TASK 8 - SYSTEM DESIGN & INTEGRATION

Engineering Evaluation of Two Stage Implantation

An engineering evaluation of two stage implantation was initiated during the reporting period. As part of this evaluation, a tentative implantation protocol was prepared for both stages. During the first stage, an electric motor would be implanted along with the blood pump and a specially designed dual mode coupling mechanism. During the second stage, the thermal converter would be implanted, engaged to the coupling mechanism, and the electric motor would be removed. This evaluation of the two stage implantation is the first step of a thorough study of implantation protocol to be jointly established by the AEC, University of Utah, and WANL.

The electric motor is designed to fit within the volume ultimately occupied by the thermal converter as shown schematically in Figure 8-1. During the second operation, the motor is designed to rotate out of the way, as shown in the "transition" sketch of Figure 8-1. After connecting the thermal converter to the coupling mechanism, the motor is used to start the thermal converter. After stable operation is achieved with the motor and thermal converter operating in tandem, the entire load is transferred to the converter and the motor is removed.

While the thermal converter is being connected to the coupling mechanism, both the motor and converter must be stopped momentarily to engage the converter output shaft with the coupling mechanism. During this period, an external oil supply must be provided to maintain sufficient oil pressure to prevent damaging the roll sock seals in the converter. The thermal characteristics of the converter are such that the converter may be stopped for up to 25 minutes before reaching operating temperature if it has been allowed to operate at speeds somewhat over design speed for several hours prior to stopping the converter. This is a much larger period of time than the recipient can tolerate a lack of blood flow, so that overheating of the converter does not represent a problem during this period.

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Figure 8-1. Implantation Sequence

During the first stage operation, the coolant lines of the system are filled, and are used to cool the electric motor. These are disconnected from the motor and connected to the thermal converter during the second stage operation. Prior to implantation, an auxiliary coolant loop provides cooling for the converter. The auxiliary coolant loop is disconnected from the converter immediately prior to connecting the converter to the implanted coolant loop. Provisions have been made to break and remake these connections without stopping the electric motor or the converter.

At present, mock-ups of two thermal converters and one blood pump are being prepared. Design work on the electric motor and the coupling mechanism is nearing completion, and, on their completion, mock-ups of these components will be constructed. One of the thermal converter mock-ups and all the remaining models will have functional mechanical connectors. These components can be used by a surgical team to simulate the implantation procedure, and to determine if the procedure, equipment and connectors are adequate for their intended use. The remaining thermal converter mock-up would be used as a gage in the actual, as well as, simulated operations to prepare the implantation site prior to implanting the thermal converter. At present, the auxiliary coolant loop and auxiliary oil supply connections have not been designed, so the connections to this equipment have not been mocked up. These components will be added as they are better defined.

TASK 9 - RELIABILITY ENGINEERING AND QUALITY ASSURANCE

During the second month of the quarter, a formal intermediate design review of the IVBM was held to obtain concurrence on the thermal converter and blood pump configuration prior to commitment to build these components. The results of the design review have been documented and action items are in the process of being closed out. Action items pertinent to reliability included:

- WANL must complete a mission sequence flow diagram or scenario for the set of events which take place at the test site. This is to include all actions and component definitions to support implantation and removal of the IVBM, carefully considering what could go wrong and providing contingency actions in these areas.

From this exercise, it is anticipated that all interface and handling requirements for the IVBM will be identified, particularly as it affects the design.

- The increase in complexity and the potential envelope change for the blood pump prompts the need for reassessment of the hydraulic configuration for the coolant pump.
- The need for identifying cleanliness requirements for all components of the IVBM has been stressed, particularly where entrapment may be a problem and where contamination affects component life.

During this quarter an implantation experiment was terminated due to an apparent flexible shaft failure; the termination occurring approximately 20 minutes after the calf accidentally fell out of the cage. Early failure analysis has shown that the 8" flexible shaft failure did cause the termination of the experiment, but the cause of this shaft failure has not yet been explicitly determined. Vendor analysis indicates an overstress condition occurred. Testing of a like shaft for approximately 200 hours has negated arguments about incorrect shaft selection by the vendor and tends to support the stress overload theory of failure. Efforts are continuing to develop requirements and obtain a short flexible shaft which is less prone to physical damage and more likely to last for the duration of upcoming experiments.

TASK 10 - THERMAL INSULATION FABRICATION

To expedite manufacturing of the IVBM 60-cup insulation packs, the aluminum alloy mandrels are being machined at WANL, and the nickel electroforming will be accomplished at the Westinghouse R&D Center. Efforts are continuing to develop electroplating vendors for producing the nickel foil cups.

TASK 11 - THERMAL CONVERTER FABRICATION

Fabrication of the displacer, heat exchanger, drive, and hot finger for IVBM thermal converter No. 1 was started.

The displacer assembly is nearly completed. The final machining has been temporarily halted pending experiments with closure techniques. Specifically, following displacer pressurization, a capillary tube will be cold welded, to lock the working gas into the displacer. Following the final choice of the cold welding procedure, the machining of the displacer assembly will be resumed.

The first machining, i.e. that done prior to brazing, of the heat exchanger parts has been completed, and the parts sent to the heat processing shop for brazing. It should be noted that most of the machining on the heat exchanger assembly occurs before brazing.

The drive part on which machining was started is the displacer rod, the most complex of the drive elements. Specifically, the oil filtering subassembly which is inserted into the lower part of the yoke was machined and assembled.

The hot finger machining, although initiated, was temporarily halted pending metallurgical investigation of the Udimet 700 bar grain structure. Specifically, after the bar was cut to the required length for the three hot fingers being fabricated, samples were also cut from the regions where the high temperature ends of the hot finger will be, and metallurgical samples prepared to study the grain structure prior to heat treating.

The four samples, one each taken from the center of the bar, 1/2 in., 1 in., and 1-1/4 in. from the center, indicated a structure characteristic of cast rather than wrought materials. The samples were therefore subjected to heat treatment; specifically, they were annealed

at 1100°C for 5 hours in argon. The results showed that the structure was going into solution, i.e., that grain boundaries were forming. Metallographs of the samples before and after heat treatment are shown in Figures 11-1 through 11-8. As a result of the study discussed above, the machining of the hot fingers is being resumed.

Figure 11-1. Metallograph of Sample Taken $\sim 1 \frac{1}{4}$ Inch
from Center of Bar As Received

Figure 11-2. Metallograph of Sample Taken $\sim 1 \frac{1}{4}$ Inch
from Center of Bar After Annealing

Figure 11-3. Metallograph of Sample Taken ~1 Inch from Center of Bar As Received

Figure 11-4. Metallograph of Sample Taken ~1 Inch from Center of Bar After Annealing

Figure 11-5. Metallograph of Sample Taken $\sim 1/2$ Inch from Center of Bar As Received

Figure 11-6. Metallograph of Sample Taken $\sim 1/2$ Inch from Center of Bar After Annealing

Figure 11-7. Metallograph of Sample Taken at Center of Bar
As Received

Figure 11-8. Metallograph of Sample Taken at Center of Bar
After Annealing

TASK 14 - PERFORMANCE TESTING

The two Mock Circulation units have been completed. One unit was shipped to the University of Utah along with an instruction book of calibration procedures.

The Mock Circulation unit (MC) using Ramapo flowmeters was adjusted to provide specified compliances. Next, the flow rate was calibrated using a continuous flow. The bellows were adjusted to provide near ideal systemic and pulmonary pressures. After calibration, the blood pump was connected in series with the MC unit and flow data were recorded. One of the MC units is shown in Figure 14-1 and a schematic is shown in Figure 14-2.

Figure 14-1. Mock Circulation - Front View

COMPLIANCES

SYS. V.	10 ml/mmHg
PUL. A.	1 ml/mmHg
PUL. V.	5 (3-1/2) ml/mmHg
AORTA	1 ml/mmHg

Figure 14-2. Mock Circulation Schematic