Advanced Energy Systems Division

NUCLEAR-POWERED ARTIFICIAL HEART PROTOTYPE SYSTEM DEVELOPMENT PROGRAM

QUARTERLY PROGRESS REPORT

April 1, 1976

Through

June 30, 1976

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NUCLEAR-POWERED ARTIFICIAL HEART
PROTOTYPE SYSTEM DEVELOPMENT PROGRAM

PHASE III
QUARTERLY PROGRESS REPORT

APRIL 1, 1976 THROUGH JUNE 30, 1976

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INTRODUCTION AND SUMMARY

The Nuclear-Powered Artificial Heart Prototype System Development Program is being conducted by the Westinghouse Advanced Energy Systems Division and its major subcontractor, the Philips Laboratories division of the North American Philips Corporation, as Phase III of the U. S. ERDA's Artificial Heart Program under Contract No. E(11-1)-3043. This report presents the technical progress and accomplishments on the active program tasks in numerical order for the period from April 1, 1976 through June 30, 1976. A better understanding of the overall program may possibly be obtained by reading the activities covered in Tasks 39 through 41 in advance of the research and development Tasks 35 through 38.

Under Task 35, Thermal Insulation and Thermal Protection Research and Development, the major accomplishments were the initiation of an evaluation of the emittance of vendor supplied cups as a function of heat treatment, the completion of the shock and vibration testing of the 26 cup insulation package, the start of outgassing of the 60 cup static vacuum insulation system without getters, and continued evaluation of foam polyethylene insulation for the TOPS coolant system.

Thermal converter development continued at Philips Laboratories under Task 36. Significant accomplishments were the improvement of Udimet 700 grain structure by modification of the previous heat treatment schedule, the initiation of an investigation into the feasibility of cold forming the thin wall hot finger, successful testing of a single fluid output shaft magnetic seal, resumption of development work on an alternate screw seal as a backup design, completion of design of a MARK I prototype magnetic coupling, the determination of eddycurrent and windage losses of the coupling, and the successful operational testing of one of the two new control valves.

Flexible shaft, bearing and lubricant development continued under Task 37. Life testing of wound-wire shafts continued but testing of the solid titanium shaft was terminated following failure. Ball bearing testing also continued through the period and one failure was recorded. The failed bearing is not typical in configuration or loading of the MARK I Prototype System

blood pump bearings. Use of dibutyl phosphite as an additive was proved very effective with all candidate oils in reducing wear but was determined to be incompatible with silastic.

Only limited testing was conducted under Task 38, Thermal and Hydraulic Research and Development, initially, because of failures of blood pump flexible coolant tubes and later a linear bushing failure. The failed tubes were of all metal design and a replacement design, part metal and part plastic, appears to have solved the short term tube failure problem. The linear bushing failure is under investigation.

Task 39, System Integration and Interfacing Research and Development, encompasses the design, analysis, and procedures required to ensure an integrated system capable of meeting the design objectives. A detailed assembly and sterilization procedure for field assembly of the converter was prepared and the corresponding test plans were updated. Design was initiated for an implantable electrically heated static vacuum insulation package version of the IVBM converter. The design specification for the MARK I Prototype System was prepared and is included as Appendix A of this report. A study of off-design performance of the thermal converter was performed. A modified IVBM blood pump with a decreased right ventricle stroke volume for use with redesigned ventricles was supplied to the University of Utah. Development of corrosion prevention coatings for magnesium blood pump housings continued.

Under Task 40, IVBM System Performance Testing, component testing of both thermal converters was continued and integrated system testing was carried out. The system, as the components are presently matched, produces about 9 l/min blood flow at 101 mm Hg mean aortic pressure and does not stall at 167 mm Hg mean aortic pressure. System testing was suspended due to the failure of the outer clad of both Pu-238 heat sources. The heat sources were returned to LASL. The heat source operating history is presented in Appendix B of this report.

Task 41 covers the fabrication of the IVBM system components. The static vacuum insulation system, IVBM No. 2, was restored to the original insulation performance level. A silicone

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rubber covering for the flexible shaft casing was designed and constructed. The dynamically pumped vacuum insulation system, IVBM No. 1, was dismantled to provide parts for the fabrication of a static vacuum electrically heated system for which heater concepts are under investigation.

The three additional blood pumps, the subject of Task 42, were completed and two were shipped to the University of Utah. The electric drive motors and controllers are being assembled in the implantation configuration.

Life testing of the IVBM blood pump under Task 43 was continued in conjunction with the Task 38 work until interrupted by the failures reported under Task 38.

Quality Assurance activities, covered by Task 46, continued at a level consistent with other program activities.

TASK 35 - THERMAL INSULATION AND THERMAL PROTECTION RESEARCH AND DEVELOPMENT

Work accomplished during the quarter included: 1) beginning of emittance evaluation of vendor supplied cups as a function of heat treatment; 2) completion of the shock and vibration testing program on the 26 cup insulation package; 3) start of outgassing of the static vacuum package number one, a 60 cup set with no getters; 4) continued characterization of the as-deposited electro-formed nickel.

EMITTANCE EVALUATION

The cups described in Table 35-1 were received from various vendors for emittance evaluation and metallurgical characterization. The emittance measuring technique was described previously*. Equilibrium results obtained thus far are listed in Table 35-2.

As the program continues the reasons for the differences between the values of emittance measured by using the optical pyrometer and those calculated will be investigated to determine if the differences are due to some experimental variable or are characteristic of the material being tested. Based on the temperature gradients measured on the outer cup it is suspected that some heat input from the guard heater is increasing the temperature of the outer cup resulting in a lower than expected emittance. These results will be verified after some adjustments to the experimental system.

SHOCK AND VIBRATION ANALYSIS

The 26 cup test insulation package with skirted cups was subjected to the shock and vibration history outlined in Table 35-3. Four 8 hour vibration tests were made. Figure 35-1 shows the vibration test geometric relationships. With the axial plane of the engine cylinder in the horizontal position, the system was vibrated in both the axial and radial directions. The axial plane of the engine was then positioned vertically, and the system was

* Westinghouse Astronuclear Laboratory, "Nuclear Powered Artificial Heart Prototype System Development Program, Quarterly Progress Report COO-3043-18, September - December, 1974," ERDA Contract E(11-1)-3043.

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TABLE 35-1
LIST OF VENDOR SUPPLIED CUPS FOR EVALUATION

Vendor	Manufacturing Process	No. of Cups 1.040 in. dia.
Gar	Sulfamate nickel electroform Activation cleaned mandrel (R&D process)	2
Servometer	Sulfamate nickel Servometer process cleaning	2
Westinghouse Research Labs	Sulfamate nickel Activation cleaning	2
Enthone	Electroless nickel deposit	1
Westinghouse Research Labs	0.7 mil sulfamate nickel plated on 0.1 mil bright nickel	2

TABLE 35-2
EMITTANCE RESULTS ON SERVOMETER CUP SERV-2-E

Outer Surface	Outer Cup Temp. (°K)	IRCON 300 Pyrometer (2.0-2.6 μ)	Calculated Total Hemispherical
As-received	498	.26	.14
	525	.23	.11

TABLE 35-3

REVISED TEST PROGRAM FOR TESTING HEART
PROGRAM THERMAL INSULATION

- 1.0 Mount test specimen to single axis vibration table. Testing will be performed on two axes in both the horizontal and vertical positions.
- 2.0 Connect all external connections (vacuum, power, etc.).
- 3.0 Conduct a low level sweep test (0.2 g) from 4 Hz to 12 Hz with a duration of approximately 5 minutes to detect any resonances.
- 4.0 After analyzing the data from resonance search, a sweep test will be performed from 4 Hz to 12 Hz for a duration of 8 hours. Sweep time 4 Hz to 12 Hz approximately 5 minutes. Vibration level to be 5 g pk.
- 5.0 Rotate test specimen 90 degrees and repeat sections 3.0 and 4.0.
- 6.0 Move specimen to the vertical test position and perform sections 3.0 and 4.0.
- 7.0 A shock test consisting of two shocks with a half sine wave envelope 0.1 sec. width and 8 g peak amplitude will be performed in the vertical direction and two at 4 g in each of the horizontal directions. (Testing to be performed on hydraulic biaxial test table.)
- 8.0 The final test to be a 30 g shock in the vertical direction at a pulse width of 0.017 sec.
- 9.0 During testing, T/C's will be recorded continuously. One accelerometer will be mounted on specimen for control and two others for response measurements. Accelerometers will be continuously recorded during testing program.

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VIBRATION DIRECTIONS IN HORIZONTAL POSITION

VIBRATION DIRECTIONS IN VERTICAL POSITION

Figure 35-1. Schematic of Vibration Direction - System Orientation

vibrated in both the axial and radial directions. Figures 35-2 and 35-3 are photographs of these horizontal and vertical experimental systems, respectively.

In addition to the vibration testing, the system was subjected to a biaxial shock loading and will be subjected to a 30 g, 0.017 sec pulse as illustrated on Figure 35-4. The biaxial shock test was done on a hydraulically activated system. Figure 35-5 shows the biaxial shock test configuration. Although the input pulse to the hydraulic activators was as specified, the acceleration experienced by the insulation package was different. However, the actual acceleration was recorded by a shock spectrum analyzer to determine the magnitude and severity of the actual shock loading.

The time-temperature history of the 26 cup insulation package during the shock and vibration testing is shown in Figure 35-6. Power is read on the right ordinate, and temperature is measured on the left ordinate. Until the insulation system can be disassembled and the cups and spacers examined visually, the only measure of the survivability of the package during shock and vibration is the thermal performance. The four vibration exposures and the biaxial shock exposure has not impaired the thermal performance of the system. The ability to maintain 1420°F with 14.1 watts power input had not changed from the beginning of the testing. These results are in line with the power and temperature data during the four month life test in the horizontal position (15.9 watts at 1525°F). All of the shock testing was done at temperature with the multifoil cavity in the 10^{-6} torr pressure range.

When the 30 g shock test is completed, this system will be disassembled and subjected to a detailed visual examination of cup integrity and spacer adherence.

STATIC VACUUM SYSTEM

The 60 cup system with no getters is being outgassed. A pinchoff will be utilized on the pumpout side of the high vacuum valve to eliminate any possibility of valve through leakage contributing to the gas load during the static performance test periods.

Figure 35-2. Photograph of Horizontal Vibration Test Configuration

Figure 35-3. Photograph of Vertical Vibration Test Configuration

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A) SHOCK TEST NO. 1 - BIAXIAL TEST

B) SHOCK TEST NO. 2 - 30 g VERTICAL SHOCK

Figure 35-4. Schematic of Shock Direction in Relation to System Orientation

Figure 35-5. Biaxial Shock Test Configuration

35-10

Figure 35-6. Thermal History During Shock and Vibration
(26 Cup Set with Powdered ZrO_2 Spacers)

MASS SPECTROMETER DATA SYSTEM

A revised program is planned for the data system to increase the system versatility. The ability to repeatedly scan a given peak such as helium will enhance the detection limit of helium and also enhance differences between background and helium possibly resulting from a leak during leak detection operations. This will increase the signal to noise ratio and eliminate statistical ion detection fluctuations.

THERMAL PROTECTION

Testing continued on the effort to determine whether foam polyethylene insulation will function adequately at the 150°F operating temperature of the developmental methanol circulated TOPS coolant system. To date, over three months of testing has been completed under conditions which simulate TOPS in-vivo operation without any sign of insulation deterioration. The average heat flux measured at the outside insulation surface has remained unchanged at 0.008 watts/cm².

TASK 36 - THERMAL CONVERTER RESEARCH AND DEVELOPMENT

HOT FINGER DEVELOPMENT

Heat treatment procedures for Udimet 700 were investigated again after it had been found that the old heat treatment (1100°C - 5 hr. air cool; 870°C - 5 hr. air cool; 982°C - 4 hr. air cool) often resulted in grains larger than the nominal thin-wall thickness of 0.004 inch. A typical example of Udimet 700 subjected to the old heat treatment is shown in Figure 36-1.

As a development goal, it is assumed that the alloy structure is at an optimum if on one hand the gamma-prime phase is dissolved and reprecipitated in a finely distributed form within the grain, while the carbides and borides are dissolved and reprecipitated at the grain boundary to improve creep resistance. This process requires a rather high annealing temperature resulting in a large grain size. On the other hand, a small grain size ($1/5$ - $1/10$ of the thin wall) is desirable to obtain an interlocking grain structure in which creep resistance is improved by the interlocking of the grains. However, the grain structure should not be excessively small because very small grain size can lead to super plasticity, a very undesirable effect. To obtain the interlocking grain structure requires rather low annealing temperatures. It is felt that a compromise has been found between precipitation of gamma-prime and carbides on one hand and the interlocking grain structure on the other hand by the new heat treatment (1050°C - 10 hrs. air cool; 870°C - 5 hr. air cool).

The new heat treatment results in a grain structure as shown in Figure 36-2. Test specimens will be prepared to test the creep rupture life of material subjected to the new heat treatment.

A Udimet 700 thin-walled creep rupture specimen was prepared from as-received stock without heat treatment. The data from this specimen will provide a baseline to judge the improvement made by heat treating. The above discussed specimen is the first made to Mark I

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100 X

Figure 36-1. Udimet 700 After "Old"
Philips Heat Treatment.

100 X

Figure 36-2. Udimet 700 After "New"
Philips Heat Treatment.

prototype specifications, which will be used for all further specimens. A new spindle for the air-gauge was received to permit accurate measurement of the thickness of the test cylinder wall. The test cylinder from as-received stock has been inspected and measured and is ready for test.

Contacts were made with Rollmet Co. in Irvine, California, a specialist for cold forming thin-walled seamless tubing. Udimet 700 was supplied to Rollmet Co. for the fabrication of samples of open-ended tubing.

During a visit to Philips Laboratories by Mr. L. F. Glasier, Jr., Manager of Engineering of Rollmet Co., the possible benefits of their process were discussed. It is possible to produce closed end cylinders to the dimensions of the Mark I prototype, thus eliminating any brazing in the thin-wall region. The cold rolling process produces thin-wall cylinders with an extremely uniform wall thickness (± 0.0002 in.) both circumferentially and longitudinally. In addition, the grain structure will be uniform and the grain size will be very small. The surface will have a smooth finish which will eliminate the need for polishing and will be beneficial for non-destructive testing, and above all, for improved creep rupture strength. The process can be used to produce even thinner walls, as needed for the regenerator liner. Ultimately this method of thin-wall fabrication may lend itself to mass production of hot fingers and reduction of their associated manufacturing costs.

The question of non-destructive testing was studied. Three methods were selected for further consideration.

- The application of "Stresscoat". After pressurization, a regular crack pattern will appear in the coating. Any deviation from the regular crack pattern will indicate an inhomogeneity. Drawbacks are the fact that the resolution might not be good enough and an uneven coating might obscure a true inhomogeneity.

- The application of photoelastic material. After pressurization, a regular fringe pattern that can be observed with polarized light appears in the coating. This method has the same advantages and disadvantages as the above method except the resolution might be better.
- The application of an interference method. The interference pattern before and after pressurization will be registered. Any deviations from the regular pattern can be associated with inhomogeneities. The disadvantage is the need of a polished surface. However, it might be possible to use the as-rolled surface of the cold rolled cylinders without polishing. This would ultimately make the interference method superior to the other two.

STRUCTURAL INVESTIGATIONS

Work during the report period was devoted primarily toward the aluminum bronze to titanium joint, development of techniques for welding mesh structures for fabrication of filter assemblies, and hard-coating of titanium.

Final and confirmed procedures for the bronze to titanium joints have yet to be completed. Although some good joints have been made (see Figure 36-3), the confidence level of achieving sound joints on a repeatable basis is not yet satisfactory. The joining techniques used involved the use of tapered pins to constrain movement of the bronze upon cooling. An improvement resulted when the diameter of the taper sections was increased. However, there is evidence that some stress relief cracking occurs long after the brazing operation. Further test samples are in process to evaluate refinements to the stress relief and constraint techniques.

Investigations into hardening the journal surface of the titanium alloy displacer rod are still in process. Initially, emphasis was placed on a proprietary commercial process called "Canadizing" (General Magnaplate, Inc.), which is a form of anodize treatment. The vendor recommended the use of 6 Al-4V titanium alloy for the parts, with an allowance of

600X

Figure 36-3. Titanium Alloy to Bronze Braze Joint

1.5 mils for radial buildup. Samples were prepared and delivered to the vendor. After two months, and many attempts, the vendor abandoned the effort and recommended the use of electroless nickel instead. At this point, the effort was picked up by the Philips Laboratories heat processing laboratory.

An investigation of plating electroless nickel of appropriate phosphorous content onto titanium is in progress. Electroless nickel plates have been successfully bonded to the alloy Ti 5 Al 2.5 Sn. Subsequent baking to give maximum hardness (R_c 70) of the plate was quite successful and no peeling or stripping of the nickel plate followed the hardening procedure. Attempts to plate electroless nickel on the alloy Ti 6 Al 4V (used in the actual displacer rods) have not as yet been successful. Poor coatings of incomplete coverage seem to result from incomplete activation of the titanium surface. Work on this plating problem continues.

Samples of electroless nickel deposited on the internal wall of a Udimet 700 cylinder have been prepared. This structure is needed for the displacer seal riding surface. Heat treatment to give a hard wear surface did not destroy the nickel-Udimet bond. Machinability and wear tests have yet to be performed.

Techniques are well in hand to edge weld stainless steel wire mesh for the fabrication of 2-micron, pleated screen filters. The plasma needle arc is particularly good for welding very thin materials. Figures 36-4 the seam weld and end triple weld can be seen. Figure 36-5 shows the seam weld in greater detail. Fabrication of some complete filter units is presently in process.

ROTARY SHAFT SEALS DEVELOPMENT

Ferrofluid Investigation

In an effort to develop long lasting ferrofluid seals, an extensive literature survey was made to develop more information about ferrofluids and, in particular, factors affecting stability.

Figure 36-4. Oil Filter Assembly Weld Specimen

Figure 36-5. Seam Weld of Mesh Pleat

It was found in a theoretical and experimental study for the Office of Naval Research (Contract N00014-67-A-0299-0020) conducted by Colorado State University in 1975 that magnetic fluid agglomeration will not take place if particle size is small enough. In fact, for particles with a coating thickness of about $30 \text{ \AA}$ and an overall diameter of $154 \text{ \AA}$ or less, agglomeration will not occur even in an infinitely strong magnetic field. With this in mind, previously ordered Pella 11 ferrofluid was procured with the smallest available particles, which are $100 \text{ \AA}$. The same $100 \text{ \AA}$ particle size had been used previously to make the Tellus 15 ferrofluid which has been on successful life test for 1700 hours in the two-pole shaft seal. Figure 36-7 in the previous quarterly report* shows an SEM photograph of this fluid. Typical coated particle size appears to be $150 \text{ \AA}$ or at about the critical limit; however, rigorous particle size measurements were not conducted at that time.

In reality, a ferrofluid contains a particle size distribution whereby the larger particles will tend to agglomerate first. Magnetic strength of the fluid diminishes by approximately the percentage of agglomeration. One might therefore conclude that if a given fluid is not sufficiently stable, strict quality control of particle size would improve this condition. At the present time, stability of the Tellus 15 ferrofluid over a duration of 1700 hours is excellent. The two-pole shaft seal is on life test at 15 psi differential pressure. The seal capability is typically 17 psi. Therefore, if any significant agglomeration has occurred, the seal would have already failed. In the literature, it is estimated that hydrocarbon base ferrofluids have typically 90 percent or more of their particles below the critical size limit. It is planned to take additional SEM photographs of the new Pella 11 ferrofluid which is the anticipated fluid for use in the Mark I engine. It is hoped to accurately obtain the fluid's particle size distribution by analysis of the photos. Also, the literature suggests another encouraging point; namely, that increased ferrofluid stability may result in rotary shaft seals where the applied field is radial (as in the test seals).

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

A quart sample of Pella 11-base ferrofluid was received this quarter. Pella 11 oil is similar in composition to low viscosity vitrea oils for rollsock compatibility. The fluid was ordered as mentioned with the smallest particle size available (nominally $100 \text{ \AA}$) with a viscosity that is higher than required at 120°F . Viscosity is easily reduced by the addition of base oil, Pella 11; however, its magnetic strength is reduced as well.

The as-received Pella 11 ferrofluid had a viscosity of 115 cp at room temperature and a magnetic strength of 230 gauss. It was diluted with Pella 11 oil until it met the engine lubrication specification of 33 cp at 120°F . Its magnetic strength reduced to 210 gauss. Figure 36-6 shows the test data taken of ferrofluid viscosity versus temperature. A Brooks viscometer was used for viscosity measurements. The base oil curve is supplied for comparison. Obtaining this data was very important since proper lubricant viscosity at the maximum operating temperature is crucial for engine performance and the resulting magnetic strength influences the shaft seal pressure capability. This 210 gauss fluid exceeds the 180 gauss Tellus 15 ferrofluid used in recent seal tests. An associated improvement in seal capability is discussed later in this report.

A test apparatus called a magnetometer, shown in Figure 36-7, was made prior to obtaining the Pella 11 ferrofluid. It can be used to determine the magnetic strength of diluted ferrofluids. Horseshoe shaped alnico magnets are located at each end of the structure. Their fields are placed in series to create a high diametral field through the cylindrical plastic container seen at the top. The magnets are wrapped with four thousand turns of wire forming a coil and the leads are attached to an integrating voltmeter. Introduction of ferrofluid into the non-magnetic cup results in a change in magnetic circuit reluctance, thereby inducing a voltage in the coil. The steady state integrated voltage reading is independent of the rate of ferrofluid injection and is related to the magnetic saturation of the ferrofluid (provided the field significantly exceeds the ferrofluid's strength). The calibration curve for the device was obtained by introducing various known strength ferrofluids. As the original Pella 11 ferrofluid was reduced in viscosity, its magnetic strength was determined using this device.

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Figure 36-6. Temperature-Viscosity Data for Pella #11 Magnetic Fluid

MAGNETOMETER CALIBRATION

Figure 36-7. Magnetometer Photograph and Calibration Curve

In theory, the magnetic strength is also related to the volumetric concentration of iron. The following formula was shown to be valid based on magnetometer measurements.

$$M_2 = M_1 \left(\frac{V_1}{V_1 + V_2} \right) \quad (1)$$

M_1 = original ferrofluid strength

M_2 = resulting ferrofluid strength

V_1 = original ferrofluid volume

V_2 = added volume of base oil

As mentioned, a Brooks viscometer was used to determine ferrofluid viscosity versus temperature and this determined the amount of base fluid to add for dilution. Unfortunately, the literature search did not provide information regarding viscosity dependence on shear rate, i.e., whether the particle-laden fluid is Newtonian at high rates of shear. The Brooks viscometer measures viscosity only at very low shear rates, about 10 sec^{-1} ; however, shear rate in the main engine bearings is about $80,000 \text{ sec}^{-1}$, and in the flywheel bearing it is about $200,000 \text{ sec}^{-1}$. Consequently, the shaft seal fixture was modified to accept a 0.160 inch diameter shaft that fit with a 1 mil radial clearance in a bearing in order to obtain shear rate data. A sensitive torsion spring dynamometer was used for measuring torque. Figure 36-8 is the data obtained showing net drive torque versus rate of shear. The torque is corrected for ball bearing friction which was obtained without oil in the bearing. Tellus 15 base oil was run as a control; its data is also shown in the figure.

A pure Newtonian fluid possesses a constant viscosity and would be represented by a straight line in Figure 36-8 that passes through the origin. For the limited data shown in the figure, this certainly appears to be the case for the ferrofluid as well as base Tellus oil. Pella 11 oil was not run as a control because its viscosity was too low for the torsion spring used. A new test shaft has just been completed. It has a reduced 0.25 mil radial bearing clearance which will extend the data out to a shear rate of $320,000 \text{ sec}^{-1}$. The data in Figure 36-8 confirms

Figure 36-8. Shear Rate Data for Pella #11 Magnetic Fluid

that the present 210 gauss Pella 11 ferrofluid will have the required viscosity in the main bearings and will also produce the anticipated crankcase churning losses. If the extended range data, yet to be taken, shows a sub-Newtonian behavior, the flywheel bearing could be modified to run at the lower viscosity level obtained in the shear rate tests.

Other, no less demanding, requirements of the ferrofluid are that it be able to be filtered and not cause premature gear pump or bearing wear. A mock oil filter and gear pump lubrication loop life test was started for determining the following. First, whether ferrofluid can be filtered using the contemplated 2 micron mesh screening and second, whether ferrofluid particles cause oil pump wear. A closed circulating system was constructed whereby an old IVBM hardened-gear pump discharged 60 psi oil into a 0.75 inch diameter filter element, then through the orifice and finally, back to the pump reservoir. A differential pressure gauge records the pressure across the filter. This pressure drop was identical (3 psi) for Tel-lus 25 oil and uncut Pella 11 ferrofluid. The system has been running in excess of 400 hours and shows excellent stability, i.e., oil pump output pressure and filter pressure drop have not changed.

Because the cross sectional area of the filter is 16 times less than that of the Mark I filter and the pump flowrate is $1/3$ that of the Mark I pump at 600 rpm, the life test speed-up factor is about 5 to 1. In other words, five times as much fluid is passed through each unit area of the test filter per hour, thereby accelerating the clogging effect of ferrofluid particles. The equivalent Mark I demonstrated life is therefore 2,000 hours to date. Because these test results are positive, work has begun on the design of a ferrofluid lubricated test bearing which can be run under load. This is the anticipated final-step in proving the applicability of ferrofluid as a satisfactory bearing lubricant. There has been no evidence in the literature that ferrofluid particles cause wear. Pending successful completion of this phase, we will be in a position to use the ferrolubricant in the rhombic drive test fixture for final testing.

To vie as a successful engine lubricant, the ferrofluid must also demonstrate rollsock compatibility for the Mark I engine. Rollsock compatibility tests were initiated this quarter and are still in progress. Rollsock samples were immersed at 120°F in Tellus 25 oil as the control and Tellus 15 ferrofluid. The containers used were sealed glass vials. Moisture was removed beforehand by heating the oil samples to 150°F for several hours; sample elongation under tension as well as sample weight were recorded. After 1000 hours of immersion, the data indicates that the ferrofluid samples performed slightly better than the Tellus 25 samples. This is attributed to the lack of boundary and anti-foaming additives in the Pella 11 base oil. Testing is still in progress and additional tests are planned where better control will be maintained over the moisture content of the oil.

With the ending of this quarter, ferrofluid miscibility tests are being discontinued. The original purpose of these tests was to make a ferrofluidic seal for sealing against another liquid. All attempts have failed. However, an understanding of the reason for failure even with immiscible systems is believed to be the following:

Although the total magnetic force on a volume of ferrofluid may be great due to a large mass of fluid, such as exists in a seal, the force on each individual particle is miniscule. Turbulent eddies and viscous boundary layer instabilities at the liquid-liquid interface produce forces orders of magnitude greater than magnetic body forces. Particles of ferrofluid diffuse, are dragged away, and cannot be held from escaping the magnetic field, even under non-rotating shaft conditions. Many of the same tests performed at Philips Laboratories on ferrofluid plugs in vertical glass columns were also performed in the Colorado State University study previously cited. Their modes of failure and basic results obtained were identical. For purposes of completeness, the following additional systems were tested without success for use as possible liquid to liquid seals during this quarter:

- Sperm oil with ethylene glycol F. F. (Ferrofluid)
- Sperm oil with silicone oil F. F.
- Silicone oil with Apiezon C F. F.

- Tri-ethyl phosphate with Tellus 15 F. F.
- Tri-ethyl phosphate with ethylene glycol F. F.

Shaft Seals Development

The tapered pole face geometry first presented in the January-March 1976 Progress Report* was thoroughly tested this quarter. Figure 36-9 shows the geometry of the seal and its test data. The ferrofluid crankcase oil is located to the left. The 60° pole face taper is used to keep the iron from magnetically saturating except very close to the air gap. An air gap field of 47,000 gauss has been achieved. This is calculated based on the pressure capability of the seal which is 17 psi (8.5 psi per stage) using 180 gauss ferrofluid. The Mark I flywheel shaft will be 4340 carbon steel. For comparison, test data was also taken for 1018 steel which is magnetically superior. As the data shows, there is actually improved performance using the 4340 steel shaft. We have recently prepared Pella 11 ferrofluid to engine use specifications. Its magnetic saturation is 210 gauss as compared to the 180 gauss fluid used in the tests. Therefore, the seal pressure would be expected to increase to 20 psi, or 10 psi per pole. This indeed has been the case.

The data in Figure 36-9 shows seal pressure capability as a function of the number of times the seal is blown past. If the crankcase is pressurized through the seal (1 blowby) the seal will then sustain about 80 percent of its initial capability. This decrease in performance is attributed to loss of fluid in the right hand pole face.

Life testing of this seal using Tellus 15 magnetic fluid was continued during most of the quarter. A constant pressure of 16 psid was imposed while the shaft was rotating at 2400 rpm. Sealing has been hermetic for the accumulated time of 1700 hours. As pointed out for the diagram of Figure 36-9, crankcase oil is sealed on the left end. The air pressure, internal to the seal between its pole faces according to magnetic theory, must always exceed the crankcase pressure for the left pole face to seal. In the Mark I engine the crankcase pressure

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

Figure 36-9. Two-Pole Shaft Seal Pressure Capability

will vary sinusoidally with a ± 7 psi differential. The left pole face acts like a check valve with respect to the gas atmosphere. Therefore, the internal zone of the seal will charge up to the maximum crankcase pressure of +7 psi upon start of the cycle. At the peak crankcase pressure, the right pole face will experience a ΔP of +7 psi, whereas its capability as mentioned is 10 psi. When the crankcase pressure decreases, the inter-pole-face gas pressure reduces until zero psi is reached and stays at this value as the cycle reaches -7 psi and again goes up to zero. As the cycle pressure once again goes positive, ferrofluid would be expected to push back into the left pole face, thereby sealing it and completing the cycle. Over a cycle, the right pole face thus experiences pressures of +7 psi down to 0 psi and should never blow by. Since the left pole face is always in communication with crankcase ferrofluid, its fluid is not diminished after blowing by ten times per second and its successive pressure capabilities should not be impaired. Test data to date verifies this concept which is discussed in more detail below.

In the above example, the flywheel housing was charged to the same mean pressure as the crankcase. This resulted in the right pole face sustaining up to +7 psi or 70 percent of its capability. This leaves a small safety factor of 1.43. However, note that by pressurizing the flywheel housing to 3.5 psi above the crankcase pressure, the right hand seal will experience only +3.5 and -3.5 psi; the minimum possible variation, as the chart below shows.

Flywheel Housing Charge Pressure (Above Crankcase Mean)	ΔP Across Right Pole Face	
	(+ Cycle)	(- Cycle)
0 (above example)	+7	0
1	+6	-1
2	+5	-2
3	+4	-3
3.5 (optimum)	+3.5	-3.5
4	+3	-4
5	+2	-5

The seal safety factor has increased under these conditions to 2.86 which should provide high reliability and good performance for even off-nominal conditions.

It is considered a milestone that the fluidic seal functions in the manner discussed above, since up to now seals sustaining high cyclic rate gas blowbys have not been successful. The good operation obtained with the test seal to date is due to complete submersion of the left pole face in magnetic fluid. Four eight-hour tests without shaft rotation have been run to date, and a full duration life test will commence shortly.

Figure 36-10 is an oscilloscope trace of the pressure waveform imposed on the seal during testing. The zero level is atmospheric pressure. Consequently, the right side was charged to +3 psi during the 8-hour tests. Even without the auxiliary 3 psi input, seal function was unimpaired.

The power consumption of the two-pole seal was measured to be about 10 mw using the spring dynamometer shown in Figure 36-7 of the October-December 1975 Progress Report*. Half of this loss was the viscous oil loss as eddycurrent losses were only about 6 mw due to the high resistivity of the 4340 steel shaft. Data was also obtained this quarter for a four-pole seal configuration. The results are shown in Figure 36-11. Note that the maximum pressure capability of the first blowby is about the same as the two pole design. This seal is not contemplated for use due to its added complexity. Furthermore, its only advantage over the two pole design is a factor of two reduction in the air gap magnetic field. This could promote ferrofluid stability. Nevertheless, unless there is a problem with the stability of the present ferrofluid, use of this seal is not contemplated.

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Figure 36-10. Sinusoidal Pressure Input Waveform to 2-Pole Shaft Seal

Figure 36-11. Four-Pole Shaft Seal Pressure Capability

A diaphragm type air compressor was purchased and fitted with a 600 rpm drive motor for the Figure 36-10 tests. It is presently being refitted with a larger, more reliable motor for use in testing the second ferrofluid seal for a long duration. The present life-test 2-pole seal will continue to run using a constant pressure input in order to test the long term stability of the ferrofluid in the seal's high magnetic field of 47,000 gauss.

Much effort has gone into developing a magnetic fluid shaft seal for the Mark I engine because of the following advantages:

1. Hermetic sealing with negligible power loss.
2. Possible use of a low molecular weight gas in the flywheel housing to reduce windage losses over that of argon.
3. Sealing at zero speed (a significant advantage for testing and implantation).
4. Use of ferrous lubricant may permit use of a minimally filled crankcase, saving a substantial portion of the 500 mw oil churning losses.

If ferrofluid cannot be used in the blood pump, another kind of shaft seal must be used for sealing the right angle gearbox oil out of the magnetic coupling armature housing. A ferrofluid engine seal cannot be employed if ferrofluid is found not to be acceptable as a crankcase lubricant. For these reasons, interest in a screw viscosity seal was revived. Previously, it was not thought possible to make a low loss, reliable screw seal. However, new design equations were formulated when it was found that the design equations formerly used were inadequate. The new analysis has led to the development of an extremely short seal, 0.125 inches long, which dissipates less than 20 mw of power at 2400 rpm. Initial prototype tests have demonstrated the leak-free character of these dynamic seals.

Figure 36-12 is a photograph of the test screw seal. There are 1.2 land threads which overlap to form only 0.2 thread grooves. The radial clearance of the lands with the housing is 0.4 mils and is made relatively large so that little viscous power is dissipated and a significant

Figure 36-12. First Prototype Screw Seal

clearance for shaft runout is provided. The maximum pressure held with this seal was 13.5 psi at 2400 rpm using 48 cp oil. This seal was fabricated without the aid of the new analysis. Its pressure capability is not sufficient for the contemplated 33 cp oil at 120°F. Since the derivation of the seal formula is quite detailed, only the results are presented. A simplified model where one dimensional flow in the thread groove is assumed was used.

Figure 36-13 shows a screw seal with parameters labeled. The volumetric rate of flow Q (in³/sec) for a centered screw of N grooves with $\ell \ll h$ is:

$$\frac{Q}{\cos^2 \phi} = \frac{\omega D}{4} h \ell \left[1 + \frac{h_o}{h} \frac{\ell}{\ell} - \pi D \frac{\sin \phi}{\cos^3 \phi} \left(\frac{1}{\ell} \frac{h_o}{h} + \frac{2}{\pi D \sin \phi} \right) \right] \quad (2)$$

$$- \Delta P \frac{h^3 \ell}{12 \mu \pi D} \left[\frac{1}{N} + \frac{1}{(N+1)} \left(\frac{h_o}{h} \right)^3 \frac{\ell}{\ell} \left(\frac{\pi^2 D^2}{\ell^2 \cos^4 \phi} + 1 \right) \right]$$

$$\text{or } Q = C_1 - C_2 \Delta P. \quad (3)$$

Figure 36-14 is the experimental data of flow versus pressure for the prototype seal.

It is a straight line relationship of negative slope as equation (2) predicts. For the following screw values,

$h = 0.0031$	$N = 0.2$ grooves
$h_o = 0.0004$	$N + 1 = 1.2$ lands
$\ell = 0.0583$	$\mu = 7.71 \times 10^{-6} \frac{\text{lb-sec}}{\text{in}^2}$ viscosity
$\ell_o = 0.0250$	$\omega = 251$ rad/sec
$D = 0.1560$	$\phi = 9.6^\circ$ or 12 thds/in

The flow rate at zero pressure is predicted to be 1.75×10^{-3} in³/sec. This is in good agreement with an experimental value of 1.5×10^{-3} seen in Figure 36-14 at the ordinate intercept.

Q was also measured to be independent of viscosity as expected.

Figure 36-13. Screw Seal Geometry

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Figure 36-14. Prototype Screw Seal Data

When used as a seal rather than as a pump, the maximum sustained screw pressure is of most importance. When $Q = 0$ in equation (2), the maximum ΔP of the seal is determined. The data in Figure 36-14 shows $\Delta P_{\max}$ to be experimentally 13.5 psi when using 48 cp oil. The effect of shaft eccentricities has now been taken into account and the expression for $\Delta P_{\max}$ is as follows:

$$\Delta P_{\max} = \frac{3\mu\pi}{(1 + \frac{1}{2}\epsilon^2)} \left(\frac{D}{h}\right)^2 \omega N \left[\frac{1 + \frac{h_o}{h} \frac{\ell_o}{\ell} - \frac{\pi D}{\ell} \frac{\tan\phi}{\cos^2\phi} \left(\frac{h_o}{h} + \frac{2\ell}{\pi D \sin\phi}\right)}{1 + \frac{N}{N+1} \left(\frac{h_o}{h}\right)^3 \frac{\ell_o}{\ell} \left[\frac{\pi^2 D^2}{\ell_o^2 \cos^4\phi} + 1 \right] \left[\frac{1 + \frac{3}{2}\epsilon_o^2}{1 + \frac{3}{2}\epsilon^2} \right]} \right] \quad (4)$$

Substitution of the previous dimensional values gives a somewhat lower pressure of 7.3 psi. This is thought to be due to the inaccuracy of a one-dimensional model. An improved analysis based on a two-dimensional model of groove flow is underway. Nevertheless, the above model has provided much insight for continued improvement in the seal. For example, a second screw seal is presently being tested which is based on optimized pressure for a given length screw. If one adds a constraint to the axial length (L) of the seal, and also substitutes for the helix angle ϕ as shown below, one finds that optimal dimensions exist to maximize ΔP .

Constraint	$L = N (\ell_o + \ell)$
geometry	$\tan\phi \approx \phi = \frac{\ell_o + \ell}{\pi D}$
	$\sin\phi \approx \phi$
	$\cos\phi \approx 1$
	$\epsilon_o = \epsilon = 0$

Substitution of these values in equation (4), then differentiating with respect to ℓ_o , and equating to zero, leads to the result that:

$$\ell_o = \ell = \frac{L}{2N} \quad (\text{a square thread where land width must equal the groove width}). \quad (5)$$

Likewise, the optimal number of threads when $\ell_o = \ell_i$ is the solution to:

$$\frac{4\alpha}{L^2} N^4 + \frac{8\alpha}{L^2} N^3 - N^2 - 2N + 1 = 0 \quad (6)$$

$$\text{where } \alpha = \pi^2 D^2 \left(\frac{h_o}{h} \right)^3$$

Figure 36-15 plots $\Delta P_{\max}$ as a function of N for a square thread using $L = 0.100$, $h = 0.0035$ and $h_o = 0.0004$. The oil viscosity used is 30 cp which is the engine oil design viscosity at 122°F. The large increase in pressure obtainable for $N = 3$ rather than 0.2 of the present design is apparent; about 5 to 1. The new seal has $N = 2.0$ grooves, 28 threads/inch, a land and groove width of 0.018 inch, and is only 0.125 inch long overall. Figure 36-15 shows that nearly optimal pressure is obtained with $N = 2$ rather than 3, yet the screw is only two-thirds as long.

The power consumption of this optimized design is due almost entirely to viscous losses. The seal power consumption for $N = 2$ at 2400 rpm with h and h_o unchanged is only 18 mw. Testing of the new seal is currently in progress and the flowrate at zero pressure is as predicted.

The static pressure performance of the new seal will be determined and it will also be tested using sinusoidal pressures to simulate conditions in the crankcase. The same air compressor which is used for testing of the magnetic fluid seal can be cooperatively employed. An unknown at this time is whether the flywheel housing pressure will be pumped down 7 psi by the sinusoidal crankcase pressure. When the crankcase pressure goes negative, additional oil will tend to leak out of the screw seal along with what is normally pumped. If the screw should empty of oil on each half cycle, the seal could see a maximum differential pressure of 14 psi instead of 7. However, since the screw pressure capability is expected to considerably exceed 14 psi, this should not pose a problem.

Figure 36-15. Seal Pressure Capability as a Function of Number of Grooves

MAGNETIC COUPLING DEVELOPMENT

The Mark I magnetic coupling was completed this quarter. The basic configuration shown in Figure 36-13 of the January-March 1976 Quarterly Report* has remained. However, eddycurrent tests showed that a hermetically sealed ceramic support structure for the thin-wall membrane was not needed. This significantly simplified the design to a hermetic, one-piece titanium structure. Membrane eddycurrent shorting due to the titanium right end face was also minimal. This is attributed to a significant reduction in axial magnetic field leakage using an improved pole face geometry. The right end plate is now axially positioned within 0.015 inch of the magnets. This reduced clearance minimized the unsupported thin wall length of the membrane enabling it to be thinner and structurally sound without buckling. Data showed that the eddycurrent loss penalty in moving a 0.070 inch thick end plate from 0.065 inch to within 0.015 inch of the magnets was only 20 mw. The actual plate will be a sandwich structure (to be discussed) and is approximated by an 0.040 inch thick plate. A plate of this thickness results in only 12 mw of loss at 0.015 inch clearance.

The left end mounting-flange-to-magnet gap was kept at 0.065 inch as explained in the above referenced Quarterly Report. At a clearance of 0.035 inch, eddycurrent loss was 50 mw for a titanium alloy test flange of 0.060 inch thickness. But, when spaced at 0.065 inch, the power loss reduced to only 24 mw. The finalized sandwich structure equivalent thickness should be thinner, giving even better results.

The flywheel housing is closely spaced using a 0.015 inch radial clearance with the flywheel OD. As also explained in the above referenced Quarterly Report, the magnetic circuit return rings were increased in thickness, which has reduced the radial magnetic field leakage to the

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

point that no eddycurrent loss could be detected. Prior to increasing the return ring thickness, it was feared that the extremely large OD tangential speed would have caused the predominant eddycurrent loss in the system. Windage data also confirmed that there was no increase in power using a 0.015 inch circumferential clearance at a charge pressure of 14.7 psia.

The following is a summary of the individual component eddycurrent losses for the prototype coupling at a speed of 2400 rpm.

<u>Component</u>	<u>Power (mw)</u>
0.005 inch thin wall	90
Right end plate	12
Left mounting flange	24
Flywheel housing	0
Miscellaneous	<u>32</u>
Total	158

The miscellaneous loss was found to be an extra loss that results when the right end plate and left flange are assembled to the thin wall. In other words, the loss of the integral membrane was 32 mw greater than that of the individual losses; this is attributed to the formation of additional eddycurrent paths with an associated decrease in membrane resistance.

The windage loss of the flywheel at 2400 rpm while spaced 0.015 inch both axially and radially in its housing was 50 mw. Air at atmospheric pressure was used in the housing. A larger gap size did not decrease this loss nor did streamlining the sides of the coupling by making it into a smooth sided disc. This was verified by performing the same test with a machined aluminum disc of identical dimensions. However, when rotating the magnetic coupling outside of the housing, a windage loss of 90 mw resulted, and after streamlining its sides, this loss reduced to 50 mw. The conclusion drawn is that the housing enclosure eliminates drag by prohibiting pumped flow or air circulation within, thereby keeping windage loss at its minimum possible value (the same as a flat disc). The elimination of a streamlined flywheel

avoids additional complexity and saves weight.

The unfavorable aspect of the above 50 mw windage loss is that it will increase to 150 mw when the housing and crankcase is pressurized to 150 psia. This is the loss if nitrogen or air is used for flywheel housing pressurization. If argon is employed, the windage loss will increase to 200 mw since both the viscosity and density of argon are higher. This turbulent flow drag loss is proportional to the square root of the product of gas viscosity and density. A decided advantage of a ferrofluidic flywheel shaft seal is now obvious; namely, it is hermetic; and any compatible, containable gas can be used to pressurize the flywheel housing. For example, a windage loss of only 56 mw would result if helium were employed but this gas can not be contained for a ten-year life. Another candidate is methane which would dissipate only 88 mw. If a screw-viscosity seal is employed, argon may be the only alternative if the flywheel housing is pumped down as explained in the shaft seal section. This entire area is currently under investigation.

Another way to lower windage loss is to reduce the flywheel diameter or its speed. This is only practical if the flywheel stored energy requirement can be reduced. If flywheel diameter can be decreased, windage loss reduces by the fourth power of the diameter. If flywheel speed can be lowered, windage loss goes down as speed to the 2.5 power. However, if speed is reduced without increasing coupling torque, the present torque safety factor of 3.0 will be proportionately reduced. For example, a torque safety factor of 2.5 would result in a windage savings of 37 percent.

The flywheel is one of the heaviest engine components (about 0.5 lb.) and it is well worth trying to reduce its requirements. This problem has been studied by taking a close look at the system as a whole. The present flywheel inertia is based on a blood pump frequency of 2 Hz with most energy supplied to the left ventricle. If the torque amplitude of the right ventricle portion of the cycle is no longer ignored, blood pump frequency effectively exceeds 2 Hz with the benefit that the required flywheel stored energy may be proportionately reduced for the same speed fluctuation. This is clear with the aid of the following formula:

$$\Delta\omega = \frac{A}{\pi I f} \quad (7)$$

where $\Delta\omega$ = change in flywheel speed (rad/sec)

f = nominal blood pump frequency (Hz)

A = half peak sinusoidal torque amplitude of blood pump (in-lb)

I = flywheel inertia (in-lb-sec²)

For example, if blood pump frequency f is doubled, only half the flywheel inertia is required. Detailed blood pump operational data is needed to decide whether to decrease flywheel weight or to reduce windage losses.

Figure 36-16 is the final torsional characteristics of the prototype magnetic coupling. Peak torque is the required 1.5 in-lb which was obtained by driving the Armco-iron 8 pole pieces to magnetic saturation. The air gap field is about 23,000 gauss. When making the Mark I coupling, the air gap will be left slightly undersize to compensate for magnet strength variations, tolerances, etc. This will result in excess torque (see Figure 36-12 of the January-March 1976 Quarterly Report* which plots torque versus air gap size). After determining torque, the air gap could be increased until the specified torque is reached. Excessive torque capability is not desired since it only increases eddycurrent losses.

Dynamic torque testing of the coupling was performed using a flywheel drive motor and a hysteresis brake to apply the simulated blood pump load to the output armature. A photograph of the setup is shown in Figure 36-17(a). Figure 36-17(b) shows the sinusoidal torque input to be unidirectional. This was obtained by adding a DC current equal to the half amplitude current of the sinusoidal hysteresis brake. The drive motor (engine) must overcome this

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Figure 36-16. Magnetic Coupling Torque Characteristics

FIG. (a)

FIG. (b)

Figure 36-17. Magnetic Coupling Experimental Test Stand

average torque. Using a strobe light, the action of the coupling was observed. Relative angular deflection was not noticeable to the naked eye at either 2 or 4 Hz. Figure 36-16 predicts an oscillation amplitude of only ± 0.75 degree maximum with 0.5 in-lb of torque applied to the virtually inertia-less armature. A complete frequency spectrum was not run because the various natural frequencies are calculated to be high, 285 Hz for the armature and 23 Hz for the flywheel, based on the spring rate determined from Figure 36-17. Resonance reactions with either the engine or blood pump are not anticipated.

Having completed the magnetic circuit design and eddycurrent testing of the coupling, the structural design of the membrane was undertaken. The photograph of a test structure which includes a thin wall is shown in Figure 36-18. The top photograph shows the left end mounting flange on the armature side which is 2.9 inches in diameter. Fifteen radial stiffening ribs were used to make a rigid yet lightweight structure. The axial rigidity of this flange when sustaining a housing charge pressure of 150 psi is of utmost importance since the armature and membrane are only separated 0.020 inch to add minimally to engine length. The armature rotates inside the circular ring which is seen at the ID of the ribs. The rib ID is also the location of the 0.005 inch thin wall cylinder which is best seen in the bottom photograph. The bottom photo is a view looking toward the right end plate of the coupling. A second ribbed structure, composed of ten smaller ribs, stiffens this portion of the membrane. The membrane was O-ring sealed and mounted to the test fixture shown which was pressurized with oil. The centerline deflection of the membrane was recorded versus pressure. A deflection of 0.013 inch at 150 psi agreed well with the calculated value. In the future, it is planned to buckle the thin wall to failure to supplement existing buckling data since this test membrane has no further use.

The primary purpose of the deflection test was for design method verification; namely, that flat plate deflection formulas can adequately predict the deflection behavior of complex sections if the section rigidity per unit circumference is used analogously with the same quantity for a flat plate. For the ribbed structure shown, this quantity is a function of radius which still leads to a very complex analysis. Therefore, analysis of the problem was

Figure 36-18. Ribbed-Prototype Structure: Magnetic Coupling Membrane

attempted using strain energy techniques which have led to a simple formula which agreed within 10 percent of the experimental data. Since membrane deflection can be analyzed as the superposition of the deflection of the outer ribbed structure, the inner ribbed structure and the right end plate, the solving of a disc deflection with a central hole is sufficient. The following formula should be valid provided $\frac{R_o}{R_i} > 1.5$ and the number of ribs > 6 .

$$\delta = \frac{P (R_o - R_i)^3 D}{3EI} \left(1 + \frac{R_o - R_i}{4 R_o} \right) \quad (8)$$

P = external pressure

D = mean diameter

E = modulus of elasticity

I = mean moment of inertia

Having tested the ribbed structure, a sandwich design was evolved which should provide even better performance. It is shown in Figure 36-19 and the above formula can also be applied. Using ribs, only the outer material surface is stressed to full design limits. Consequently, some material is not fully utilized. However, the sandwich structure shown in Figure 36-19 stresses nearly all of the material to its maximum limit and is therefore an optimal weight design. This is the anticipated final membrane configuration, and as the table below shows, it permits a large weight reduction over the use of a flat plate structure. It also possesses the same stiffness and maximum stress as the ribbed design, with half the axial length.

<u>Construction</u>	<u>Weight</u>
Sandwich (Figure 36-19)	18 gm
Ribbed (Figure 36-18)	36 gm
Flat Plate	57 gm

The mounting flange axial length of 0.11 inch for the sandwich structure is ideal for incorporation of a screw viscosity seal on the output armature shaft (used for excluding blood pump oil

Figure 36-19. Sandwich-Membrane Structure

from the armature housing). Adding armature length and an anticipated seal length of 0.12 inch permits the combined length to be incorporated inside the mounting flange. The armature shaft need only be supported; i.e., by the gearbox bearings.

CONTROL SYSTEM DEVELOPMENT

One of the two new control valves was thoroughly tested on IVBM thermal converter No. 1 this quarter and gave the best operation to date. Figure 36-20 shows the engine response to positive (Figure 36-20a) and negative (Figure 36-20b) step inputs of torque. The engine and oil pressure response to a torque increase closes the control valve and also results in a control system gain reduction. The system is thus considerably overdamped and results in a smooth transition to its lowered steady state speed. Referring to Figure 36-20b, when the load is reduced, the control valve opens and steady state speed increases. One cycle of damped oscillation is noticeable, which then settles out to steady state in only four seconds. System gain increases as the valve continues to open due to the non-linear flow rate characteristics of orifice flow and for this input, creates a nearly optimal response; one that is quick-acting, yet produces little overshoot. The peak overshoot in Figure 36-20b is only about 50 rpm. It should be mentioned that the oscilloscope trace is slightly misaligned with the horizontal scale accounting for the apparent speed decrease after steady state conditions in each of the figures.

Figures 36-21a and 36-21b are the results of larger negative step inputs at reduced engine power. In Figure 36-21a, the step size was increased 1 in-oz from Figure 36-21b, resulting in 1-1/2 oscillations. Theoretically, dropping the torque to zero load should not produce any noticeable increase in oscillation. Yet, Figure 36-21b shows that at the maximum system power dissipation (3 watts) about six damped oscillations had resulted. This condition is due to flow choking of the working space tubing which was intentionally kept as small as possible to minimize engine working space dead volume. Calculation of the control valve poppet cross sectional flow area at the 3 watt dissipation point shows it to be numerically equal to the flow cross sectional area of the tubing. Consequently, valve flow momentarily becomes

Figure 36-20. Control System Response to Step Inputs

a) 12.0 in.-oz. STEP DOWN TO 7.0 in.- oz.

b) 11.0 in.-oz. STEP DOWN TO 4.0 in.- oz.

Figure 36-21. Control System Response at Low Engine Speed

saturated and the control system transiently loses its optimal capability. Tests have shown that when the load is fully removed from the engine, the control system will function well, although not optimally. It is anticipated that slightly larger diameter tubing will be used on the Mark I valve and dead volume might even be reduced by using a shorter length.

Figure 36-22 shows the combined engine and control valve hysteresis. The dynamometer was used as a load on the engine to decrease and then increase power, thereby obtaining the hysteresis loop. The speed or hysteresis band is extremely small, on the order of 5 rpm which is a repeatability of only ± 0.4 percent. Low valve hysteresis is exemplified by the very low ripple existing during steady state conditions seen in the previous figures which increases slightly as engine load is removed. The speed band obtained from Figure 36-22 is about 40 rpm from zero to full load. The steepness of the control curve decreases as load increases due to the reduced gain of the control loop as the valve closes. At about 635 rpm, the valve is apparently fully closed and the small increase in power thereafter is due to engine load characteristics. Power loss due to valve leakage in its closed position could not be detected. This was verified with a separate engine test without the valve mounted. Also, if the valve were cranked tightly closed by using the set point adjustment, no increase in engine power would occur. Even though this valve has a stiffer set point spring, enabling the engine to work up to 200 psi charge pressure, the speed band was lower than anticipated. This is attributed to an increase in oil pressure gain of the larger gear pump. Pump pressure was typically 75 psi instead of 60 at 600 rpm using Tellus 25 oil.

Improved phase shift data was obtained using the IVBM engine with the control valve installed. The data spread has been significantly reduced compared to the averaged curve data presented in Figure 36-33 of the January-March 1976 Quarterly Report*. The sinusoidal type curve resulting agrees with theory and is shown in Figure 36-23. The maximum phase lead of 17.5° agrees well with an analysis worked out for turbulent flow valve characteristics.

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

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Figure 36-22. Combined Control Valve and Engine Hysteresis

Figure 36-23. IVBM Engine Short Circuit Phase Shift Data

Figure 36-24 is a comparison sketch of the Mark I control valve concept developed this quarter. Its similarity with the IVBM valve lies in its use of the proven lead screw poppet actuator being controlled by an oil pressure bellows. It differs in that a second bellows is used to partially unload the set point spring force F_S . Since the crankcase is now pressurized by the introduction of this pressure to the right bellows, the set point spring force can be made to approach zero. This force is on the order of 8 lbs. in the IVBM valve and use of a long, heavy duty spring was required. With the new concept, a short, light duty spring is used and it can be located within the small bellows. Valve length as the figure shows is significantly reduced; it is in fact no longer than the width of the Mark I engine.

Analysis of the actuation force formulas in Figure 36-24 shows that an added advantage of the new design is reduced sensitivity to long term drifts in either the crankcase mean pressure or the buffer space mean pressure. About a factor of 2 reduction in set point sensitivity is expected with the new design.

LUBRICATION SYSTEM DEVELOPMENT

Detailing of all parts for the lubrication system test fixture was completed this quarter. Figure 36-25 depicts the prototype engine drive mechanism. All the drive parts are now in fabrication, in various stages of completion. The piston rod and wrist pins are in final machining. The displacer rod and connecting links are partially completed. Completion of the former part is awaiting successful demonstration of electroless nickel/titanium coating, while the latter are awaiting successful completion of bronze/titanium braze samples (see "Structural Investigations"). The crankshafts are up to the brazing operation, to be followed by nitriding and final machining. An order for timing gears and flywheel shaft/pinion gears was placed with Franke Gear Works of Chicago. Fabrication of the crankshaft counterweights and pleated oil filters was begun, while manufacture of the drive enclosure (simulated crankcase and its component parts) will begin early in the next quarter. New rollsock dies and clamps will be fabricated.

IVBM DESIGN

$$F = (P_b - P_c) A_1 - P_0 A_1 - F_s$$

MARK I DESIGN

$$F = (P_b - P_c)(A_1 - A_2) - P_0 A_1 - F_s$$

F = ACTUATION FORCE
P₀ = OIL PRESSURE RELATIVE TO CRANKCASE
P_b = BUFFER PRESSURE
F_s = SET POINT SPRING FORCE
A = BELLOWS AREA

Figure 36-24. Mark I Control Valve Concept

Figure 36-25. Mark I Prototype Engine Rhombic Drive

Two additional 3/16 inch pitch diameter two-gear pumps were fabricated this quarter. The gears and integral shafts were machined from stainless steel stock pinion gears. The back-leakage characteristics of these pumps were very good, typically only 9 percent at 100 psig. However, the input power requirements were greater than expected. At 50 psi and 0.0192 cubic inches per second of Tellus 25 oil, the pump required 0.3 watts of input power for an efficiency of 36 percent. Using Tellus 21 oil, the efficiency would be greater. Examination of the pumps revealed scratches and bronze pickup on the journal surfaces. Also, some evidence of wear on the relatively soft surface of the stainless steel gear teeth was evident. Analyses of the pressure loading of the pump gear support bearings indicated them to be marginally sized. In the prototype engine gear pumps, the shaft diameter will be increased from 0.094 to 0.110 inches.

Stock stainless steel gears had been used to date in these new pumps strictly as an expediency. The Mark I engine gear pump will utilize gears made from hardenable alloy steel. A gear cutter was purchased and will be used in the fabrication of the prototype engine pump gears.

TASK 37 - BLOOD PUMP AND POWER COUPLING RESEARCH AND DEVELOPMENT

WOUND-WIRE FLEXIBLE SHAFTS

A new IVBM flexible shaft with a very high strength music wire core was procured from S.S. White and put on life test to determine whether a longer fatigue life could be achieved by increasing the strength of the core wire. A photograph of this shaft is shown in Figure 37-1. Life testing of other shafts manufactured by S.S. White was continued. The flexible shaft test hours accumulated and the stiffness changes observed during this reporting period, as well as the total test hours to date, are shown in Table 37-1.

OTHER TORQUE TRANSMISSION SHAFT INVESTIGATIONS

Although the examination of the titanium flexible shaft after the 185 hour test reported in the previous quarterly report revealed no surface damage, a permanent bend with a radius of 4 inches and an offset of 0.25 inch was observed at one end of the shaft. Because it was believed that this permanent bend resulted from non-uniform bending of the casing, observed when the shaft assembly was mounted in the test rig, testing of the shaft was resumed with the same bend radius and torque loading but without a casing. The shaft failed at about the center of the permanent bend after it had operated 5 additional hours. A photograph of the shaft taken after failure is shown in Figure 37-2.

No further testing of the titanium flexible shaft is planned at this time because a special casing that bends with an approximately uniform radius will be required to ensure uniform bending of the shaft. A casing of this type will be procured and further development of the titanium shaft will be carried out if difficulty is experienced in meeting the 10 year operating life objective with a wound-wire shaft.

BLOOD PUMP BEARINGS AND LUBRICANTS

A special roller bearing for the pump mechanism yoke housing guide rollers, procured from the Miniature Precision Bearing Corporation (MPB), was immersed in Jojoba oil and put on

Figure 37-1. IVBM Flexible Shaft with Super Music Wire Core

TABLE 37-1
WOUND-WIRE FLEXIBLE SHAFT LIFE TEST DATA

<u>Type Shaft</u>	<u>Torque Loading</u>	<u>Test Hours April thru June</u>	<u>Total Test Hours To Date</u>	<u>Stiffness Change (in-oz./deg) April thru June</u>
S.S. White - 3/16 Inch Dia. Core	Normal 2.75 x Normal	---- 2188	2088 7747	----- None (No significant stiffness change observed since measurement after 435 hours of testing)
S.S. White - IVBM Configuration (3/16 Inch Dia. Core), Grease Filled	Normal 2.75 x Normal	---- 1693	964 2616	----- Decreased from 0.28 to 0.23.
S.S. White - IVBM Configuration (3/16 Inch Dia. Core), Oil Filled	2.75 x Normal	2158	3182	Decreased from 0.48 to 0.20
S.S. White - IVBM Configuration (3/16 Inch Dia. Core), Grease Filled, Super Music Wire Core	2.75 x Normal	2074	2074	Decreased from 1.5 to 0.46.
S.S. White - 1/4 Inch Dia. Core	Normal 1.5 x Normal* 2 x Normal*	283 481 871	4966 481 871	Decreased from 0.46 to 0.43. None None
S.S. White - 1/4 Inch Dia. Core	Normal 2.75 x Normal	---- 1563	184 5368	----- Decreased from 0.28 to 0.23.

*Torque loading was increased because lower loading could not be maintained by hysteresis brake.

Figure 37-2. Titanium Flexible Shaft after Failure

life test under the maximum design load. This bearing was designed for this application and is expected to have a significantly higher operating life than that of the special double row ball bearing (F155) also supplied by MPB for this application. Life testing of other MPB standard and special bearings operating in candidate mechanism oils was continued. The test hours accumulated during this quarter and the total test hours to date are as follows:

<u>MPB Bearing No.</u>	<u>Lubricant</u>	<u>Test Hours This Quarter</u>	<u>Total Test Hours To Date</u>
SR4	Vitrea 21 Oil	2280	11,304
F155	Sperm Oil	2016	6,024
F160	Sperm Oil	2280	4,920
F338	Jojoba Oil	672	672
S4	Vitrea 21 Oil	864	7,272

The S4 bearing test was terminated after 7,272 hours of operation because a noticeable buildup of wear debris in the oil occurred in a 24 hour period. When the bearing was examined, excessive wear of the ball retainer and some ball and race damage, attributed to the retainer wear, were observed. Although the life of the S4 bearing probably could be improved by using a plastic rather than a steel retainer, no further testing of this type bearing is planned because the special bearings that will replace it in the Mark I Prototype System blood pump will be more lightly loaded and will have plastic retainers. No frictional torque change or evidence of excessive wear of any of the other bearings was observed during the testing.

Additional wear tests were performed at 120°F with the four-ball tester to evaluate several mechanism oil-additive combinations that could extend the operating life of the pump bearings. The results obtained are as follows:

<u>Oil</u>	<u>Additive (5% wt.)</u>	<u>Average Scar Diameter (mm)</u>
Sperm	None	0.50
Sperm	Dibutyl Phosphite	0.22
Jojoba	None	0.44
Jojoba	Dibutyl Phosphite	0.22
Vitrea 21	None	0.60
Vitrea 21	Dibutyl Phosphite	0.22
Tellus 25	None	0.55
Tellus 25	Dibutyl Phosphite	0.30

Although dibutyl phosphite was very effective in reducing the wear with all the candidate mechanism oils, subsequent compatibility tests have shown that this additive cannot be used in the mechanism oil because it attacks silastic and causes it to lose almost all of its strength in less than a month.

TASK 38 - THERMAL AND HYDRAULIC RESEARCH AND DEVELOPMENT

The replacement connecting bellows were installed in the bellows pump coolant circulating system that was being tested in an IVBM blood pump, and testing was resumed. The test setup is shown in Figure 38-1. However, after 3 hours of operation, the axial crack shown in Figure 38-2 developed in one of the flexible coolant tubes. After the cracked tube had been replaced and another 6 hours of testing had been accumulated, a similar crack developed in the second original flexible coolant tube.

Since the tubes that failed and all the other flexible tubes on hand had been made from the same tubing stock, it was decided to continue the coolant circulating test with flexible tube assemblies that incorporated a plastic section. A photograph of one of these tube assemblies is shown in Figure 38-3. After 55 additional hours of operation, testing was stopped because oil leakage into the coolant loop was again observed.

When the ventricles were removed from the blood pump so that the coolant system components could be examined, it was found that one of the linear bushings had failed and the bushing balls had damaged one of the bellows assemblies. Photographs of the broken linear bushing and damaged bellows are shown in Figures 38-4 and 38-5. A leakage test of the coolant loop revealed that the oil leakage into the loop had occurred through a tear in the bellows that had been damaged by the bushing balls. Microscopic examinations of the broken linear bushing and the damaged bellows are being performed to determine the cause of the failure. Based on a successful 30 day life test of a similar bellows, it seems almost certain that the bushing failure caused the bellows failure*.

*After the end of this reporting period, the pump was rebuilt with a different linear bushing and a replacement bellows. Operation for the last three weeks has been without incident.

Figure 38-1. Bellows Pump Coolant Circulating System Test Setup

Figure 38-2. Flexible Coolant Tube with Axial Crack

Figure 38-3. Flexible Coolant Tube with Plastic Section

Figure 38-4. Linear Bushing after Failure

Figure 38-5. Damaged Bellows Assembly

TASK 39 - SYSTEM INTEGRATION AND INTERFACING RESEARCH AND DEVELOPMENT

IVBM IMPLANTATION PREPARATION

The assembly and sterilization sequence outlined in the January-March 1976 Quarterly Report* has been expanded and refined to provide a detailed procedure for field assembly and sterilization of the converter. The exposure time required to achieve a sterile condition using Oxyfume[®]-12 is expected to be about 12 hours and will be experimentally verified prior to shipment of the system. On the advice of the University of Utah, the polyethylene carrying bag for transporting the sterile converter from the glove box to the operating room will be replaced with a sterile cloth wrap. In addition, based on the experience at the University of Utah, the purge time for removing ethylene oxide from the surface of the converter has been tentatively reduced from 72 hours to 24 hours.

An analysis was carried out to define allowable leakage to insure that the ethylene oxide concentration in the laboratory remains below the permissible concentration of 50 ppm by volume. A release of 0.4 pounds of Oxyfume[®] mix into the laboratory would fill the room to a depth of 4.5 feet at this concentration. Over a period of 24 hours (twice the expected sterilization time), the total released Oxyfume[®] can be maintained below 0.4 pounds if the internal pressure is not allowed to exceed 6 inches of water, and the air leak rate of the glove box is less than its design leak rate which is equivalent to a pressure fall less than 1 inch of water/hour at a pressure of 4 inches of water.

The test plans for the sterilization test as well as the IVBM final assembly procedure have been updated to incorporate an air test of the glove box prior to sterilization and to limit the maximum internal pressure to 6 inches of water during the test procedure. To eliminate the possibility of contamination from the Oxyfume[®] bottles, the bottles have been moved outside

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

the laboratory, and two stop valves are incorporated into the lines. The portion of line between the two stop valves is to be vented to the outside atmosphere except where Oxyfume[®] is actually being used.

Because of performance difficulties encountered with the static vacuum foil insulation package (reported under Tasks 40 and 41), it was decided to rework the dynamically pumped package to provide a backup static package for use with the nuclear heat source. The design for this reworked package had been completed when the failures of both nuclear heat sources occurred as reported in Task 40. After this event, work on the backup nuclear insulating package was stopped, and efforts were initiated to modify the existing electrically heated head to provide an implantable electrically heated version of the IVBM. These modifications are discussed under Task 41.

To minimize the possibility of damage due to the test animal's activity after an implant, the implantable electrically heated converter will incorporate a static vacuum package. Because the heaters used heretofore are not suitable for use in a static vacuum system, and no practical means could be found to isolate the heater from the vacuum, a new heater is being designed for this application. To eliminate the outgassing problems of the previous heater, the new heaters will be made of high purity alumina and Kanthal[®] A-1 resistance wire. At present, two heater design concepts are being pursued. One requires specially manufactured ceramic parts, and may be subject to a long lead time in obtaining the ceramics. A request for quotations has been issued to obtain a realistic estimate of the time required to manufacture these ceramic components. The second design concept can be manufactured from stock ceramic shapes using equipment presently available at Westinghouse Advanced Energy Systems Division. Thermal calculations indicate that the wire temperature for this design approaches the operating limits for reliable long life operation of the heater wire. One of these heaters is being constructed and will be tested to determine if satisfactory life can be obtained.

The IVBM converter has been fitted with sturdier roll socks. Testing with these roll socks has confirmed that no backup oil pressure need be supplied while pressurizing the converter prior

to startup. As a result, the auxiliary oil system is no longer needed and has been removed from the auxiliary cart. The cart space made available is being used to install alarms to alert operating personnel to low coolant flow, low buffer pressure and high or low converter speed.

Bench testing of the IVBM system, as reported under Task 40, indicates that a considerable margin exists in terms of the converter's ability to drive the system with higher than design blood pressures. It is desirable to provide higher flow capability than observed in the tests with the design aortic pressure of 100 mm Hg. As one step to achieving higher flow, a gear set to provide a 1.1:1 step up between the converter and the blood pump has been ordered to replace the 1:1 set presently installed in the right angle gear box. This will have the effect of allowing the blood pump to operate at a higher speed for a given converter speed, improving the flow capability of the system at moderate blood pressures. It will, of course slightly reduce the torque margin available for handling higher than design blood pressure.

Some additional increases in net flow are expected with improved valves. Since the improved valves will only prevent back flow of blood already pumped, the gains achieved by this method will not significantly affect the maximum power requirements for a given pump speed. Based on experimental data on the ventricles, increased flow also can be realized by stiffening the ventricles, reducing their dilation during systole. This gain will increase the maximum power requirement for the pump operating at a given speed. If significant flow increases are achieved by this method, it may be necessary to return to the 1:1 gear set for implantation. The actual gear ratio and ventricles used for implantation will be selected based on experimental data.

MARK I PROTOTYPE SYSTEM

The design specification for the Mark I Prototype System has been prepared. The performance goals are based on the review of the blood hydraulic power requirements and the conceptual configuration for the Mark I Prototype converter as reported in the previous Quarterly Progress

Report*. The system design specification is included as Appendix A of this report.

More detailed specifications for the Thermal Converter and Blood Pump and Coupling Mechanism are being prepared based on the over-all requirements spelled out in the System Specification. In preparation for writing the detailed specifications for the converter, several studies are being carried out.

Present plans call for the heat source to be incorporated as a permanent part of the hot finger/vacuum foil insulation assembly. This configuration will improve converter performance by minimizing the dead space inside the converter working volume. Also, it will remove the requirement that the heat source fit inside the converter hot finger, therefore de-coupling the converter and heat source dimensions. Both the heat source and working volume can be designed to their optimum proportions, again improving converter performance and minimizing the overall length and weight of the converter assembly. Before this feature could be incorporated as part of the design specification, several areas required investigation. Based on discussions with LASL, it appears likely that suitable braze materials to provide good thermal contact between the hot finger and heat source should be available. To identify a particular material, and to specify the design requirements for the joint, the following items must be defined:

- Maximum process temperature of braze
- Expected operating temperature
- Expected loads in normal operation

A thermal analysis indicates that in the event the heat source separates from the hot finger, unacceptably high temperatures can be reached before the hot finger fails and the insulation effectiveness is reduced. To insure adequate protection for the heat source in the event of

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a malfunction of the system, a mechanical joint must be incorporated between the hot finger and heat source with the braze primarily functioning to provide a low thermal impedance contact. Alternatively, reliable back-filled devices must be incorporated in the design. The problems of building the insulation package around the fueled heat source were evaluated and it was determined that this process is feasible. Several alternative routings to transfer the helium released from the vent out of the vacuum insulation package were examined to determine that such venting is feasible. A brazed joint with a mechanical back-up will therefore be specified as part of the hot side interface specification, with a purely mechanical joint as a back-up. The temperature and loads discussed above are being investigated, and numerical values will be incorporated in the specification.

An effort to define the expected torque variations on the converter has been initiated. To be defined are the instantaneous torque as a function of crank angle for the design case, and for 25% overload. In addition, the expected mean torque as a function of blood pump speed and mean blood pressure is being investigated. These values will be incorporated as part of the coupling mechanism - converter interface specification.

MARK I PROTOTYPE THERMAL CONVERTER

A preliminary study of off-design performance of the Mark I prototype thermal converter was conducted early this quarter. The purpose of the study was to present Westinghouse Advanced Energy Systems Division systems analysts with data upon which they could make preliminary decisions on the matching of system components for optimal prototype system performance.

Figures 39-1 and 39-2 present the results of this study. The baseline engine design is as outlined in Table 39-2 of the January-March 1976 Quarterly Report*. That is, at 600 rpm, with a 33 watt heat source, a charge pressure of 17.9 atmospheres, a heater temperature of

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- A. FIXED CHARGE PRESSURE & OIL VISCOSITY
- B. OIL VISCOSITY DIFFERENT AT EACH SPEED FOR OPTIMUM PERFORMANCE
- C. CHARGE PRESSURE & OIL VISCOSITY DIFFERENT AT EACH SPEED

Figure 39-1. Mark I Prototype Engine Off-Design Characteristics

Figure 39-2. Mark I Prototype Engine Transient Characteristics

925°K, and lubricating oil viscosity of 33 centipoise, the output power is 7.1 watts. Figure 39-1 illustrates how the engine steady-state output power and heater temperature would vary versus operating speed for several pertinent cases. Case A represents the output power and temperature versus speed for the charge pressure and oil viscosity quoted above. If the engine were operated at speeds higher than 600 rpm, the available shaft power would be decreased, as would the heater temperature. This type of operation would be permissible since the maximum heater temperature of 925°K (for ten years life) would not be exceeded, and the hydrodynamic bearings would have more than adequate load carrying capability at the higher speed. However, operation at lower speed than 600 rpm would not be permissible (dotted lines in Curves A) since the equilibrium heater temperature would be higher than that permissible for long life, and the hydrodynamic bearings would not have sufficient load carrying capacity. However, one can see, for example, that if the system were operated so that the nominal load on the engine were 5.5 watts and the engine operated at 670 rpm, under overload conditions the load could increase to 7.1 watts (for 33 watts isotope inventory) and the engine slowdown to 600 rpm. Higher loads would risk stalling the engine.

Curves B of Figure 39-1 show the locus of all performance points assuming the minimum oil viscosity needed for successful bearing operation at each speed. However, operation at any point to the left of a given point on Curve B would not be permissible since operation at lower speed with the minimum oil viscosity for a higher speed would overload the bearings. Note that the heater temperature varies identically with speed as in Case A.

Curves C of Figure 39-1 show what the performance would be if the minimum requisite oil viscosity were used at any speed, and the charge pressure were adjusted at each speed to maintain maximum permissible heater temperatures. Here it can be seen that power outputs slightly greater than 7.1 watts could result if operation were at less than 600 rpm. However, the flywheel mass and magnetic coupling capability would have to be increased for operation at the lower speed. In that sense, the 600 rpm baseline operating speed is now and always has been a compromise figure.

Figure 39-2 typifies the difference between instantaneous power output and steady state power output from the engine for a step-change in load torque from the baseline operation at 600 rpm. The curves labeled "steady-state" are Curves A from Figure 39-1. The curves labeled "transient" assume no speed control system. If, after operating for a long time at 600 rpm, the load torque were reduced, the instantaneous output power from the engine would increase and the engine speed increase in a manner dictated by the transient curve. This is because the engine is drawing sensible heat from the heat source. However, as the temperature decreased, the output power available would approach that defined by the steady state curve. A similar but opposite effect holds true for a sudden torque increase.

Table 39-1 shows a weight estimate for the Mark I prototype thermal converter. The total weight is presently estimated to be 1770 grams. Preliminary indications from LASL are that the 300 grams allotted to heat source and insulation may be inadequate for the new, short L/D heat source. The weight estimate for the crankcase and flywheel housings of 425 grams, however, should prove to be conservative. With proper design, significantly lower weights may yet be achievable. Thus, while the weight of individual components will probably be driven up or down as the detailed design phase progresses, the total should remain no higher than indicated in the table.

TABLE 39-1
MARK I PROTOTYPE WEIGHT ESTIMATE

<u>Item</u>	<u>Weight, gms</u>
Heat Source and Vacuum Insulation	300
Hot Finger and Ambient Cooler	200
Rhombic Drive	250
Flywheel/Magnetic Coupling	280
Crankcase and Flywheel Housings	425
Oil	65
Other (Controls, Fittings, Gears, Oil Pump, etc.)	<u>250</u>
Total	1,770

BLOOD PUMP VENTRICLE INVESTIGATIONS

At the Heart Program Contractor's Interface Meeting held at Westinghouse Advanced Energy Systems Division on May 11-12, 1976, the University of Utah reported the following:

- a) High left atrial pressures, above 30 mm Hg, were experienced in the ERDA No. 32 implant experiment. However, it was found that by banding the pulmonary artery and causing the right ventricle pressure to approach 100 mm Hg, the pulmonary flow was decreased sufficiently to cause the left atrial pressure to drop to 15 mm Hg. The animal's condition improved after this second surgery, and he was then able to breathe on his own and eat and drink.
- b) Measurements of the ventricle housing distension due to internal pressure indicated that the higher pressures in the left ventricle caused its output to be about 15 percent smaller than that of the right ventricle.

The above results led to the conclusion that the high left atrial pressures and pulmonary edema observed in many of the ERDA blood pump implant experiments performed to date probably were caused by an imbalance of the pulmonary and systemic flows that resulted from the higher volumetric efficiency of the right ventricle.

WAESD has supplied the University of Utah a modified IVBM blood pump mechanism with pusher cups that make the stroke volume of the right ventricle 9 percent smaller than that of the left ventricle. A blood pump with this mechanism and redesigned ventricles has been evaluated in mock loop tests at Utah and found to deliver balanced pulmonary and systemic flows under normal operating conditions. An implant experiment with this pump is scheduled to be started in July.

The required differential stroke volume of the blood pump mechanism was obtained by increasing the outside diameter of the left ventricle pusher cup 0.021 inch and decreasing the outside diameter of the right ventricle pusher cup 0.281 inch. The pump mechanism yoke housing was reworked to accommodate the smaller right ventricle pusher cup.

BLOOD PUMP WEIGHT REDUCTION

Testing to establish the degree of salt water corrosion protection provided magnesium by a covering of silicone rubber was continued with emphasis placed upon evaluating Medical Grade Silastic sheeting. The results of corrosion tests of magnesium coated with Medical Grade Type A Silastic adhesive, both alone and in combination with sheeting were reported in the previous Quarterly Report*. Described in this report are the results obtained in corrosion tests of magnesium covered solely with Silastic sheeting in 0.020, 0.040 and 0.060 inch thicknesses.

Photographs revealing the appearance of a magnesium specimen covered with a 0.020 inch thickness of Silastic sheeting and then exposed to 100°F salt water for 17 to 55 days are shown in Figure 39-3. Exposure to the saline solution caused no apparent corrosion of the magnesium specimen for up to 30 days, even though water penetrated the sheeting barrier (entrapped droplets of water can be seen in the Figure 39-3 photographs). Some corrosive attack, as evidenced by darkening of the specimen surface, was apparent after 48 days. The test was terminated at 55 days, and the sheeting removed to allow closer inspection of the specimen. A superficial surface attack had been sustained similar in degree to that of a deep tarnish. The attack was less severe than reported for corrosion specimens coated with the Type A Silastic adhesive, and could be removed for the most part by buffing vigorously with fine steel wool.

Photographs showing the appearance of magnesium specimens coated with 0.040 and 0.060 inch thicknesses of Silastic sheeting and exposed for 37 days to salt water are shown in

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

(a)

(b)

(c)

(d)

Figure 39-3. Magnesium Coated with 0.020 inch Thick Silastic Sheeting and Exposed to Saline Solution. (a) 17 days exposure in 100°F 1% salt water; (b) 30 days exposure; (c) 48 day exposure; (d) 55 days exposure (covering opened)

Figure 39-4. These specimens tarnished in a manner similar to that noted for magnesium protected with 0.020 inch thick sheeting. Very small droplets of water, which are not apparent in the Figure 39-4 photographs, were also present inside the Silastic covering. Testing of these specimens is continuing.

An effort to evaluate the salt water corrosion protection provided beryllium by Nedox coatings has been completed. Nedox coatings are produced by proprietary processes of the General Magnaplate Corporation, and combine a deposit of electroless nickel with a teflon sealing treatment. The coatings are reported to be resistant to attack in sea water*, and their applications have included desalination plant tubing.

Two Nedox coatings whose characteristics are described in Table 39-2 were evaluated. The major differences between the coatings were in the color and hardness of the Teflon surface seal, the U-4 treatment producing a black hard surface layer, and the SF-2 treatment a soft clear surface layer. A photomicrograph showing a cross section through the U-4 coating as-applied to beryllium is shown in Figure 39-5. The Teflon and electroless nickel layers measured nominally 0.0010 and 0.0005 inch in thickness, respectively.

TABLE 39-2

NEDOX COATINGS EXAMINED FOR CORROSION PROTECTION OF BERYLLIUM

<u>Treatment</u>	<u>Appearance</u>	<u>Specification</u>
U-4	Black, Hard**	} Mil-C-26074
SF-2	Clear, Soft***	

* Nedox Data Sheet, General Magnaplate Corporation, Linden, N. J.

** Similar to hard black Teflon commonly found on cookware.

*** Easily removed with thumb nail.

As-coated

37 Days
Exposure

Figure 39-4. Magnesium Specimens Coated with 0.040 and 0.060 inch Thicknesses of Silastic Sheeting and Exposed to Saline Solution. 0.040 inch sheeting thickness on Spec. No. 18; 0.060 inch thickness on Spec. 17.

Figure 39-5. A Photomicrograph Showing the Cross Section of Nedox U-4 Coated Beryllium

The coatings were applied to 2 inch diameter by 1/8 inch thick discs of beryllium. Photographs showing typical coating specimens are presented in Figure 39-6 and 39-7. Visual examination revealed the presence, in all cases, of coating defects, typical examples of which are also shown in Figures 39-6 and 39-7. Chips in the outer Teflon layers, and raised (blistered) areas on the U-4 coating at regions of hole locations were observed.

The condition of the coated specimens after 1, 9, and 23 days exposure to the saline solution is displayed in Figures 39-8 and 39-9. Corrosion developed at a corner site on the Nedox SF-2 treated specimen during the first day of saline water exposure. Three local corrosion sites were developed in 23 days of exposure, with very deep pits forming at one region. Galvanic coupling between the beryllium and the electroless nickel plate is suspected to have occurred at the deep pitted site.

The Nedox U-4 coated specimen was somewhat more resistant to attack, in that major coating failure took between 9 and 23 days to develop (Figure 39-9). Corrosion developed at hole locations which were sites where the as-applied coating was generally blistered. A very deep corrosion pit developed at one of these sites, probably a result of galvanic coupling.

The presence of defects in the Nedox coatings applied to the beryllium samples is suspected to have caused early corrosion failure upon salt water exposure.

ELECTRIC MOTOR DRIVE SYSTEM FOR BLOOD PUMP IN VIVO TESTING

A seven pin electrical connector instead of the original five pin connector is now required at the end of the umbilical cord of the new implantable blood pump drive motor assembly. This connector is required to accommodate the additional wiring between the controller and motor that was added when the drive system was modified to obtain the required motor speed control. Since the new connector would cause an objectionable increase in the size of the end of the umbilical cord, the shape of the connector adapter was modified. With this change, it was possible to make the envelope of the end of the umbilical cord smaller than it was originally.

1X

10X

Figure 39-6. Nedox U-4 Coated Beryllium Corrosion Test Specimen.
(Note the chipped and raised coating revealed by the
higher magnification view of a hole location.)

1X

10X

10X

Figure 39-7. Nedox SF-2 Coated Beryllium Corrosion Test Specimen.
(Note coating defects revealed in the higher magnification photomicrographs.)

Figure 39-8. The Progressive Corrosion of Nedox SF-2 Coated Beryllium
Caused by Exposure to 1 Percent Salt Water Held at 100°F

Figure 39-9. The Progressive Corrosion of Nedox U-4 Coated Beryllium Caused by Exposure to 1 Percent Salt Water Held at 100°F

TASK 40 - IVBM SYSTEM PERFORMANCE TESTING

IVBM thermal converter No. 1 was run-in and tested at Philips Laboratories. A problem noted during these tests was that of low oil system pressure. When the drive was disassembled to investigate, the inlet tube broke off the oil filter. It is possible that a crack in the joint had been responsible for the low oil pressures. After repair of the oil filter, the thermal converter was returned to test. The functioning of the new control valve was evaluated as reported under Task 36. The steady-state power output with 33 watt heat input was 5.5 watts at 600 rpm and a charge pressure of 195 psia.

The modifications to the IVBM static vacuum insulation package to accommodate the coated Pu-238 heat source (HF-3) were completed early in this quarter. After completion of modifications, a dynamometer test of the Pu-238 fueled thermal converter No. 2 was performed using the modified insulation package. After about 10 hours of operation, the internal temperature and power output of the converter began to decrease to ~ 3 watts. Concurrently, the external temperature of the insulation package began increasing. The test was terminated and the insulation package checked as reported under Task 41.

An additional test with the same converter using the dynamic insulation package verified earlier tests with the dynamic head: ~ 5 watts power output. This tended to confirm that the lower output using the static head was related to insulation problems rather than to engine performance per se.

In the process of checking the static insulation package its performance recovered to the level recorded after assembly. Since no source of leakage could be pinpointed, it was decided to continue IVBM testing using this insulation package, while the dynamic package was re-furbished to provide a static back-up package. After a short converter check-out on the dynamometer, a system test was performed using the static vacuum insulation package on

IVBM thermal converter No. 2. During this test, flows of approximately 9 liters per minute were obtained with a mean aortic pressure of 101 mm Hg, and the converter operating at about 760 rpm (101 bpm). The mean aortic pressure was increased to 167 mm Hg while adjusting the atrial pressures so that the ventricles were completely filling on each stroke. In this condition, the converter was operating at 622 rpm (83 bpm) with no difficulty. The flow fell off to 5.5 l/min at this operating point for the system. In the course of this run, there were no signs of problems with the insulation package, and the test confirmed the ability of the IVBM system to operate successfully against any expected blood pressure for "in-vivo" experiments.

Following this test, the components of the system were set up to simulate an implantation. While operating the converter on the dynamometer in preparation for transferring the converter to the blood pump, a sudden drop in argon pressure was observed. This was traced to a roll sock failure. After disassembly, it was observed that the outer clad of the Pu-238 heat source had failed. A thorough examination of the external surfaces of the Pu-238 heat source capsule, of the glove box internals, and the laboratory area determined that no leak of radioactive material resulted from this outer clad failure.

New roll socks were installed in the converter, and plans were made to operate using the second heat source (HF-2). On trying to pressurize the converter, it was found that the bellows in the speed control valve had failed. After disassembling the converter, the outer clad on the second heat source was observed to have failed in a manner similar to the first failure. Another thorough examination of the external surfaces of the Pu-238 heat source capsules, of the glove box internals, and the laboratory area confirmed that no leak of radioactive material resulted from either of the two outer clad failures. Both heat sources were returned to LASL to investigate the cause of the failures. To assist LASL in their analysis of the heat source failures, a history of the usage of each heat source was prepared. These histories are presented as Appendix B of this report. Figures 40-1 and 40-2 show the failures of the coated and uncoated sources respectively.

Figure 40-1. Coated Pu238 Heat Source

Figure 40-2. Uncoated Pu238 Heat Source

An examination of the dismantled converter suggests the following to be the probable sequence of events leading to the roll sock failure. The inner O-ring on the hot finger/cooler interface may have partially popped out of its groove as the two assemblies were brought together. As the screws were tightened, the O-ring became trapped in a position wherein it was forced against the displacer wall. When the unit was started, this O-ring was torn apart, and pieces entered the 15 mil gap between the regenerator liner and displacer. Large pieces of charred material were found adhering to the hot region of the displacer in two adjacent quadrants, and bits of uncharred rubber were found at the cold end of these quadrants after disassembly at Westinghouse Advanced Energy Systems Division. The O-ring probably exerted a sufficient side load on the displacer to deflect the displacer rod 1 to 2 mils and to cause the inner clamping ring of the inner roll sock to abrade the roll sock just below its outer clamping ring. This is where the break occurred. Evidence of abrasion was noted on the short side of the break (the part held by the outer clamping ring) but nowhere on the longer piece. This indicates that rubber did not rub against rubber (as when a roll sock becomes inverted) but rather against some other agent, such as the inner clamping ring.

One step which will be taken to help avoid a recurrence is to replace all O-rings in the field with ones which have been prestretched at Philips Laboratories. The prestretching affords a better fit in the groove, with less tendency to pop out. Further, the assembly procedure cautions regarding watching the O-ring as the hot finger and cooler are brought together must be rigorously followed.

A check of the pressures to which the failed control valve bellows was exposed showed these pressures to be within the bellows design capability. Internal stops in the valve would have limited over-stretch and over-compression. The most probable cause of the failure is thus a problem with the metallurgical quality of these stock bellows.

The bellows assembly was removed from the control valve and the site of the failure located. Two pinhole-sized leaks were noted in one convolution. The bellows were sent to the Philips metallography laboratory for microscopic examination which is still in process.

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Following the failures of the two heat sources, testing of the IVBM system has been suspended until a new heat source becomes available, or modifications to the electrically heated IVBM to make it suitable for implantation are completed.

TASK 41 - IVBM SYSTEM FABRICATION AND FABRICATION SUPPORT

IVBM THERMAL INSULATION SYSTEM NO. 2

As reported under Task 40, the IVBM system test was terminated when the thermal converter power output decreased and the insulation package external temperature increased, symptomatic of loss of vacuum in the static insulation package. The insulation package, IVBM No. 2, was returned to the Thermal Insulation Laboratory for diagnosis. Initial heatup on the test stand did verify a degradation in thermal insulation performance during the system test. A temperature of 714°C with a power input of 12.5 watts was initially recorded. However, after seven days operation, the thermal performance had improved and was equivalent to that measured before the system was delivered initially, achieving 842°C with 12.5 watts power input. Following the improvement in thermal performance, the heater cavity was pressurized with 30 psia argon for seven days and 30 psia helium for two days without any observable degradation in thermal performance. Table 41-1 lists the daily thermal performance of IVBM No. 2 while under test and evaluation in the Thermal Insulation Laboratory.

FLEXIBLE SHAFT PROTECTION

A silicone rubber covering for the IVBM flexible shaft casing was designed and constructed. The design is described in Figure 41-1 and consists of 0.040 inch thick sheeting wrapped and bonded to 3/8 inch ID x 5/8 inch OD tubing. The sheeting wrap is extended beyond both ends of the tube producing glands used to stretch over and seal onto the end fittings which would be attached to the flexible shaft during an implant. Medical grade Silastic® silicone rubber is used for the tube and sheet components of the covering.

The initial step in constructing the Silastic® covering is placement of the 5 inch length of 3/8 x 5/8 inch tubing over the IVBM flexible shaft casing. This is accomplished by immersing the tube in Xylene for 5 minutes to soften, swell and lubricate it, then sliding it over one of the 3/4 inch diameter knurled nuts at the end of the flexible shaft casing.

TABLE 41-1

HISTORY OF IVBM NO. 2 INSULATION PACKAGE
PERFORMANCE TESTING AFTER SYSTEM TEST MALFUNCTION

Days	Outside Temp. (°C)	Hot Finger Temp. Range (°C)	Power (watts)	Comments
1	53.9	20.4 - 792.8 *	21.8	From 445°C to 792.8°C in 2 hrs.
1	44.0	792.8 - 714.6 *	12.5	
2	44.3	714.6 - 760.4 *	12.5	
3	41.4	760.4 - 808.4 *	12.5	
4	37.4	808.4 - 823.9 *	12.5	
5	31.6	823.8 - 832.1 *	12.5	
6	31.0	832.0 - 833.0 **	12.5	Steady temperature reached
7	29.9 - 31.0**	830.0 - 833.0 **	12.5	Steady temperature reached
8	32.5 - 31.4*	842.0 - 780.3 *	12.5	Argon at 15 psi absolute introduced into heater. Temperature decrease results from added heat loss in heater insulation - note steady outside temperature.
9	30.5 - 32.6**	779	12.5	
10	31.1 - 33.0**	776	12.25	
11	30.3 - 32.0**	777	12.25	
11	30.0 - 34.0*	777 - 808.4 *	13.03	Increased argon pressure to 30 psi absolute in heater cavity
12	33.5 - 41.2*	808.5 - 816.6 *	13.02	
13	40.0 - 42.0**	816.6 - 817.3 **	13.02	Weekend - Lab air conditioner was off due to malfunction causing higher outside temperatures
14	39.9 - 42.0**	815.9 - 817.3 **	13.02	

* Indicates temperature trend throughout the day

** Indicates temperature range fluctuation at equilibrium

Table 41-1 (Continued)

Days	Outside Temp. (°C)	Hot Finger Temp. Range (°C)	Power (watts)	Comments
15	40.1 - 33.4 *	815.9 - 808.0 *	13.02	Air conditioner on and outside temperature decreases Helium added to heater cavity at 30 psi absolute pressure after evacuating the argon. Due to helium conductivity in heater cavity
15				
15	33.7 - 28.1 *	808.0 - 675.2 *	13.02	
16	28.1 - 27.1 *	675.2 - 669.1 *	13.02	
16	28.1 - 35.7 *	669.1 - 837.9 *	21.66	
16	35.5 - 35.7 *	835.8 - 840.4 *	20.44	
16	35.7 - 36.3 *	844.7 - 844.7 *	19.54	
16	36.3	844.7 - 827.8 *	19.10	
17	36.3 - 32.6 *	818.3 - 846.4 *	12.57	
				Evacuated system and reestablished the initial power

Component	Materials*	Dimensions
Tube	Vulcanized Tubing	3/8" ID x 5/8" OD x 5" L
Gland	Unvulcanized Sheeting	0.040" Thick

* Medical Grade Silastic®; gland sheeting wrapped on tube and bonded by vulcanizing 3 hours at 300°F.

Figure 41-1. IVBM Flexible Shaft Cover

The tube is positioned so that its ends are equidistant from the corresponding ends of the casing. At this point, the assembly is aired for a day to evaporate residual Xylene and to allow the Silastic[®] tube to develop a snug grip on the casing.

After airing, a wrap of 0.040 inch thick unvulcanized Silastic[®] sheeting is applied over the tube-casing assembly and bonded to the tube by vulcanizing for 3 hours at 300°F.

A photograph of the Silastic[®] cover applied to an IVBM casing is presented in part a of Figure 41-2. The sealing glands are shown rolled back in the photograph as would be the situation when prepared for attachment to the motor and blood pump. The remaining photographs in Figure 41-2 show the sequence of gland and fitting changes which would occur during a two stage implant. These are as follows:

- Part b Attachment of the covered casing to the blood pump and electric motor.
- Part c Silastic[®] glands stretched over the blood pump and motor fittings. Note that the gland at the blood pump end seals on a piece of Silastic[®] tubing covering the fitting. The electric motor fitting will also have a covering of Silastic[®] (not shown) onto which the other gland will seal.
- Part d Silastic[®] gland drawn back and the converter fitting attached. The converter fitting is also covered with a Silastic[®] tube onto which the gland will seal.
- Part e Glands stretched over the blood pump and converter fittings.

IVBM THERMAL INSULATION SYSTEM NO. 1 DISASSEMBLY

Initially, the dynamically pumped IVBM thermal insulation system No. 1 was to be disassembled to reinstrument the hot finger with ceramic metal feedthroughs to permit use as a static vacuum system. Problems with the isotope capsules (see Task 40) have delayed this rebuild. Instead,

Blood Pump
End

Electric Motor and
Converter End

(a)

(b)

Blood Pump
Gear Box
Fitting

Electric
Motor
Fitting

Silastic Tube

(c)

(d)

Converter Gear
Box Fitting

Silastic Tube

(e)

Figure 41-2. Operating sequence of Silastic[®] IVBM Flexible Shaft Cover for Two Stage Implant (1/2X). The Silastic[®] cover is of Type A Design. (a) Covered IVBM flexible shaft casing with glands drawn back; (b) blood pump and electric motor in place; (c) Glands pulled over pump and motor fittings; (d) Gland drawn back from motor-converter end with converter in place; (e) Glands over pump and converter fittings.

the system was disassembled to obtain the cups for use in a static vacuum electrically heated insulation package.

The system was disassembled as follows:

1. The outer container was removed by grinding away the weld bead using a hand held rotary grinder.
2. The outer container was then lifted off. Figure 41-3a and b show two views of the outer nickel cup. The circular blemish 1/3 from the bottom corresponds to the position of the pumpout port used to dynamically evacuate the package since the sheathed thermocouples began to leak.
3. Cups 60, 59, and 58 were removed individually from the 60 cup set. Cups 1 through 57 were removed as a stacked unit. Figure 41-4 shows the adherent powdered ZrO_2 on the outside of cups 59, 58, and 57. Figure 41-5 shows the oxide coating on the hot finger. The fluff foil insulation is also shown around the thin wall section of the lower part of the hot finger. The fluff foil was then removed. The Ti getter annular rings were brittle indicating absorption of interstitial elements such as O_2 , H_2 , and carbon. The Nb-1Zr getter rings were colored a deep blue indicating a surface oxidation reaction. The nickel expanded mesh and the nickel rings were relatively clean.

Figure 41-6 is a view looking down into cup No. 1 nested in with cups 1 through 57 after removal from the IVBM No. 1 hot finger. Note the deformation at the bottom caused by the cups resting on the raised top radius of the hot finger.

ELECTRICAL HEATER FABRICATION

Several electrical heater concepts are being evaluated to select the most reliable design for the electrically heated IVBM implantable system. Three basic systems are being investigated. All three are designed to produce 33 watts with 90 volts applied to the terminals. The first

41-8

(a)

(b)

Figure 41-3. Views of Outer Nickel Cup After Outer Container Removal Showing Discoloration Opposite Vacuum Pumping (IVBM No. 1)

Figure 41-4. Adherent ZrO_2 Powdered Spacer on Cups 59, 58, and 57
After Removal from the IVBM No. 1 Insulation Package

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Figure 41-5. IVBM No. 1 Hot Finger and Fluff Foil after the Removal of the 60 Concentric Nickel Cups

Figure 41-6. View Looking Down into the Bottom of Cup No. 1
After Removal from the IVBM No. 1 Hot Finger

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two involve wrapping Kanthal wire around an Al_2O_3 insulating form. The third concept involves the use of a sheathed heater. A mockup of the Kanthal- Al_2O_3 heater manufactured from stock ceramic shapes is being fabricated and will be tested to determine if heater wire temperatures are low enough ($< 1150^\circ\text{C}$) to insure reliable operation in a vacuum environment.

The modifications to the electrically heated hot finger involve machining the Udimet flange off of the existing head, brazing on a monel flange, and making a new monel outer container for the insulation package. The Udimet to monel braze, the general configuration of the monel flange, and the outer container have been defined. The exact locations for the thermocouple and heater feedthroughs and the vacuum pinchoff port are being checked to insure that no interference exists with the crankcase, and that proper routing of the leads can be achieved.

THERMAL CONVERTER FABRICATION

Work was initiated on the fabrication of a new IVBM nuclear hot finger. The monel flange to Udimet 700 braze joint had to be redesigned to accommodate the present Udimet stock size, and the width of the slits in the nickel heater ring was decreased to 0.005 inch to increase the heat exchanger effectiveness. This latter design change was made possible by fabrication advancements developed this year.

All parts required for the hot finger were roughed out, the Udimet was heat treated using the new Philips Laboratories heat treatment schedule, and the nickel isotope sleeve and heat exchange ring were brazed together. At that point, further effort was postponed because of the heat source problems encountered during testing and reported in Task 40.

In mid-June, work began on the construction of an implantable electric hot finger. A spare IVBM hot finger will be reworked to accomplish this. The Udimet 700 instrumentation flange will be machined off, and a monel flange brazed to the Udimet in its place. The monel flange will then be machined to the final dimensions and configuration of a nuclear hot finger. This

work has yet to be accomplished. The spare electric hot finger is presently being tested to verify vacuum integrity of the thin-wall before proceeding with the rework.

Replacement pressure transducers of a new type were purchased from Kulite, Inc. for use in IVBM testing at Philips Laboratories. The new transducers are functionally the same as the previous devices but are of a more rugged construction. During testing at Philips Laboratories, adapters are fitted to the oil and buffer space ports to permit the use of the new transducer, while the model still stocked at Westinghouse Advanced Energy Systems Division can be used in field testing. The working space fitting, however, has been permanently modified to accept the new model, and spares will accompany the engine for use in field tests.

TASK 42 - ADDED IVBM BLOOD PUMP FABRICATION

The three additional blood pumps were completed and two were shipped to the University of Utah. The third will be held at Westinghouse Advanced Energy Systems Division for life test. The electric drive motors and controllers were received and are being assembled in the implantation configuration.

TASK 43 - IVBM SYSTEM LIFE TESTING

Testing of the IVBM blood pump with silastic ventricles was performed as described under Task 38. To date, a total of 150 operating hours have been accumulated. No evidence of leakage of pump mechanism oil into the mock blood circulation loop was observed during this testing. The cause of the linear bushing failure reported under Task 38 is being investigated.

TASK 46 - RELIABILITY AND QUALITY ASSURANCE

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

APPENDIX A

DESIGN REQUIREMENTS AND PERFORMANCE OBJECTIVES
FOR
NUCLEAR POWERED ARTIFICIAL HEART
MARK I PROTOTYPE SYSTEMTABLE OF CONTENTS

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1.0 GENERAL

This specification establishes the performance objectives and design requirements for equipment identified as the Nuclear Powered Artificial Heart MARK I Prototype System. This MARK I prototype system shall be a Stirling-mechanical system, comprising: 1) a thermal energy converter based on the Stirling cycle which converts input heat from a Pu-238 heat source to rotary output shaft power, 2) a blood pumping unit, and 3) an interconnecting coupling mechanism which transmits the power from the thermal converter to the blood pump. The coupling mechanism unit may also provide routing to deliver the cycle waste heat from the thermal converter to the blood pumping unit.

2.0 PERFORMANCE OBJECTIVES

2.1 PEAK POWER OBJECTIVE

The MARK I prototype system shall have the ability to deliver varying amounts of hydraulic power to the blood in response to physiological demands, with the capability of delivering at least 3.1 watts under maximum demand at the end of its 10 year design lifetime. The heat input from the Pu-238 heat source required to meet this objective shall not exceed 30.5 watts at the end of the system lifetime (33 watts initially). The 3.1 watts peak power requirement represents a blood flow of 12 liters per minute with a mean aortic pressure of 100 mm Hg and a mean pulmonary arterial pressure of 15 mm Hg and mean atrial pressures of 0.0 mm Hg. The system shall be designed to meet these flow conditions with a blood pump speed not to exceed 120 beats per minute. The system shall be designed so that it can be adapted to accommodate higher than design blood pressures up to a mean of 150 mm Hg in the aorta by limiting the maximum blood flow capability to maintain the maximum load at 3.1 watts.

To provide guidance for the design of the major components of the system, Table A-1 presents tentative performance objectives for each major component. In each of the two sections representing initial and end of life conditions, the central column represents the power in watts

at each component. The right hand column represents the efficiency goal for each component when the system is delivering 3.1 watts to the blood. The parenthetical figures in the left hand column represent the power available at each step if the efficiency goals for each component can actually be met.

2.2 CONTROL OBJECTIVE

The system shall have the capability of varying the blood flow rate in response to physiologically induced changes in venous return pressure. The sensitivity of the system should be such that the flow can vary from one half of maximum flow to maximum flow with the mean venous return pressure varying between limits not to exceed -5 mm Hg and +15 mm Hg. For any given flow rate within the operating range, the difference between the mean venous return pressure to the left ventricle and to the right ventricle should not exceed 7.5 mm Hg, and neither pressure shall fall outside the range specified above.

The required variation in flow shall be accomplished by varying the displaced volume of the blood pump ventricles in response to venous return pressure. The converter shall be fitted with a speed control to maintain approximately constant speed under normal operating conditions. The coupling mechanism shall be designed to provide appropriate speed matching between the components of the system, so that each component can be operated at its most advantageous design operating speed.

2.3 OVERLOAD CAPABILITY

The system shall be designed so that it is capable of continuing operation with reduced power output with blood pressures up to 25 percent higher than the operating pressure. For example, if the system is set up to pump 12 l/minute with a mean aortic pressure of 100 mm Hg, it shall be capable of operating at reduced speed against a mean aortic pressure of 125 mm Hg, without stalling. If the system has been adapted to operate against a mean aortic pressure of 150 mm Hg, it shall be capable of continued operation at 188 mm Hg mean aortic pressure without stalling.

TABLE A-1
 MARK I PROTOTYPE SYSTEM COMPONENT PERFORMANCE OBJECTIVES

	INITIALLY		AFTER TEN YEARS	
• THERMAL INPUT (WATTS)		33.0		30.5
CONVERTER EFFICIENCY		21.5%		19.0%
• AVAILABLE CONVERTER OUTPUT (WATTS)		7.1		5.8
• REQUIRED CONVERTER OUTPUT (WATTS)	(7.1)	5.5	(5.8)	5.5
PUMP DRIVE EFFICIENCY		80%		80%
• POWER TO PUMP DIAPHRAGMS (WATTS)	(5.7)	4.4	(4.6)	4.4
VENTRICULAR EFFICIENCY		70%		70%
• POWER AVAILABLE TO BLOOD (WATTS)	(4.0)	3.1	(3.2)	3.1

3.0 DESIGN REQUIREMENTS

3.1 DIMENSIONS, WEIGHT AND ARRANGEMENT

The MARK I system, when assembled, shall form a smooth envelope with well rounded corners and generous fillets around all protrusions. The system components shall have the following maximum dimensions (shown schematically in Figure A-1) and weights:

Thermal Converter

Envelope	18 cm x 9 cm x 8 cm
Volume	0.8 liter
Weight	1.7 kg

Blood Pump

Envelope	13 cm x 12 cm x 9 cm
Volume	0.7 liter
Weight	0.8 kg

Coupling Mechanism

Nominally, the coupling mechanism shall cause the thermal converter and blood pump to be arranged as shown schematically in Figure A-1. An alternative arrangement with the long axis of the coupling mechanism normal to the long axis of the converter may be considered if surgically acceptable.

Envelope	a) The area of the interface with the converter shall have the envelope dimension: 2.6 cm x 4.5 cm x 4.5 cm. b) The remainder of the coupling mechanism shall fit within an elliptical cylinder 3.0 cm x 2.0 cm with a nominal length of 15 cm.
Volume	The enclosed volume of the coupling mechanism shall be minimized.
Weight	The weight of the coupling mechanism shall be minimized.

NOTE: Not to scale; Envelope dimensions in cm.

Figure A-1. Prototype System Coupling Mechanism Arrangement

3.2 WASTE HEAT REJECTION

The heat rejection system shall be designed for the transfer of the thermal converter waste heat to the blood pump via the coupling mechanism for ultimate rejection to the circulating blood. Coolant shall be supplied to the thermal converter at a temperature not to exceed 49°C (120°F) when the recipient's temperature is normal (37°C).

The reference design of the MARK I system shall be insulated to provide a heat flux no greater than 0.02 w/cm^2 through surfaces in contact with tissue, while the system is operating normally. The maximum heat flux through surfaces in contact with flowing blood shall not exceed 0.1 w/cm^2 . For calculational purposes, a surface temperature of 37°C (99°F) shall be assumed when computing heat fluxes to tissue.

3.3 CHECKOUT AND TESTING

The MARK I system shall provide sufficient instrumentation to permit testing and checkout of the system and provide the means for monitoring of parameters required to provide assessment of operational performance. The design of the system shall be such that it can be tested prior to implantation.

3.4 ENVIRONMENT

The system shall be able to operate satisfactorily in any orientation with respect to gravity. Variations in atmospheric pressure such as experienced in normal commercial plane travel, or driving to an altitude of 3000 m (10,000 ft.) shall not significantly affect the operation of the system.

The system shall be designed to operate normally while subjected to a vibration level of 0.5 g at 4 to 12 Hz with the plane of oscillation in any orientation. It must be able to withstand a 30g shock with pulse width of 0.017 seconds from any orientation without damage or stalling after implantation.

The system shall be designed to be compatible with blood, bone and tissue at contact interfaces and to be hermetically sealed from body fluids. Vibration or acoustic levels from the operating system must not endanger nor cause unacceptable discomfort to the host. Moving parts must not impose unacceptable reaction or gyroscopic forces on the host.

The system must operate satisfactorily in the presence of normally encountered external magnetic fields. The maximum magnetic field to which the system can be exposed and continue to operate reliably shall be computed.

3.5 RELIABILITY

The design reliability goal is a mean life of 10 years with a maximum failure rate of 0.01 units/year for the first five years.

3.6 HELIUM DISPOSAL

Provision shall be made to accommodate the helium released by the Pu-238 heat source pressure relief device. This gas (approximately $60 \text{ cm}^3/\text{year}$) may be vented to the body tissue or stored at a pressure not to exceed 10 atmospheres for the 10 year operating life of the system. Back diffusion of atmospheric oxygen to the heat source pressure relief device shall be prevented.

3.7 SAFETY

The MARK I system shall be designed so that no malfunction of the system will jeopardize the integrity of the Pu-238 heat source. In particular, the design must prevent overtemperaturing of the heat source in the event of failure of any component of the system.

3.8 PRE-IMPLANTATION

The MARK I system must be amenable to reasonable surgical procedures, time schedule, and ease of implantation. COO-3043-19, "Engineering Advisory Document, IVBM Calf Experiments"

outlines the definition of "reasonable surgical procedures." This document will be maintained via revisions as required to reflect current surgical and engineering philosophy. The system should be amenable to a reasonable operational shelf life after integration with the fuel capsule and check-out operating tests.

APPENDIX B

HEAT SOURCE OPERATING HISTORY

Both sources were received on January 10, 1973 and were stored in a controlled environment in the shipping container when not in use. In addition to converter testing, the sources were removed from the containers and exposed to air at ambient temperature for the following tests:

1. The uncoated source was sent to the x-ray department for one working day.
2. Each source was sent to quality control for dimensional checking (approximately three days for each source).
3. Each source was sent to the materials department for a check of helium flow using a mass spectrometer immediately prior to each converter test series. (In this check, the source was exposed to a vacuum of approximately 10^{-6} torr for several hours, with a total time out of the shipping container of one working day.)

Table B-1 lists the use of the sources in the bench model converter. During operation of the converter, the sources were exposed to a vacuum of approximately 10^{-6} torr. At the end of each run, helium was bled into the insulation package to provide a cooldown transient approximately 35 minutes long. The startup transient lasted approximately 1-1/4 hours.

The first two columns of Table B-1 show the date and time of the event noted in the "Remarks" column. The third column shows the maximum hot finger temperature reached during the experiment, while the fourth column indicates which heat source was used for a given experiment. The fifth column shows the elapsed time for each experiment. The bench model experiment history can be summarized as:

- The coated Pu-238 heat source was inserted in and removed from the bench model thermal converter five (5) times and was retained in the thermal converter for 935 hours.

TABLE B-1

BENCH MODEL THERMAL CONVERTER Pu-238 HEAT SOURCE OPERATION

Date	24 ^h Time	Temp. Max. °F	Pu238 Heat Source		Δ Time		Log Book Page Number	T/Cnv	REMARKS
			Coated	Uncoated	h	m			
5/8/73	1047		X				2060907	2	Started
5/11/73	1520	1565	X		76	33	2060915	2	Stopped
7/25/73	1020			X			2060920	2	Started
7/26/73	1350	1490		X	25		2060922	2	Stopped
8/14/73	0950		X				2060927	2	Started
8/14/73	1003		X			13	2060927	2	Stopped - Power failure on bench.
8/20/73	0950		X				2060935	2	Started
8/20/73	1225		X		2	35	2060935	2	Stopped - Roll sock leak
8/29/73	0920		X				2060935	2	Started
8/29/73	1150		X		2	30	2060935	2	Stopped - Roll sock defective
10/17/73	1000		X				2060938	2	Started
10/18/73	1630	1355	X						Stopped-Source stored in engine under He backfill overnight
10/19/73		1430	X						Restarted test
11/21/73	2115		X		853	15	2060961	2	Stopped test

- The uncoated Pu-238 heat source was inserted in and removed from the bench model thermal converter one (1) time and was retained in the thermal converter for twenty-five (25) hours.

Table B-2 lists the use of the heat sources in the IVBM converter. In this converter, the source was located inside the working volume of the converter. In this location, it was subject to an argon environment at a mean pressure of approximately 180 psig, with a fluctuating pressure wave of ± 30 psig at approximately 10 Hz superimposed. The heatup transient lasted approximately 1-1/4 hours in the IVBM converter. At the end of testing, the argon pressure was normally reduced to atmospheric pressure over a period of approximately 10 minutes. The hot heat source was then removed from the converter. Prior to February 17, 1976, the hot sources were exposed directly to room temperature argon for cooldown. After this date, the sources were placed in a preheater oven, and cooled down over a period of approximately six hours, with a maximum cooldown rate of 360°F/hour.

- The coated Pu-238 heat source was inserted in and removed from the IVBM model thermal converter eight (8) times and was retained in the thermal converter for 105 hours.
- The uncoated Pu-238 heat source was inserted in and removed from the IVBM model thermal converter eighteen (18) times and was retained in the thermal converter for a total of seventy-three (73) hours.

After failures on May 11, 1976, both heat sources were stored in an air environment in the shipping containers until May 21, 1976, when they were turned over to LASL personnel for shipment to LASL.

TABLE B-2

IVBM THERMAL CONVERTER Pu-238 HEAT SOURCE OPERATION

Date	24 ^h Time	Temp. Max. °F	Pu238 Heat Source		Δ Time		Log Book Page Number	T/Cnv	REMARKS
			Coated	Uncoated	h	m			
4/23/75	1530			X			151049-1	IVBM	Test Pu238 Fit. Started
4/23/75	1843	693		X	3	13			Stopped
4/24/75	1110			X					Started
4/24/75	1224	675		X	1	10	-4		Stopped
5/16/75	1036			X			-6		Started
5/16/75	1341	1315		X	3	05			Stopped
7/14/65	1055			X			-10		Started - Heat source stuck going in
7/14/75	1058			X					
7/14/75	1504	1375		X	4	09			Stopped
7/17/75	0946			X					Started
7/17/75	1308	1320		X	3	22	-13		Stopped
8/13/75	0922			X					Started
8/13/75	1205	1480		X	2	43	-14		Stopped
8/15/75	1158			X					Started
8/15/75	1504	1400		X	3	06	-16		Stopped
10/23/75	0925			X					Started
10/23/75	1105	930		X	1	35	-18		Stopped
10/24/75	0851			X			151049-19	IVBM	Started
10/24/75	1415	1522			5	24			Stopped

TABLE B-2 (CONTINUED)

Date	24 ^h Time	Temp. Max. °F	Pu238 Heat Source		Δ Time		Log Book Page Number	T/Cnv	REMARKS
			Coated	Uncoated	h	m			
10/31/75	0940			X					Source in engine.
10/31/75	1240	1282		X			151049-20	IVBM	Source removed from engine-no running.
11/4/75	0940								Started
11/4/75	1515	1400		X	5	35	-21		Stopped
11/7/75	0913								Started
11/7/75	1422	1360		X	5	09	-23		Stopped
11/11/75	0841			X					Started
11/11/75	1427	1452			5	46	-24		Stopped
11/12/76	1030			X					Started
11/12/76	1600	1323			5	30	-26		Stopped
11/14/75	0855								Started
11/14/75	1200	1404		X	3	05	-28		Stopped
12/12/75	0920								Started
12/12/75	1740	1496		X	8	20	-29		Stopped
12/16/76	0945								Started
12/16/75	1545	1360		X	6	--	-30		Stopped
12/17/75	0920			X			-31		Started
12/17/75	1300	1390			3	40			Stopped-Radioactivity on source at defueling.
2/17/76 *	0908		X						Started
2/17/76	1430				5	22	-37		Stopped
2/18/76	1031	1522	X				-38		Started
2/19/76	1505	1490			28	24			Stopped
3/2/76	0933		X						Started Sys. Test-Pumped vacuum insulation
3/2/76	1455	1380			5	22	-40		Stopped
3/3/76	1320	1455	X				-41		Started Sys. Test-Pumped vacuum insulation
3/4/76	1529	1460			26	09	-42		Stopped
3/30/76	0926	1520	X						Started Sys. Test- Static vacuum insulation
3/31/76	1150				25	24	-44		Stopped
4/1/76	0922		X						Started - Pumped vacuum insulation
4/1/76	1525	1463			6	03	-46		Stopped
4/30/76	0925		X						Started Sys. Test- Static vacuum insulation
4/30/76	1415	1431			4	50	-47		Stopped

*At this date and thereafter, source was always placed in oven for controlled cooling after defueling.

TABLE B-2 (CONTINUED)

Date	24 ^h Time	Temp. Max. °F	Pu 238 Heat Source		Δ Time		Log Book Page Number	T/Cnv	REMARKS
			Coated	Uncoated	h	m			
5/11/76	1010		X		2	20	-48	IVBM	Started
5/11/76	1230								Stopped - Roll sock ruptured - heat source lost cap.
5/11/76	2000			X	1	33			Started
5/11/76	2133			X					Stopped - heat source lost cap.

TOTALS: Bench Model 960
IVBM Model 176.4