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Advanced Energy Systems Division

ARTIFICIAL HEART DEVELOPMENT PROGRAM

VOLUME II, SYSTEM SUPPORT PHASE III SUMMARY REPORT

July 1, 1973

Through

September 30, 1977

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ARTIFICIAL HEART
DEVELOPMENT PROGRAM

PHASE III
SUMMARY REPORT - VOLUME II
APPENDICES ON SYSTEM SUPPORT

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JULY 1, 1973 THROUGH SEPTEMBER 30, 1977

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

PREPARED FOR

UNITED STATES DEPARTMENT OF ENERGY

UNDER CONTRACT NO. EY-76-C-02-3043*.000

APPENDIX A (VOLUME II)

SYSTEM SUPPORT

Appendix A covers major activities of the Artificial Heart Development program that supported the design, fabrication, and test of the system demonstration units.

Section A.1.0 provides a listing beyond that of the body of the report on the components needed for an implantation. It also presents glove box sterilization calibration results and results of an extensive mock circulation calibration.

Section A.2.0 provides detail procedures for assembly, preparing for use, and the use of the system and major components.

Section A.3.0 covers the component research and development activities undertaken to improve components of the existing system units and to prepare for a future prototype system.

Section A.4.0 provides a listing of the top assembly drawings of the major systems variations fabricated and tested

APPENDIX A (VOLUME II)

TABLE OF CONTENTS

		<u>Page</u>
A.1.0	SUPPORT EQUIPMENT	A.1-1
A.1.1	EQUIPMENT LIST	A.1-1
A.1.1.1	Blood Pump Assembly and Equipment Recovery Equipment	A.1-1
A.1.1.2	First Stage Implantation Equipment List	A.1-2
A.1.1.3	Thermal Converter Field Assembly Equipment List	A.1-2
A.1.1.4	Thermal Converter Implantation Equipment List	A.1-3
A.1.1.5	Post-Operative Thermal Converter Monitoring Equipment List	A.1-3
A.1.1.6	Thermal Converter Recovery Equipment List	A.1-3
A.1.2	CALIBRATIONS	A.1-4
A.1.2.1	Sterilization	A.1-4
A.1.2.2	Mock Circulation (Loop) Calibration	A.1-7
A.2.0	EXPERIMENTAL PROCEDURES	A.2-1
A.2.1	THERMAL CONVERTER STERILIZATION	A.2-1
A.2.2	BLOOD PUMP MECHANISM DISASSEMBLY AND ASSEMBLY	A.2-4
A.2.2.1	Mechanism Disassembly	A.2-4
A.2.2.2	Mechanism Assembly	A.2-4
A.2.2.3	Field Assembly of Blood Pump and Coupling Mechanism	A.2-5
A.2.3	PUMP VENTRICLE INSTALLATION AND SEATING	A.2-7
A.2.3.1	Ventricle Installation	A.2-7
A.2.3.2	Ventricle Seating	A.2-8
A.2.3.2.1	Dry Blood Pump with Pinned Ventricles	A.2-8
A.2.3.2.2	Oil-Filled Blood Pump with Pinned Ventricle	A.2-8
A.2.3.2.3	Dry Blood Pump with Free-Type Ventricles	A.2-8
A.2.3.2.4	Oil-Filled Blood Pump with Free-Type Ventricles	A.2-9
A.2.3.2.5	Oil Filled ΔV Blood Pump with Pinned Ventricles	A.2-9
A.2.3.2.6	Oil Filled ΔV Blood Pump with Free-Type Ventricles	A.2-10

TABLE OF CONTENTS (Continued)

	<u>Page</u>
A.2.4	THERMAL CONVERTER PROCEDURES A.2-10
A.2.4.1	Electrical Thermal Converter Preparation A.2-10
A.2.4.2	Electrical Thermal Converter Pretest A.2-11
A.2.4.3	Electrical Thermal Converter Assembly A.2-11
A.2.4.4	Electrical Thermal Converter Startup A.2-12
A.2.4.5	Electrical Thermal Converter Shutdown A.2-13
A.2.5	IMPLANTATION PROTOCOLS A.2-14
A.2.5.1	First Stage Implantation A.2-14
A.2.5.1.1	Calf Pre-Operative Workup A.2-14
A.2.5.1.2	Field Assembly of Pump A.2-14
A.2.5.1.3	Blood Pump Implantation A.2-14
A.2.5.1.4	Post Operative Care A.2-16
A.2.5.1.5	Post-Operative Equipment Monitoring A.2-16
A.2.5.2	Second Stage Implantation A.2-16
A.2.5.2.1	Calf Pre-Operative Workup A.2-16
A.2.5.2.2	Thermal Converter Check-Out and Sterilization A.2-16
A.2.5.2.3	Thermal Converter Implantation A.2-17
A.2.5.2.4	Post-Operating Care and Monitoring A.2-18
A.2.5.2.5	Equipment Recovery Procedures A.2-18
A.2.5.2.5.1	Thermal Converter Recovery Procedure A.2-18
A.2.5.2.5.2	Blood Pump Recovery Procedure A.2-19
A.3.0	COMPONENT RESEARCH AND DEVELOPMENT A.3-1
A.3.1	STIRLING ENGINE A.3-1
A.3.1.1	Hot Finger Material Investigation A.3-1
A.3.1.2	Metals Joining Investigations A.3-26
A.3.1.3	Engine Control System Development A.3-38
A.3.1.4	Magnetic Seals Investigation A.3-81
A.3.1.5	Lubrication System Development A.3-153
A.3.1.6	Bellows Piston Seal Development A.3-189
A.3.1.7	"Y" Piston Ring A.3-196
A.3.2	THERMAL INSULATION A.3-199
A.3.2.1	High Temperature Vacuum Foil Insulation A.3-199
A.3.2.2	Low Temperature Protection Insulation A.3-275

TABLE OF CONTENTS (Continued)

	<u>Page</u>
A.3.3 INTERCONNECTING SHAFTS	A.3-282
A.3.3.1 Wire Wound Shafts	A.3-282
A.3.3.2 Other Torque Transmissions Shaft Investigations	A.3-301
A.3.4 BLOOD PUMP MECHANISM	A.3-318
A.3.4.1 Oils and Ventricle Material Compatibility	A.3-318
A.3.4.2 Wear Tests	A.3-321
A.3.4.3 Bearing Friction and Life	A.3-323
A.3.4.4 Mechanism Endurance Test	A.3-327
A.3.4.5 Drive Mechanism Power Losses	A.3-331
A.3.4.6 Magnesium and Beryllium Corrosion Investigation	A.3-340
A.3.4.7 Mechanism - Ventricles Interface Investigations	A.3-360
A.3.4.8 Pump Size - RPM Tradeoff Analysis	A.3-363
A.3.4.9 Analysis of Linear Bearing Failure	A.3-373
A.3.5 WASTE HEAT REJECTION	A.3-378
A.3.5.1 Bellows Pump Water System	A.3-378
A.3.5.2 Thermal Oscillation Pump System	A.3-396
A.3.5.3 Heat Pipes	A.3-417
A.3.5.4 ΔV Pusher Cup Oil System	A.3-432
A.3.5.5 Gear Pump Oil System	A.3-434
A.3.5.6 Other Methods	A.3-440
A.4.0 TOP ASSEMBLY DRAWINGS LIST	A.4-1

APPENDIX A (VOLUME II)

LIST OF FIGURES

<u>Figure</u>		<u>Page</u>
A.1-1	Mock Loop Flowmeter Calibration Test Setup	A.1-1
A.1-2	Pulmonary Error Versus Flow (<u>W</u> Brush Recorder Data) at Fixed Valve Position	A.1-13
A.1-3	Pulmonary Error Versus Flow (<u>W</u> Digital Data) at Fixed Valve Positions	A.1-14
A.1-4	Pulmonary Error Versus Flow (Utah Brush Recorder Data) at Fixed Valve Positions	A.1-15
A.1-5	Pulmonary Error Versus Flow (Utah Digital Data) at Fixed Valve Positions	A.1-16
A.1-6	Systemic Error Versus Flow (<u>W</u> Brush Recorder Data) at Fixed Valve Positions	A.1-17
A.1-7	Systemic Error Versus Flow (<u>W</u> Digital Data) at Fixed Valve Positions	A.1-18
A.1-8	Systemic Error Versus Flow (Utah Brush Recorder Data) at Fixed Valve Positions	A.1-19
A.1-9	Systemic Error Versus Flow (Utah Digital Data) at Fixed Valve Positions	A.1-20
A.1-10	Pulmonary Side ΔP Versus Measured Flow at Fixed Valve Positions for Pulmonary Valve	A.1-22
A.1-11	Systemic Side ΔP Versus Measured Flow at Fixed Valve Positions for Systemic Valve	A.1-23
A.3-1	Rupture Stress Versus Larson-Miller Parameter for Udimet 700	A.3-3
A.3-2	Thermal Conductivity of Udimet 700	A.3-5
A.3-3	Udimet Bar No. 1 As-Received (100X)	A.3-5a.
A.3-4	Bar No. 1 After Heat Treat (100X)	A.3-5a.
A.3-5	Udimet Bar No. 2 As-Received (100X)	A.3-6
A.3-6	Bar No. 2 After Heat Treat (100X)	A.3-6
A.3-7	Powdered Udimet As-Received (100X)	A.3-8
A.3-8	Powdered Udimet After Heat Treat (100X)	A.3-8
A.3-9	Heat Treatment No. 3 (100X)	A.3-9
A.3-10	Heat Treatment No. 4 (100X)	A.3-9
A.3-11	Heat Treatment No. 4 (100X)	A.3-10
A.3-12	Heat Treatment No. 5 (100X)	A.3-10

LIST OF FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
A.3-13	Schematic of High Temperature Stress Rupture Approach	A.3-12
A.3-14	Thin-Walled Udimet 700 Stress-Rupture Specimen	A.3-13
A.3-15	SEM Micrograph of the Inside of Specimen P/M 800	A.3-15
A.3-16	Enlargement of the Hole Shown in Figure A.3-15	A.3-16
A.3-17	Thin-Wall Thickness Gaging System	A.3-18
A.3-18	Ground Udimet 700 Test Specimen After Creep-Rupture Test	A.3-22
A.3-19	Percent of Thick-Section Udimet 700 Creep-Rupture Life Versus Minimum Wall Thickness	A.3-24
A.3-20	Sample Braze Joint	A.3-27
A.3-21	35 Power Micrograph of Nickel-Udimet	A.3-28
A.3-22	270 Power Micrographs of Nickel-Udimet	A.3-28
A.3-23	Titanium Alloy/Titanium Alloy Braze Joint, 400X	A.3-31
A.3-24	Titanium Alloy/Udimet 700 Braze Joint, 400X	A.3-31
A.3-25	Fe-Cu Laminated Pole Face Structure (40 X)	A.3-33
A.3-26	Fe-Cu Laminated Pole Face Structure (400 X)	A.3-33
A.3-27	Electroless Nickel on 6Al-4V Titanium Alloy	A.3-34
A.3-28	Electroless Nickel on Udimet 700	A.3-35
A.3-29	Oil Filter Assembly Weld Specimen	A.3-36
A.3-30	Seam Weld of Mesh Pleat	A.3-36
A.3-31	Generalized Proportional Control System	A.3-41
A.3-32	Control System Block Diagram for Blood Pump Torque Input	A.3-42
A.3-33	Control Valve Mod 0	A.3-44
A.3-34	Short-Circuit Control Valve Performance	A.3-46
A.3-35	Flow Through Short-Circuit Control Valve Versus Poppet Opening	A.3-48
A.3-36	Lead Screw Assembly	A.3-50
A.3-37	Bellows Assembly	A.3-50
A.3-38	Parts Layout	A.3-50
A.3-39	Completed Valve	A.3-50
A.3-40	Hysteresis Data on Control Valve Mod No. 1	A.3-52
A.3-41	Control System Stability Test Apparatus	A.3-53

LIST OF FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
A.3-42	Control System Transient Characteristics	A.3-54
A.3-43	Control System Engine Power Versus Speed	A.3-57
A.3-44	Pressure Drop Test Schematic	A.3-58
A.3-45	Engine with Valve Pressure Drop	A.3-59
A.3-46	Engine Pressure Drop Data	A.3-60
A.3-47	Pressure Drop Data - Valve With Fittings	A.3-63
A.3-48	Theoretical Flow Rate Data on Control Valve Characteristics	A.3-65
A.3-49	Experimental Data on Control Valves #1 and #2	A.3-66
A.3-50	Typical Change in Engine Output Power Versus Speed at Constant Temperature	A.3-67
A.3-51	Phase Shift Data	A.3-68
A.3-52	Control System Response to Step Inputs	A.3-71
A.3-53	Control System Response at Low Engine Speed	A.3-72
A.3-54	Combined Control Valve and Engine Hysteresis	A.3-74
A.3-55	IVBM Engine Short-Circuit Phase Shift Data	A.3-75
A.3-56	IVBM Control Valve Bellows Defect 1000X	A.3-76
A.3-57	Control Schematic - Flow Losses	A.3-77
A.3-58	Control Schematic - Flywheel Use	A.3-78
A.3-59	Temperature Compensated Speed Control Valve	A.3-80
A.3-61	Magnetic Coupling/Slip Clutch Characteristics	A.3-84
A.3-62	Baseline Magnetic Coupling	A.3-85
A.3-63	Mod 0 Experimental Magnetic Coupling	A.3-87
A.3-64	Titanium Membrane	A.3-87
A.3-65	Magnetic Coupling Test Fixture	A.3-88
A.3-66	Coupling Torque Data Versus Angular Displacement	A.3-90
A.3-67	Peak Torque Data and Energy Storage Versus Pole Face Width	A.3-91
A.3-68	Peak Torque Data Versus Radial Air Gap	A.3-92
A.3-69	Mod 1 Experimental Magnetic Coupling Design	A.3-93
A.3-70	Mod 1 Magnetic Coupling	A.3-95
A.3-71	Torque Versus θ With Large and Small Magnets	A.3-96

LIST OF FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
A.3-72	Torque Versus θ and Radial Air Gaps	A.3-98
A.3-73	Peak Torque and Energy Versus Radial Air Gap	A.3-99
A.3-74	Magnetic Coupling Configuration	A.3-100
A.3-75	Magnetic-Field Leakage Measurements	A.3-103
A.3-76	Eddycurrent Loss Parameter Study	A.3-104
A.3-77	Buckling Pressure Versus Wall Thickness and Length for Titanium-Alloy Membrane	A.3-106
A.3-78	Structural Test Titanium Membrane After Buckling	A.3-107
A.3-79	Magnetic Coupling Torque Characteristics	A.3-111
A.3-80	Magnetic Coupling Experimental Test Stand	A.3-112
A.3-81	Ribbed-Prototype Structure: Magnetic Coupling Membrane	A.3-114
A.3-82	Sandwich-Membrane Structure	A.3-116
A.3-83	Magnetic Fluid Piston Seal	A.3-117
A.3-84	Magnetic Fluid Piston	A.3-119
A.3-85	Magnetic Fluid "O-Rings" On Test Piston	A.3-119
A.3-86	Oscilloscope Traces of Pressure Wave Form Above Piston With and Without Magnetic Fluid	A.3-121
A.3-87	Linear Magnetic Fluid Seal Concept	A.3-123
A.3-88	Linear Magnetic Fluid Seal Feasibility Test Fixture	A.3-124
A.3-89	Magnetic Fluid Miscibility Tests	A.3-126
A.3-90	Rotary Magnetic Fluid Shaft Seal	A.3-129
A.3-91	Prototype Magnetic Fluid Shaft Seal	A.3-130
A.3-92	Prototype Rotary Shaft Seal	A.3-92
A.3-93	Rotary Shaft Seal Test Fixture - Seal End	A.3-131
A.3-94	Rotary Shaft Seal Test Fixture - Motor End	A.3-132
A.3-95	Instrumented Rotary Shaft Seal Test Fixture	A.3-133
A.3-96	Rotary Test Seal Geometry	A.3-134
A.3-97	Shaft Seal Pressure Capability	A.3-135
A.3-98	Eddycurrent Losses	A.3-136
A.3-99	Resealing Seal Data	A.3-137
A.3-100	Magnetic Fluid Shaft Seal Pressure Capability With Argon	A.3-139

LIST OF FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
A.3-101	Magnetic Fluid Shaft Seal Pressure Capability With Magnetic Fluid	A.3-140
A.3-102	Two-Pole Shaft Seal Pressure Capability	A.3-142
A.3-103	Four-Pole Shaft Seal Pressure Capability	A.3-144
A.3-104	TEM Photograph of Tellus Magnetic Fluid	A.3-147
A.3-105	Temperature-Viscosity Data for Pella 11 Magnetic Fluid	A.3-148
A.3-106	Magnetometer Photograph and Calibration Curve	A.3-149
A.3-107	Magnetic Fluid Behavior as a Function of Shear Rate	A.3-151
A.3-108	Viscosity of Magnetic Fluid and Mineral Oil at High Rates of Shear	A.3-152
A.3-109	IVBM Gear Pump	A.3-156
A.3-111	Experimental Results With IVBM Gear Pump #3	A.3-159
A.3-112	Three-Gear Oil Pump for IVBM	A.3-160
A.3-113	Performance Data for 3-Gear Pump	A.3-162
A.3-114	3/16-in. Pitch Diameter Gear Pump	A.3-163
A.3-115	Gear Pump Performance Comparison	A.3-165
A.3-116	Thermal Converter/Blood Pump Seal Test Apparatus	A.3-167
A.3-117	Output Shaft Seal Configured With Lip Seal	A.3-169
A.3-118	Screw Seal Geometry	A.3-170
A.3-119	Screw Seal Data	A.3-171
A.3-120	Seal Pressure Capability as a Function of Number of Grooves	A.3-173
A.3-121	Optimized Screw Seal Pressure Versus Thread Depth and Eccentricity	A.3-174
A.3-122	Blood Pump Screw Seal Performance for 3/16 and 1/4 Shaft Sizes	A.3-176
A.3-123	Output Shaft Screw Seal Pressure Capability	A.3-178
A.3-124	Capillary Wick Test Setup	A.3-180
A.3-125	Apparatus to Determine Flow Resistance of Intake Screens	A.3-183
A.3-126	Mesh Resistance Versus Time (With Air Bubbles)	A.3-186
A.3-127	Resistance Versus Time (With Air Bubbles but No Mesh)	A.3-187
A.3-128	Magnetic Oil Pump Intake	A.3-188

LIST OF FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
A.3-129	Bellows Tester Design	A.3-191
A.3-130	Bellows Parameters	A.3-193
A.3-131	Bellows Configurations	A.3-193
A.3-132	Bellows Seal Test Fixture	A.3-194
A.3-133	"Y" Piston Ring	A.3-196
A.3-134	Piston Pressure Waveform	A.3-198
A.3-135	Waveform With Bleed Holes	A.3-198
A.3-136	Scanning Electron Micrographs of an Electroformed Nickel Cup Cracked During Mandrel Dissolution (560X) (Gar Cup No. MD21)	A.3-201
A.3-137	Scanning Electron Micrographs of an Electroformed Nickel Cup Pinholed During Mandrel Dissolution (1560X) (Gar Cup No. MD41)	A.3-202
A.3-138	Metallographic Cross-Section of Gar Electroformed Nickel Foil with Apparent Second Phase at Surface Adjacent to Mandrel. (Etchant dil. 5% H ₂ O ₂) (Gar Cup No. MD21)	A.3-203
A.3-139	Metallographic Cross-Sections of Gar Electroformed Nickel Foil Showing Effect of Vacuum Annealing. (500X, Etchant dil. HCl-5% H ₂ O ₂) (Gar Cup No. MD42)	A.3-204
A.3-140	Metallographic Cross-Section of Gar Electroformed Nickel Cup No. MD41 Showing Residual Traces of Apparent Second Phase (See Figure 137 for SEM of surfaces). (2000X, Etchant Electrolytic 5% H ₂ SO ₄)	A.3-205
A.3-141	Metallographic Cross-Sections of Gar Electroformed Nickel Cups No. MD41 Showing Possible Progression of Pinhole Formation From Outer to Inner Surface (500X Unetched)	A.3-206
A.3-142	Appearance of Electroformed Nickel Foil Prepared at the Westinghouse Research Laboratories	A.3-208
A.3-143	Appearance of Gar Electroformed Nickel Foil From a Later Shipment of Cups (Gar Cup No. GA17C)	A.3-209
A.3-144	Typical Ion Microprobe Trace on a Cross-Section from Gar Electroformed Nickel Cup No. MD42	A.3-211
A.3-145	Scanning Electron Micrographs (SEM) Showing the Effect of Temperature on Electroplate Ni Cups (a-b)	A.3-214
A.3-145	(c-d) - Continued	A.3-215
A.3-145	(e-f) - Continued	A.3-216

TABLE OF FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
A.3-146	Square Wire Mesh Stainless Steel Masks	A.3-221
A.3-147	ZrO ₂ Plasma Sprayed onto Ni Foil Through 50 and 80 Mesh Screens	A.3-223
A.3-148	30 Mil Diameter Plasma Sprayed Dot Patterns After 2 Passes (two coats of ZrO ₂) (a-c)	A.3-227
A.3-148	(d-f) Continued	A.3-228
A.3-149	Cups as Plasma Sprayed Through Wire Mesh Spacers	A.3-229
A.3-150	Time-Temperature History 20 Cup Foil Life Test	A.3-231
A.3-151	Effect of the Life Test Exposure on the Heat Loss of the 20 Cup Insulation Package as a Function of Temperature	A.3-232
A.3-152	Guard Heater Hot Finger After Removal of the 20 Cup Pack Showing the Points Where Sintering Occurred	A.3-233
A.3-153	Dipping Technique Used to Coat Nickel Foil Cups for 40 Pack	A.3-235
A.3-154	As-coated Nickel Foil Cups Prior to Assembly of the 40 Cup Pack Test	A.3-237
A.3-155	Bottom View of 40 Cup Pack Test After Assembly	A.3-238
A.3-156	Several Cups after Disassembly of a Trial Assembly Showing that ZrO ₂ was not Scratched or Removed during Assembly	A.3-239
A.3-157	Total Heat Loss as a Function of Temperature for the Nickel Foil Cup Pack Tests	A.3-242
A.3-158	Total Heat Loss vs. Temperature-Min-K 2002 Calibration Test Runs	A.3-244
A.3-159	Effect of Test System on Heat Loss of 40 and 5 Cup Packages	A.3-248
A.3-160	Correction Curve to Rationalize Heat Losses Between the Mark III* and the IV* Heaters	A.3-249
A.3-161	Effect of the Emissivity of the Outer Cup on the Heat Loss of a 5 Cup Pack on System No. 2	A.3-250
A.3-162	Comparison of Min-K-2002 Calibration Insulation After Rebuilding Test System No. 1	A.3-353
A.3-163	Comparison of 40 Cup Pack System After Rebuilding Test System No. 1	A.3-254
A.3-164	Effect of Various Outer Cups on the Heat Losses of the 7 Cup Pack	A.3-256

LIST OF FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
A.3-165	Effect of the Emissivity of the Outer Cup on the Heat Loss of a 5 Cup Pack on Test System No. 2	A.3-257
A.3-166	Graphite Dot Sprayed Onto an Outer Cup for Emittance and Temperature Measurements	A.3-258
A.3-167	Layout Drawing of the New Scram System	A.3-263
A.3-168	View of Helium Backfill System Component Parts	A.3-264
A.3-169	Assembled Cutting Mechanism with Capillary Tube in Position	A.3-265
A.3-170	Typical Polyethylene Insulation Test Sample	A.3-278
A.3-171	Thermal Conductance Testing of Polyethylene Foam Insulated IVBM Coolant Tubing under Simulated In Vivo Conditions	A.3-279
A.3-172	Apparatus Arrangement for Thermal Conductance Measurements of Insulated IVBM Coolant Tubing	A.3-280
A.3-173	Measured Heat Losses Through Polyethylene Foam Insulated IVBM Coolant Tubing	A.3-281
A.3-174	In Vivo Experiments External Motor	A.3-283
A.3-175	Motor for Abdominal Implant	A.3-285
A.3-176	Flexible Shaft Test with Blood Pump Load/Laboratory Motor	A.3-287
A.3-177	Flexible Shaft Test with Blood Pump Load/Implantable Motor	A.3-289
A.3-178	Flexible Shaft Test with Hysteresis Brake Load/Laboratory Motor	A.3-290
A.3-179	Shafts 1 and 2 Initial Values	A.3-292
A.3-180	Shaft 1 Torque Change	A.3-293
A.3-181	Shaft No. 1 After Test	A.3-295
A.3-182	As Received Shaft Compliance (Torsional Stiffness)	A.3-297
A.3-183	Universal Joint Flexible Shaft Assembly	A.3-302
A.3-184	Universal Joint Test Setup	A.3-304
A.3-185	Test Setup for Universal Joints No. 2 and 3	A.3-305
A.3-186	Universal Joint Flexible Shaft Test Setup	A.3-306
A.3-187	Curtis Universal Joint Test Setup with Wick Lubrication	A.3-309
A.3-188	Servometer SC-7 Bellows Coupling No. 1 After Failure	A.3-310

LIST OF FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
A.3-189	Bellows Coupling Shaft after Bellows Failure	A.3-312
A.3-190	Titanium Shaft Test Setup	A.3-314
A.3-191	Failure Site of Titanium Flexible Shaft	A.3-315
A.3-192	Titanium Flexible Shaft and Casing	A.3-317
A.3-193	Ventricle Specimen Oil Diffusion Test Setup	A.3-320
A.3-194	Bearing Test Apparatus	A.3-325
A.3-195	Life Test Setup	A.3-328
A.3-196	Evoloid Gear After 6321 Hours	A.3-329
A.3-197	Evoloid Pinion After 6321 Hours	A.3-330
A.3-198	Oil Filled Mechanism Test Stand	A.3-332
A.3-199	Blood Pump Mounted on Test Stand	A.3-333
A.3-200	Gear Set Test Stand	A.3-336
A.3-201	Planetary Gear Set Mounted on Test Block	A.3-337
A.3-202	Effect of Oil in Pump Drive on No-Load Input Power	A.3-339
A.3-203	Gear Set Power Loss Versus Input Power	A.3-341
A.3-204	Corrosion of Coated Magnesium Specimens Caused by 1 to 9 Days of Exposure in 1 Percent Salt Water Held at 100°F	A.3-344
A.3-205	Corrosion of Magnadize Coated Specimens Caused by 21 Days Exposure in 1 Percent Salt Water Held at 100°F	A.3-345
A.3-206	Corrosion of Magnesium Alloy ZK60A in 1 Percent Salt (NaCl) Water Held at 100°F	A.3-246
A.3-207	Magnesium Specimen No. 1 Brush Coated with Silastic Adhesive Type A. Note the presence of coating defects caused by air entrapment	A.3-248
A.3-208	Silastic Type A Adhesive and Vulcanized Sheeting Coated Magnesium (Specimen No. 26) Exposed in Saline Solution	A.3-249
A.3-209	Vulcanized Silastic Sheeting Coated Magnesium Exposed in Saline Solution	A.3-351
A.3-210	Magnesium Coated with 0.020-in. Thick Silastic Sheeting and Exposed to Saline Solution	A.3-351
A.3-211	Magnesium Specimens Coated with 0.040 and 0.060-in. Thicknesses of Silastic Sheeting and Exposed to Saline Solution. 0.040-in. Sheeting Thickness on Spec. No. 18; 0.060-in. Thickness of Spec. 17	A.3-353

LIST OF FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
A.3-212	A Photomicrograph Showing the Cross Section of Nedox U-4 Coated Beryllium	A.3-355
A.3-213	Nedox U-4 Coated Beryllium Corrosion Test Specimen	A.3-356
A.3-214	Nedox SF-2 Coated Beryllium Corrosion Test Specimen	A.3-357
A.3-215	The Progressive Corrosion of Nedox SF-2 Coated Beryllium Caused by Exposure to 1 Percent Salt Water Held at 100°F	A.3-358
A.3-216	The Progressive Corrosion of Nedox U-4 Coated Beryllium Caused by Exposure to 1 Percent Salt Water Held at 100°F	A.3-359
A.3-217	Yoke After Vacuum Test	A.3-361
A.3-218	Crankshaft After Vacuum Test	A.3-362
A.3-319	Blood Pump Mechanism with Unrestrained Ventricle Diaphragms	A.3-364
A.3-220	Unrestrained Ventricle Diaphragm Test Rig	A.3-365
A.3-221	Unrestrained Ventricle Diaphragm in Full-In and Full-Out Position	A.3-366
A.3-222	Cardiac Output as a Function of Venous Pressure and Speed for Present Pump	A.3-367
A.3-223	Cardiac Output as a Function of Venous Pressure and Speed for 100 cc Pump	A.3-368
A.3-224	Effect of Variation of Flow Parameters on Performance for 111 cc Pump at 130 B/M	A.3-369
A.3-225	Effect of Variation of Flow Parameters on Performance for 111 cc Pump at 110 B/M	A.3-370
A.3-226	Construction of the Linear Blood Pump Bearing (4.5X)	A.3-374
A.3-227	The Failed Linear Bearing	A.3-375
A.3-228	Outer Sleeve Cracking of a 1975 Vintage Linear Bearing	A.3-377
A.3-229	Converter Coolant System Schematic Diagram	A.3-379
A.3-230	Bellows Coolant Pump	A.3-383
A.3-231	Bellows Coolant Pump Failure Site	A.3-384
A.3-232	Damaged Blood Pump Internal Coolant Line	A.3-385
A.3-233	Modified Blood Pump Internal Coolant Line	A.3-386
A.3-234	Bellows Pump Coolant Circulating System Test Setup	A.3-389
A.3-235	Flexible Coolant Tube with Axial Crack	A.3-390
A.3-236	IVBM Bellows Coolant Loop Test Setup	A.3-392

LIST OF FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
A.3-237	Coolant-Blood Pump Temperature Drop	A.3-394
A.3-238	Temperature Drop Across Blood Pump Diaphragm	A.3-395
A.3-239	Thermal Oscillation Pump Concept Schematic	A.3-397
A.3-240	Thermal Oscillating Pump (TOP) Heat Transport System Test Configuration	A.3-403
A.3-241	Methanol Subcooled Boiling Mode with Transition into Nucleate Boiling Mode	A.3-404
A.3-242	Subcooled Boiling Mode with Transition into Nucleate Boiling Mode	A.3-405
A.3-243	Range of Condenser Area Versus Power Level	A.3-407
A.3-244	TOP Component Board	A.3-409
A.3-245	Temperature Difference from the Condenser Exit Fluid (Methanol) to Water in the Mock Loop Versus Heater Power	A.3-410
A.3-246	Temperature Difference from the Condenser Exit Fluid (Acetone) to Water in Mock Loop Versus Heater Power	A.3-411
A.3-247	Schematic of TOP System/Implantable Thermal Converter Interface	A.3-412
A.3-248	Dual Loop Thermal Oscillating Pump (TOP) System Schematic	A.3-418
A.3-249	Longitudinal Section of Hughes Flexible Heat Pipe	A.3-419
A.3-250	Transverse Section of Hughes Flexible Heat Pipe	A.3-419
A.3-251	Oblique Section of Hughes Flexible Heat Pipe	A.3-420
A.3-252	Breadboard Heat Pipe	A.3-422
A.3-253	Thermal Performance of Breadboard Heat Pipe (Straight)	A.3-424
A.3-254	Thermal Performance of Breadboard Heat Pipe (5-in. Radius Bend in Bellows)	A.3-425
A.3-255	Prototype Heat Pipe Heat Rejection System	A.3-426
A.3-256	Thermal Performance of Left Hand Heat Pipe with Pipe Straight	A.3-427
A.3-257	Thermal Performance at Left Hand Pipe with Pipe Bent	A.3-428
A.3-258	Thermal Performance of Right Hand Heat Pipe with Pipe Straight	A.3-429
A.3-259	Thermal Performance of Right Hand Heat Pipe with Pipe Bent	A.3-430

LIST OF FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
A.3-260	ΔV Pusher Cup Oil System	A.3-433
A.3-261	Temperature Rise through Thermal Converter as a Function of Hydraulic Pumping Power	A.3-436
A.3-262	Temperature Rise through Thermal Converter as a Function of Pressure Drop	A.3-437
A.3-263	Temperature Rise through Thermal Converter Versus Flow Rate	A.3-438
A.3-264	Film Temperature Drop as a Function of Fin Height	A.3-441
A.3-265	Pressure Drop as a Function of Fin Height	A.3-442
A.3-266	Gear Pump Envelope where Gears are Supported by Ball Bearings	A.3-443
A.3-267	Gear Pump Envelope where Gears are Supported by Journal Bearings	A.3-444
A.3-268	Schematic of Pumped Loop Waste Heat Rejection	A.3-448

APPENDIX A (VOLUME II)

LIST OF TABLES

<u>Table</u>		<u>Page</u>
A.1-1	Sterilization Calibration Results	A.1-6
A.1-2	Pulmonary Side Flowmeter Data for Mock Loop Calibrations Tests	A.1-9
A.1-3	Systemic Side Flowmeter Data for Mock Loop Calibration Tests	A.1-10
A.1-4	Pulmonary Flowmeter Calibration Data for Flow Errors	A.1-11
A.1-5	Systemic Flowmeter Calibration Data of Flow Errors	A.1-12
A.1-6	Functions Best Describing the Flow Data With ΔP and Valve Position	A.1-24
A.3-1	Udimet 700 Creep Rupture Test Summary	A.3-20
A.3-2	Properties of Potential Hot Finger Materials	A.3-26
A.3-3	Mod 1 Magnetic Coupling Characteristics	A.3-94
A.3-4	Analysis of Candidate Wire Gauge Mesh Sizes	A.3-179
A.3-5	Mesh Resistance Test Results	A.3-184
A.3-6	Surface Tension Seal Calculations	A.3-185
A.3-7	Relations Developed for Bellows Design	A.3-192
A.3-8	Dead Volumes for Various Configurations	A.3-192
A.3-9	Bellows Test Results	A.3-195
A.3-10	Energy Dispersive X-Ray (EDX) Analysis of Nickel Foil Surfaces (1)	A.3-210
A.3-11	Plasma Spray and Metal Preparation Equipment and Process Parameters	A.3-218
A.3-12	Random Sprayed Zirconia On 1 Mil Ni Foil	A.3-219
A.3-13	Summary of Wire Mesh Masks - 1 Mil Ni Foil - Flat - 2" x 12" - ZrO_2	A.3-222
A.3-14	Geometry of Chemically Milled Masks - Round Holes in 5 Mil Stainless - Staggered Centers	A.3-224
A.3-15	Chemically-Milled Masks for Refractory Spacer Application (30 Mil Diameter Openings)	A.3-226
A.3-16	Equilibrium Thermal Data for the 40 Nickel Cup Pack and the 5 Nickel Cup Pack	A.3-241
A.3-17	Calibration Data from Guarded Heater Test	A.3-245

LIST OF TABLES (Continued)

<u>Table</u>		<u>Page</u>
A.3-18	Description of Guarded Insulation Test System Heaters	A.3-246
A.3-19	Equilibrium Thermal Data from Guarded Heater Tests	A.3-247
A.3-20	Test System No. 2 Data Summary	A.3-251
A.3-21	Test System No. 1 MK IV Data Summary	A.3-252
A.3-22	Preliminary Emittance Measurements on Electroplated Outer Cup	A.3-259
A.3-23	Scram Activation Procedure	A.3-260
A.3-24	Procedure for Applying ZrO_2 Spacer Material to Nickel Foil Cups	A.3-266
A.2-25	Procedure for Etching Nickel Foil Cups From Aluminum Alloy Mandrels	A.3-267
A.3-26	Component and Subassemblies Leak Testing Procedures	A.3-268
A.3-27	Leak Testing Procedure Using UTI Mass Spectrometer	A.3-269
A.3-28	Mass Spectrometer Data Acquisition Program for UTI 100 C Mass Spectrometer and Hewlett-Packard 2116B Mini-Computer	A.3-271
A.3-29	Static Vacuum Package Processing Histories to (3/31/78)	A.3-276
A.3-30	1/8-Inch Diameter S. S. White Core	A.3-296
A.3-31	Wire Wound Flexible Shafts Bench Test Results	A.3-298
A.3-32	Swell of Ventricle Elastomers in Mechanism Oils	A.3-319
A.3-33	Wear Test Data for Sperm Oil-Additive Combinations	A.3-322
A.3-34	Pump Mechanism Bearings	A.3-324
A.3-35	Bearing Life Tests	A.3-326
A.3-36	Average Power Loss in Watts for Various Pump Drive Mechanism Configurations and Conditions	A.3-340
A.3-37	Candidate Thin Coatings for Corrosion Protection of Magnesium	A.3-342
A.3-38	Silastic Coating Conditions Evaluated for the Protection of Magnesium	A.3-347
A.3-39	Nedox Coatings Examined for Corrosion Protection of Beryllium	A.3-354
A.3-40	Comparison of Fluids for Thermal Pump Heat Transport System	A.3-401

LIST OF TABLES (Continued)

<u>Table</u>		<u>Page</u>
A.3-41	Other Possible Working Fluids Considered for the Artificial Heart Thermal Oscillating Pump (TOP) Heat Transport System but Rejected	A.3-415
A.3-42	Comparison of Fluids for Possible Application in the Artificial Heart Thermal Oscillating Pump (TOP) Heat Transport System	A.3-416
A.3-43	Comparison of Waste Heat Rejection Concepts	A.3-445

A.1.0 SUPPORT EQUIPMENT

The two major pieces of supporting equipment, required to carry out the procedures outlined in Section A.2.0 of this document, consist of a glove box and the converter auxiliary cart. The glove box provides a clean, sterile, nearly oxygen-free environment for final assembly of the thermal converter.

The auxiliary cart provides the support equipment needed to start the converter, maintain normal operation prior to implantation and to monitor the hot finger temperature after implantation. In addition, it provides the excitation voltage for the pressure transducers on the converter. Section 4.3 contains a schematic of the auxiliary cart including a schematic of the system test of the IVBM artificial heart. This illustrates the connections between the auxiliary cart and the artificial heart, and also identifies the instrumentation required for check-out and operation of the converters.

In addition, bench test simulations often required the use of a mock circulation (loop). An extensive calibration of the mock loop used is reported.

A.1.1 EQUIPMENT LIST

A.1.1.1 Blood Pump Assembly and Equipment Recovery Equipment

1. Blood pump mechanism, coupling mechanism, and ventricles.
2. Laminar flow hood or equivalent clean work area.
3. Blood pump mechanism locking fixture.
4. Coolant pump locking fixtures (2).
5. Vacuum system capable of reaching 0.1 mm-Hg.
6. Sterile valves and vacuum accumulator.
7. Flask of deaerated sterile water.
8. Vacuum gage.
9. Supply of sterile sperm oil or other oil as specified.*
10. Punch and anvil for removal and insertion of pusher cup locking pins.
11. Sterile container for completed blood pump.
12. Hemostats (2).

*Other Possible Oils - Jojoba, Vitrea 21, Brayco 811

13. Thermal sealer for coolant tubing.
14. Miscellaneous miniature wrenches and screwdrivers.
15. Non-flammable mixture of freon and ethylene oxide.
16. Permeable plastic bags.
17. Polyethylene bag.
18. Cap for gear box.
19. Bellows retainers (2).
20. Nose cone for flexible shaft and coolant loop.

A.1.1.2 First Stage Implantation Equipment List

1. Assembled blood pump, drive motor and coupling mechanism.
2. Set of quick connects.
3. Noise cone for shaft and coolant loop.
4. Sterile syringe filled with sterile sperm oil or other specified oil.
5. Supply of carbon dioxide.
6. Adapter for flexible shaft with shaft seal and fill port.

A.1.1.3 Thermal Converter Field Assembly Equipment List

1. Thermal converter and isotope heat source.
2. Auxiliary coolant loop components.
3. Auxiliary oil loop components if required by thermal converter configuration.
4. Auxiliary cart with pre-heat chamber.
5. Radiation monitoring equipment.
6. Glove box.
7. Argon supply, sterile, deaerated water and converter lubricating oil.
8. Vacuum system.
9. Insertion arbor for liner assembly.
10. Buffer space fill valve actuator.
11. Vacuum scram tool.
12. Miscellaneous wrenches, allen head wrenches and tools.
13. Multi-channel recorder.
14. Non-flammable mixture of freon and ethylene oxide.
15. Fairing and insulation for cold end heat exchanger.
16. Oscilloscope.
17. Graduate - 300 ml.
18. 250 ml graduate, filled with sterile sperm oil or alternate oils and aluminum foil sealed (2).

A.1.1.4 Thermal Converter Implantation Equipment List

1. Operating fueled thermal converter in pre-heat chamber with starter motor/dummy load.
2. Auxiliary cart.
3. Vacuum scram tool.
4. Coolant tube transition insulation and silastic membrane.
5. Multi-channel recorder.
6. Sterile syringe filled with sterile sperm oil.
7. Auxiliary oil line pinch-off tool.
8. Nose cone for vacuum scram line and converter instrumentation leads.
9. Hemostats for clamping coolant lines (4).
10. Thermal sealing tool for coolant lines.
11. Crimping tool for coolant shunt.
12. Preformed insulation for coolant tubing.
13. Preformed Silastic biological barrier for above.

A.1.1.5 Post-Operative Thermal Converter Monitoring Equipment List

1. Strip chart recorder capable of 200 mm/sec chart.
2. Auxiliary cart.

A.1.1.6 Thermal Converter Recovery Equipment List

1. Auxiliary cart.
2. Vacuum scram tool.
3. Glove box with supply of argon.
4. Cooling water supply.
5. Hemostats (2).
6. Starter/dummy load motor.
7. Buffer volume fill valve.
8. Miscellaneous wrenches for removal of hot finger.
9. Replacement auxiliary oil lines.
10. Stroboscopic tachometer.
11. Water container suitable for cooling vacuum foil insulation.
12. Heat source storage container.
13. Sponge for temporary cooling.

A.1.2 CALIBRATIONS

A.1.2.1 Sterilization

A sterilization calibration test was carried out in the glove box in accordance with the IVBM final assembly and sterilization procedure as outlined in Section A.2.0. The procedure used for this calibration primarily reflected use of the implantable electrically heated thermal converter rather than the isotope heat source converter. The only deviations from the established procedure during the test were that:

- The electrically heated thermal converter and related instrumentation were not in the glove box.
- The covers over the glove box portals were never removed after the test to check for possible oxyfume permeation of the rubber gloves.

The sterilizing gas used in the test was a mixture of ethylene oxide and dichlorodifluoromethane commercially called "oxyfume." The formula for the active ethylene oxide component is $\text{CH}_2\text{CH}_2\text{O}$. Its properties are:

- Colorless gas at ordinary temperatures,
- Mobile, colorless liquid at low temperatures,
- Soluble in the usual organic solvents and miscible with water in all proportions,
- Boiling point of 10.7°C , specific gravity (20/20°C) of 0.8711 and viscosity (0°C) of 0.32 cps., and
- Extremely flammable, vapors harmful and may cause skin burns.

However, to minimize the danger of handling ethylene oxide, a nonflammable hospital grade mixture of 12 percent by weight of ethylene oxide inerted with 88 percent by weight of dichlorodifluoromethane gas (oxyfume) was used.

Prior to the sterilization check-out, the glove box was leak tested numerous times using nitrogen gas. After painstaking efforts to seal every possible leak, which included the installation of a special inflatable rubber seal around the door, the minimum average leak rate that could be obtained using N_2 gas was $4 \rightarrow 2$ inches H_2O in ~45 minutes. This was unacceptable with the 1-inch per hour technical specification requirement and the 2-inch per hour max. allowable leak rate calculated for a 50 ppm oxyfume (ethylene oxide) level in

the Heart Laboratory. Therefore, to avoid possible hazard to personnel, the glove box was moved and relocated directly under the large overhead ventilation hood in the laboratory. The duct work from the hood goes to a large suction fan located on the roof of the building. Any leakage from the box was immediately sucked into the hood and expelled safely to the outside environment.

After the glove box was thoroughly cleaned and leak-tested, five sterile test specimens were placed at various locations in the box. Also, one was placed in the pre-filter for a total of six specimens. A scale was then placed under the oxyfume cylinder. Several fans were located around the glove box and directed to blow air upward around the box into the exhaust hood above. The laboratory door was also blocked open to allow for additional ventilation. The glove box was then purged with argon gas in the normal manner as described in the procedure. After purging the valve, the pre-filter was closed. The fans were turned on and the glove box was purged with 23-lb of oxyfume. This took about 45 minutes. After the oxyfume purging, positive pressure was maintained in the box throughout the 12-hour sterilization test. The test results are outlined in detail in Table A.1-1 and are believed self-explanatory. Note that air samples around the box were periodically taken during the test to check on oxyfume leakage levels. The only appreciable leak detected was around the glove box exhaust fan housing during oxyfume purge at the start of the run and purging out of oxyfume with argon at the end of the run.

After the sterilization cycle, the Steritest were removed from the box and examined for a change in indicator color from yellow to blue. This color change demonstrates that the Steritest unit was exposed to ethylene oxide gas. The ampule in the Steritest unit was then broken according to procedure, and the broth shaken onto the spore strip. The Steritest was then incubated at 37°C (98.6°F) for 12 hours. If the culture medium is clear and the color remains blue-green, the spores are sterile and the location of that Steritest unit in the glove box can be considered sterile. If the culture medium is cloudy and the color is yellow, the spores are not sterile and the location of that unit must be considered not sterile. All five Steritest in the glove box showed good sterilization. However, the one located in the pre-filter failed both the indicator color change test and the incubator test. This indicated

TABLE A.1-1
STERILIZATION CALIBRATION RESULTS

Time (Hour)	Added Oxyfume Equivalent (in H ₂ O)	Approx. Glove Box Pressure Range (in H ₂ O)	Wgt. Oxyfume Cylinder (lbs.)	Note
0847	-	2	166	Glove box initially purged with argon gas. Start of Oxyfume purge. Note, 15 psi on unflatable seal.
0932	-	2 → 4	143	End of Oxyfume purge (23 lbs Oxyfume used). Start of 12 hr. test. Leak detected at glove box exhaust fan.
1250	1.5	2.5 → 4	-	Many air samples taken around door seal and glove box fittings, etc. No leakage detected. Glove box exhaust fan valved off.
1345	3.5	0.5 → 4	-	No leak around box detected. However, pressure drop indicates leak.
1535	3.5	0.5 → 4	-	Continue air sampling. No leak detected.
1720	3.5	0.5 → 4	-	Air samples taken. No leak detected.
1915	3.5	0.5 → 4	-	Air samples taken.
2045	3.0	1 → 4.5	-	Air samples taken.
2132	0	3.7	140	12 hr. test over (3 lbs. Oxyfume used).
2135	0	3.7	-	Start of Argon purge.
2157	0	4	-	End of Argon purge.
2235	0	0	-	Steritest specimen incubation begins. Specimens put in oven at 37°C for 12 hours. Glove box specimens show good sterilization.

that ethylene oxide gas never sufficiently diffused down to the pre-filter during the 12-hour test. This is necessary since the pre-filter is used for inert gas purging of the glove box after the sterilization. The test overall does demonstrate the applicability of the test procedure. The results are positive indicating complete sterilization of culture mediums (bacterial spores) in the box. To eliminate any possibility of recontamination of specimens after the 12-hour sterilization, ethylene oxide was introduced directly into the pre-filter during subsequent sterilizations. This ensures sterilization of the filter, line, and valve prior to inerting.

A.1.2.2 Mock Circulation (Loop) Calibration

The mock loop flowmeter calibration test setup was assembled as shown in Figure A.1-1. The test request indicates that the dimension "O" should be adjusted as required to set the gate valve in the systemic side as well as the pulmonary side of the mock loop in approximately the 10, 25, 50, 75 and 100 percent open flow area positions. This was not adhered to explicitly due to the magnitude of flow that results with the higher open flow area positions. Instead, the dimension of the systemic side was adjusted to be approximately 15, 17, 19 and 21 percent open; and the dimension of the pulmonary side adjusted to the 14, 16, 18 and 20 percent open flow area positions, approximately.

The experimental data obtained in the tests are given in Tables A.1-2 and A.1-3 for both the pulmonary and systemic sides of the mock loop, respectively. Here, the Ramapo flowmeters in the mock circulation loop and the Ramapo flowmeter returned to Ⓢ Advanced Energy Systems Division by the University of Utah were calibrated by comparing flowmeter readings against a volumetric measured amount of water collected in a calibrated container. The error differences between measured flow and flowmeter readings were calculated and are given in Tables A.1-4 and A.1-5.

As shown in Figures A.1-2 through A.1-9, plots of these error differences versus flowmeter flow for both the pulmonary side and systemic side seems to be rather random and independent of valve setting. Correlation of the data was made, however, by using a least square fit of error and flow. A second order fit of the

Figure A.1-1. Mock Loop Flowmeter Calibration Test Setup

TABLE A.1-2

PULMONARY SIDE FLOWMETER DATA FOR MOCK LOOP CALIBRATION TESTS

RUN #	%	TEMP H ₂ O °F	INLET PRES- SURE mm Hg	ΔP mm Hg	OUTLET PRES- SURE mm Hg	Flow Liters Recorder	AESD	Liters Digital	Flow Liters Recorder	UTAH	Liters Digital	MEASURED FLOW		Dimension	Outlet Valve
							Meter Gal Digital			Meter Gal Digital		Liters	Time		
6	14	75	10	9.0	1.0	5.25	1.35	5.1	5.5	1.45	5.48	5.620	1 min.	3.436	Fully
7	14	75	20	17.1	2.9	7.25	1.98	7.49	7.5	2.07	7.83	8.040	1 min.	3.436	Fully
8	14	75	30	26.05	3.95	9.50	2.32	8.78	9.5	2.54	9.61	9.78	1 min.	3.436	Fully
9	14	75	40	35.75	4.25	10.50	2.73	10.33	10.75	2.86	10.83	11.20	1 min.	3.436	Fully
10	14	75	50	45.0	5.00	11.50	3.00	11.36	11.50	3.15	11.92	12.28	1 min.	3.436	Fully
11	16	75	10	7.75	2.25	7.0	1.73	6.54	7.0	1.76	6.66	6.94	1 min.	3.448	Fully
12	16	75	20	16.75	3.25	9.5	2.39	9.04	9.5	2.48	9.39	9.38	1 min.	3.448	Fully
13	16	75	30	25.5	4.5	11.0	2.84	10.75	11.0	2.95	11.17	11.36	1 min.	3.448	Fully
14	16	75	40	34.25	5.75	12.5	3.30	12.5	12.5	3.35	12.68	13.14	1 min.	3.448	Fully
15	16	75	50	43.75	6.25	14.5	3.70	14.0	14.0	3.80	14.69	14.86	1 min.	3.448	Fully
16	18	75	10	7.0	3.0	8.5	2.11	7.99	8.5	2.22	8.40	8.53	1 min.	3.460	Fully
17	18	75	20	15.0	5.0	11.5	2.89	10.94	11.5	3.03	11.47	11.54	1 min.	3.460	Fully
18	18	75	30	23.5	6.5	13.75	3.59	13.59	14.0	3.72	14.08	14.24	1 min.	3.460	Fully
19	18	75	40	32.0	8.0	15.0	4.01	15.18	15.5	4.10	15.52	15.84	1 min.	3.460	Fully
20	18	75	50	40.0	10.0	17.5	4.60	17.41	17.75	4.66	17.64	18.34	1 min.	3.460	Fully
21	20	75	10	6.0	4.0	9.0	2.37	8.97	9.5	2.52	9.54	9.68	1 min.	3.472	Fully
22	20	75	20	14.75	5.25	13.0	3.29	12.45	13.0	3.39	12.83	13.20	1 min.	3.472	Fully
23	20	75	30	22.0	8.0	16.0	4.05	15.33	16.5	4.23	16.01	16.20	1 min.	3.472	Fully
24	20	75	40	29.5	10.5	18.5	4.86	18.40	18.75	4.96	18.77	18.80	1 min.	3.472	Fully
25	20	75	50	38.0	12.0	20.0	5.19	19.64	20.00	5.32	20.14	20.88	1 min.	3.472	Fully

TABLE A.1-3

SYSTEMIC SIDE FLOWMETER DATA FOR MOCK LOOP CALIBRATION TESTS

RUN	DIMEN- SION	TEMP WATER °F	INLET PRES- SURE mm Hg	OUTLET PRES- SURE mm Hg	ΔP mm Hg	AESD		Liters Digital	UTAH		Liters Digital	MEASURED FLOW	
						Flow Liters Recorder	Meter Gal Digital		Flow Liters Recorder	Meter Gal Digital		Liters	Time
1. 15%	3.342	75	80	2.5	77.5	5.00	1.35	5.11	5.00	1.36	5.14	5.18	1 min.
2. "	3.342	75	100	2.7	97.3	6.00	1.59	6.02	6.00	1.57	5.94	6.00	1 min.
3. "	3.342	75	120	3.2	116.8	6.5	1.75	6.62	6.50	1.77	6.70	6.82	1 min.
4. "	3.342	75	140	3.7	136.3	7.25	1.95	7.38	7.50	1.97	7.46	7.50	1 min.
5. 15%	3.342	75	160	4.0	156.0	8.00	2.05	7.76	8.00	2.13	8.06	8.00	1 min.
1. 17%+ (16% on Brush)	3.357	75	80	3.0	77.0	6.00	1.58	5.98	6.00	1.60	6.05	6.27	1 min.
2. "	3.357	75	100	3.5	96.5	6.625	1.78	6.74	7.00	1.82	6.89	7.05	1 min.
3. "	3.357	75	120	4.0	116.0	7.625	2.04	7.72	8.00	2.08	7.87	8.10	1 min.
4. "	3.357	75	140	4.3	135.7	8.5	2.23	8.44	8.75	2.30	8.71	8.88	1 min.
5. "	3.357	75	160	5.0	155.0	9.25	2.45	9.27	9.5	2.50	9.46	9.7	1 min.
1. 19% (18% on Brush)	3.368	75	80	3.7	76.3	7.25	1.89	7.15	7.45	1.92	7.27	7.45	1 min.
2. "	3.368	75	100	4.5	95.5	8.25	2.16	8.18	8.5	2.20	8.33	8.52	1 min.
3. "	3.368	75	120	4.9	115.1	9.25	2.40	9.08	9.5	2.45	9.27	9.46	1 min.
4. "	3.368	75	140	6.1	133.9	10.0	2.62	9.91	10.0	2.69	10.18	10.32	1 min.
5. "	3.368	75	160	7.0	153.0	10.75	2.83	10.71	11.00	2.90	10.93	11.165	1 min.
1. 21%	3.380	--	80	7.1	72.9	10.25	2.68	10.14	10.50	2.72	10.30	10.9	1 min.
2. "	3.380	--	100	7.5	92.5	11.5	3.04	11.51	11.75	3.07	11.62	11.9	1 min.
3. "	3.380	--	120	8.8	111.2	13.00	3.40	12.87	13.25	3.45	13.06	13.21	1 min.
4. "	3.380	--	140	10.5	129.5	14.00	3.72	14.08	14.25	3.75	14.19	14.54	1 min.
5. "	3.380	--	160	12.0	148.0	15.00	3.98	15.06	15.5	4.05	15.32	15.42	1 min.

TABLE A.1-4

PULMONARY FLOWMETER CALIBRATION DATA OF FLOW ERRORS

RUN #	Valve Open %	ΔP (mm Hg)	WESTINGHOUSE FLOWMETER				UTAH FLOWMETER				Measured Flow (l/min)
			Flow Brush Record (l/min)	Error (measured - Brush) (l/min)	Flow Digital (l/min)	Error (measured - Digital) (l/min)	Flow Brush Record (l/min)	Error (measured - Brush) (l/min)	Flow Digital Record (l/min)	Error (measured - Digital) (l/min)	
6	14	9.0	5.25	0.37	5.1	0.52	5.5	0.12	5.48	0.14	5.62
7	14	17.1	7.25	0.79	7.49	0.55	7.5	0.54	7.83	0.21	8.04
8	14	26.05	9.50	0.28	8.78	1.00	9.5	0.28	9.61	0.17	9.78
9	14	35.75	10.50	0.70	10.33	0.87	10.75	0.45	10.83	0.37	11.20
10	14	45.0	11.50	0.78	11.36	0.92	11.50	0.78	11.92	0.36	12.28
11	16	7.75	7.0	-0.06	6.54	0.40	7.0	-0.06	6.66	0.28	6.94
12	16	16.75	9.5	-0.12	9.04	0.34	9.5	-0.12	9.39	-0.01	9.38
13	16	25.5	11.0	0.36	10.75	0.61	11.0	0.36	11.17	0.19	11.36
14	16	34.25	12.5	0.64	12.5	0.64	12.5	0.64	12.68	0.46	13.14
15	16	43.75	14.5	0.36	14.0	0.86	14.0	0.86	14.69	0.17	14.86
16	18	7.0	8.5	0.03	7.99	0.54	8.5	0.03	8.4	0.13	8.53
17	18	15.0	11.5	0.04	10.94	0.60	11.5	0.04	11.47	0.07	11.54
18	18	23.5	13.75	0.49	13.59	0.65	14.0	0.24	14.08	0.16	14.24
19	18	32.0	15.0	0.84	15.18	0.66	15.5	0.34	15.52	0.32	15.84
20	18	40.0	17.5	0.84	17.41	0.93	17.75	0.59	17.64	0.70	18.34
21	20	6.0	9.0	0.68	8.97	0.71	9.5	0.18	9.54	0.14	9.68
22	20	14.75	13.0	0.20	12.45	0.75	13.0	0.20	12.83	0.37	13.20
23	20	22.0	16.0	0.20	15.33	0.87	16.5	-0.30	16.01	0.19	16.20
24	20	29.5	18.5	0.30	18.40	0.40	18.75	0.05	18.77	0.03	18.80
25	20	38.0	20.0	0.88	19.64	1.24	20.00	0.88	20.14	0.74	20.88

TABLE A.1-5

SYSTEMIC FLOWMETER CALIBRATION DATA OF FLOW ERRORS

RUN #	Valve Open %	ΔP (mm Hg)	WESTINGHOUSE FLOWMETER				UTAH FLOWMETER				Measured Flow (L/min)
			Flow Brush Record (L/min)	Error (measured -Brush) (L/min)	Flow Digital (L/min)	Error (measured -Digital) (L/min)	Flow Brush Record (L/min)	Error (measured -Brush) (L/min)	Flow Digital Record (L/min)	Error (measured -Digital) (L/min)	
1	15	77.5	5.00	0.18	5.11	0.07	5.0	0.18	5.14	0.04	5.18
2	15	97.3	6.00	0	6.02	-0.02	6.0	0	5.94	0.06	6.00
3	15	116.8	6.50	0.32	6.62	0.2	6.5	0.32	6.7	0.12	6.82
4	15	136.3	7.25	0.25	7.38	0.12	7.5	0	7.46	0.04	7.50
5	15	156.0	8.00	0	7.76	0.24	8.0	0	8.06	-0.06	8.00
1	17/16 Brush	77.0	6.00	0.27	5.98	0.29	6.0	0.27	6.05	0.22	6.27
2	17/16	96.5	6.625	0.425	6.74	0.31	7.0	0.05	6.89	0.16	7.05
3	17/16	116.0	7.625	0.475	7.72	0.38	8.0	0.10	7.87	0.23	8.10
4	17/16	135.7	8.5	0.38	8.44	0.44	8.75	0.13	8.71	0.17	8.88
5	17/16	155.0	9.25	0.45	9.27	0.43	9.5	0.2	9.46	0.24	9.7
1	19/18 Brush	76.3	7.25	0.2	7.15	0.3	7.45	0	7.27	0.18	7.45
2	19/18	95.5	8.25	0.27	8.18	0.34	8.5	0.02	8.33	0.19	8.52
3	19/18	115.1	9.25	0.21	9.08	0.38	9.5	-0.04	9.27	0.19	9.46
4	19/18	133.9	10.0	0.32	9.91	0.41	10.0	0.32	10.18	0.14	10.32
5	19/18	153.0	10.75	0.415	10.71	0.455	11.0	0.165	10.98	0.185	11.165
1	21	72.9	10.25	0.65	10.14	0.76	10.50	0.4	10.30	0.6	10.9
2	21	92.5	11.5	0.4	11.51	0.39	11.75	0.15	11.62	0.28	11.9
3	21	111.2	13.0	0.21	12.87	0.34	13.25	-0.04	13.06	0.15	13.21
4	21	129.5	14.0	0.54	14.08	0.46	14.25	0.29	14.19	0.35	14.54
5	21	148.0	15.0	0.42	15.06	0.36	15.5	-0.08	15.32	0.1	15.42

Figure A.1-2. Pulmonary Error Versus Flow (Ⓜ Brush Recorder Data) at Fixed Valve Position

Figure A.1-3. Pulmonary Error Versus Flow (Ⓢ Digital Data) at Fixed Valve Positions

Figure A.1-4. Pulmonary Error Versus Flow (Utah Brush Recorder Data) at Fixed Valve Positions

Figure A.1-5. Pulmonary Error Versus Flow (Utah Digital Data) at Fixed Valve Positions

Figure A.1-6. Systemic Error Versus Flow (⊗ Brush Recorder Data) at Fixed Valve Positions

Figure A.1-7. Systemic Error Versus Flow (Ⓜ Digital Data) at Fixed Valve Positions

Figure A.1-8. Systemic Error Versus Flow (Utah Brush Recorder Data) at Fixed Valve Positions

Figure A.1-9. Systemic Error Versus Flow (Utah Digital Data) at Fixed Valve Positions

data seems to give the best results. These curves are also presented in Figures A.1-2 through A.1-9 along with the appropriate second order functions describing them. In all future testing using the above Ramapo flowmeters, these calibration curves or functions should be used to correct flow readings within the appropriate flow range of the data.

In an attempt to better describe the gate valve characteristics for both the pulmonary and systemic sides of the mock loop, respectively, correlation of the flow, pressure drop and valve position were made. Here, the OMNIFIT Least Squares Fit Code for non-linear functions was used. As shown in Figure A.1-10, the pulmonary gate valve least squares fit to measured data was excellent. Of the following functions tried:

$$\omega = C_1 \theta^{C_2} \Delta P^{0.5} \quad (1)$$

$$\omega = C_1 \theta^{C_2} \Delta P^{C_3} \quad (2)$$

$$\omega = C_1 \theta^{C_2} \Delta P^{C_3} + C_4 \Delta P^{C_3} \quad (3)$$

$$\omega = C_1 e^{C_2 \theta} \Delta P^{C_3} \quad (4)$$

$$\omega = C_1 e^{C_2 \theta} \Delta P^{C_3} + C_4 \Delta P^{C_3} \quad (5)$$

where:

ω = flow, L/min

θ = valve position open, %

ΔP = pressure drop, mm-Hg

C_1, C_2, C_3, C_4 = Constants

Equation (2) type seem to best describe the pulmonary side valve. For the systemic side, Equation (4) was a better fit. Figure A.1-11 gives the least squares fit to the systemic measured data. Here the systemic valve characteristic function percent error was greater than the pulmonary valve and also was more biased (less random). This was probably due to the different physical shape of the valves. The pulmonary gate valve is more circular and more consistent in flow area changes with valve position. The systemic valve is sort of wedged shape and would tend to be more abrupt.

Figure A.1-10. Pulmonary Side ΔP Versus Measured Flow at Fixed Valve Positions for Pulmonary Valve

Figure A.1-11. Systemic Side ΔP Versus Measured Flow at Fixed Valve Positions for Systemic Valve

Table A.1-6 lists the flow functions and the appropriate constants for the various flowmeters/recorders data in Tables A.1-1 and A.1-2. Again Equations (1) through (5), above, were tried using OMNIFIT for each set of data. Note that the functions correlated in Table A.1-6 are only as good as the experimental data. Since valve position readings were difficult to measure and reproduce (a micro-caliper was used to measure valve position open in the tests), these functions should only be used as representations of the approximate gate valve characteristics. However, they do describe the performance of the gate valves and can be used to establish an approximate valve setting for a desired pressure drop and flow or vice-versa.

TABLE A.1-6
FUNCTIONS BEST DESCRIBING THE FLOW DATA
WITH ΔP AND VALVE POSITION

DATA	FUNCTION	
	PULMONARY VALVE	SYSTEMIC VALVE
Ⓢ Brush	$\omega = 0.01899\theta^{1.801}\Delta P^{0.4328}$	$\omega = 0.06313 e^{0.1231\theta}\Delta P^{0.5729}$
Ⓢ	$\omega = 0.01649\theta^{1.821}\Delta P^{0.4521}$	$\omega = 0.06275 e^{0.1224\theta}\Delta P^{0.5765}$
UTAH Brush	$\omega = 0.01843\theta^{1.828}\Delta P^{0.4211}$	$\omega = 0.06139 e^{0.1234\theta}\Delta P^{0.5817}$
UTAH Brush	$\omega = 0.02180\theta^{1.753}\Delta P^{0.4371}$	$\omega = 0.06073 e^{0.1215\theta}\Delta P^{0.5905}$
MEASURED	$\omega = 0.02245\theta^{1.741}\Delta P^{0.4449}$	$\omega = 0.07106 e^{0.1226\theta}\Delta P^{0.5570}$

A.2.0 EXPERIMENTAL PROCEDURES

A.2.1 THERMAL CONVERTER STERILIZATION

- Assemble electrically heated thermal converter.
- Check thermal insulation around package and repair or replace if required.
- Check condition of coolant line extensions. Repair or replace if necessary.
- Install coolant line extensions and leak check between line extensions and cold end heat exchanger.
- Repair thermal converter coating with RTV if required. Allow sufficient time to cure. Clean external surface of converter with surgical soap and water.
- Thoroughly clean glove box inside and out with surgical soap and water. Also, clean thermal converter, Argon fill line, and adapter and place inside of glove box. Make sure fill line and adapter are thoroughly dry.
- Seal coolant lines to converter with Silastic or RTV and connect thermal converter coolant lines to empty auxiliary coolant system through glove box fittings.
- Place thermal converter, starter motor and hysteresis brake with flex shaft in glove box.
- Prepare thermal converter according to Section A.2.4.
- Place polyethylene bag or cloth wrap, instrumentation and coolant line, protective coverings, Argon fill valve cap, flywheel housing cap and everything to be sterilized in glove box.
- Provide container with 0.025 pound (0.69 in³) of water to provide required relative humidity during sterilization.
- Flush coolant system with distilled water using three or more complete changes of water. Be sure both pump circuits are flushed.
- Fill coolant system completely with "Betadine" antiseptic solution and circulate "Betadine" for three hours. Make sure both pump circuits are used. Drain coolant system and refill and flush to operating levels with sterile distilled water.
- Close and seal glove box and all inlet-outlet valves.
- Open valve to glove box pressure gage.
- Pressurize glove box with air or nitrogen to 4-in. water and let stand for one hour. Check glove box pressure. Gage should read

over 3-in. water. If not, look for leaks and correct, where possible. When satisfied that glove box is sealed properly, depressurize and proceed to next step.

- Place covers over the glove portals. Connect Argon to "T" and valve mounted at glove box. Close valve. Make sure Oxyfume manifold valves are closed. Close Oxyfume vent valve external to work area and open feed valve. Close all valves on Oxyfume line and leak check. Also, open Oxyfume feed valve external to work area. Close Oxyfume valve at inlet to glove box. Install scale under Oxyfume cylinder to measure quantity of gas used during sterilization. (Note: Oxyfume is the commercial name for ethylene oxide-freon mixture.)
- Pressurize Argon regulator to 120 psig and open valve to Oxyfume line. Check all joints, Oxyfume manifold and cylinder for leaks. Correct any leaks that show up. (Note: All Oxyfume cylinders should be properly grounded. Absolutely NO SMOKING, open flames, or sparks near Oxyfume cylinders and/or related equipment.)
- Close valve from Argon supply and disconnect Argon supply from Oxyfume line.
- Set up fans to evenly distribute any Oxyfume leakage into Heart Lab. (Note: May want to open all hallway doors to allow for additional ventilation resulting from natural convection.)
- Open vent valve on glove box and start exhaust fan. Check glove covers on glove box making sure they are secured.
- Purge glove box with at least 90-ft³ of Argon (4-in. over 20 min). Close valve to Argon prefilter and desiccator.
- Break vials and place steritest alphone vials in filter chamber before main filter.
- Purge glove box with at least 23 pounds of Oxyfume, making sure Oxyfume penetrates the Argon fill lines up to and through millipore filter. Positive pressure not exceeding 4-in. of water should be maintained in the glove box during this step. After the purge, make sure the glove box is holding its pressure. If not, proceed to the next step, otherwise check for leaks and correct. (Note: When working in glove box with Oxyfume, both cloth and surgical gloves must be worn. Exhaust fan is on during Oxyfume purge only.)
- Shut off Oxyfume at glove box and close vent and Argon lines, sealing the box with an internal pressure of 4-in. of water.
- Allow glove box to sit for 12 hours while sterilization is underway. During this period, a small flow of oxyfume may be required to maintain a slight (0.2 to 4-in. of water pressure) positive internal pressure. (Note: Keep a log of Oxyfume pressure loss and replacement. A pressure makeup of 20-in. H₂O should not be exceeded.)
- Take a sample of air near the glove box and pass it through the detector tube attached to the Matheson toxic gas detector. (Toxic detector Kit No. 801K, Oxyfume detector tubes No. 8014-122.)

Repeat approximately every hour. (Note: If large quantities of toxic Oxyfume gas are detected, shut off Oxyfume cylinders, open vent valve and purge glove box with at least 90 cubic-feet of Argon [4-in. over 20 min]. Otherwise, sample testing shall not interfere with laboratory test procedures associated with sterilization of the starter motor/thermal converter assembly.)

- After 12 hours sterilization, open vent valve and purge glove box with at least 90 cubic-feet of Argon (4-in. over 20 min). Maintain positive internal pressure (0.2 to 4-in. of water) in the glove box at all times during this operation, and until converter is polyethylene bagged or cloth wrapped. Open Oxyfume line vent and purge Oxyfume line. Purge fill line with argon.
- Start slow purge through Argon inlet line and millipore filter using at least 4-ft³/hr of clean air supplied by small air pump or Argon with limited output capacity. Maintain positive pressure (0.2 to 4-in. of water) in glove box.
- Connect starter motor and hysteresis brake to thermal converter. Start T/CNV according to Section A.2.4.4. Run converter for minimum 4-hour performance test. (Note: This can be done during continuous slow purge with air until Oxyfume has diffused out of converter surfaces. This diffusion step can be expected to take from 24 to 72 hours. The thermal converter may be stopped at the end of the performance run or left operating through the entire diffusion period at the Principal Investigator's option.) (Note: Prior to using gloves, check for presence of Oxyfume.)
- Remove buffer fill valve and install buffer cover cap.
- Seal bag around instrumentation lead feed-through. If applicable, move glove box to operating room.
- Open glove box, remove converter in polyethylene bag or cloth wrap and place on sterile cloth on auxiliary cart if applicable. DO NOT open or damage bag or cloth wrap at this time.
- Move bagged operating converter and auxiliary cart to operating room. (Note: If move made with non-operating thermal converter, it can be started at this time to check out instrumentation on cart prior to delivery to operating room.)
- Open bag or cloth wrap and place contents of bag into sterile operating field. (Note: Exterior portions of bag or wrap to be handled by non-sterile personnel. Be careful that exterior portions of bag do not come into contact with the contents of the bag or the scrubbed operating room personnel who will handle the contents of the bag.)
- Disconnect instrumentation leads, feed through incision.
- Re-connect instrumentation.
- Disconnect coolant line, and feed through incision.
- Re-connect coolant lines.
- Converter is ready for implantation.

A.2.2. BLOOD PUMP MECHANISM DISASSEMBLY AND ASSEMBLY

A.2.2.1 Mechanism Disassembly

- Remove pusher cups.
- Remove bellows.
- Remove screws from hardened tracks.
- Loosen yoke housing screws.
- Remove gear box and shims.
- Remove rear main bearing and shims.
- Remove yoke housing screws.
- Disengage hardened tracks and remove yoke housings.
- Remove crank, yoke and hardened tracks.
- Leak check housing at 3 psig using end caps to check inserts for leaks. If inserts leak or displace, the following steps should be performed.
- Strip old Loctite out with Oakite #157 and clean housing and inserts in ultrasonic with trichloroethane.
- Put primer "T" on one window area and one insert and let air dry.
- Put Loctite #40 on housing and insert and clamp in place. (Note: Do not wipe off excess.)
- Place in oven for one hour at 250°F with insert at bottom of "V" block.
- Remove from oven after one hour and let cool with clamp in place. (Note: Do second insert the same way.)
- Check inserts with finger pressure and clean excess cement off of outside of insert. (Note: Do not use any solvent.)
- Put Loctite #290 on outside of both inserts and place in oven with inserts at the sides for one hour at 250°F.
- Clean all excess cement off and leak check at 3 psig with end caps on.

A.2.2.2 Mechanism Assembly

- Clean all parts in ultrasonic with trichloroethane, except blood pump housing. Hand clean housing with ethyl alcohol. Inspect parts for wear and replace as required.
- For dry blood pump housing, perform the following:
 - Grease bearings.
 - Lube linear bearings in Teresstic #52 oil for one hour, under vacuum.

- Rub Neolube on yoke, yoke roller and hardened tracks.
- Coat gears in gear box with Molydag #210. Fill large gear teeth with Andok plus 5% MoS₂.
- Check bearings visually after cleaning, use micro-duster and let air dry before greasing.
- For oil filled blood pump housing, perform the following:
 - Coat bearings with jojoba, sperm or Brayco 811 oil after cleaning.
 - Lube linear bearings in jojoba, sperm or Brayco 811 oil for one hour under vacuum.
 - Check bearings visually after cleaning, use micro-duster and let air dry before coating with oil.
- Place yoke and hardened tracks on crankshaft keeping orientation marks facing in right direction.
- Place crankshaft into housing from either open side, push through front bearing opening and back into rear bearing opening.
- Center yoke with one hand and slide one-half of yoke housing on yoke placing groove roller in groove first.
- Hold and center yoke housing, check orientation marks, make sure yoke is fully recessed in yoke housing, and slide second half of yoke housing in place.
- Snap hardened track into position and hold with two screws. (Note: Do not tighten.)
- Snap other hardened track into position and hold with two screws. (Note: Do not tighten.)
- Place front "O" ring in position.
- Place front shims in position.
- Install gear box. (Note: Do not tighten.)
- Place rear "O" ring in position.
- Place rear shims in position.
- Install rear main bearing and housing, tighten screws.
- Tighten gear box screws.
- Position yoke housing screws and nuts, and tighten.
- Install remainder of hardened track screws and tighten all of them.

A.2.2.3 Field Assembly of Blood Pump and Coupling Mechanism

Blood pump mechanism has been assembled and checked for binding. The coolant lines up to the point where the shunt is attached are installed. Insulation and the Silastic protective covering have been applied to the outer surfaces

of the pump mechanism. Proceed with the following:

- Peel back Silastic protective coating and insulation from blood pump shaft coupling. Remove shaft and shaft core.
- Examine mechanism visually for wear and misalignment of moving parts.
- Manually operate mechanism and check for binding.
- Remove insulation from discharge check valve.
- Loosen flanged end of crossover tube from discharge check valve.
- Test discharge and suction check valves for proper operation.
- Re-install crossover pipe.
- Re-install insulation on discharge check valve.
- Unscrew pusher cups, noting alignment of pusher cup with mechanism. Be careful not to interchange cups.
- Using appropriate fixture, lock mechanism with both bellows in the neutral position. Attach both bellows retainers, if applicable.
- Install fixture between pumping bellows and moving platform to prevent motion of pumping bellows, if applicable.
- Attached unused shunt and coolant tube extensions. Place permeable plastic bag on inlet tube extension.
- Connect outlet side of cooling system to an accumulator followed by a valve and connection for a vacuum system.
- Sterilize internals of coolant system with ethylene oxide/freon mixture.
- Connect inlet side of cooling system to flask of sterile, de-aerated water through a sterile valve.
- Connect accumulator to vacuum pump, open valve to vacuum pump and pump coolant system and accumulator to a pressure of 0.1 mm-Hg or less.
- Close valve to vacuum pump.
- Crack valve to water flask and allow coolant system to fill slowly.
- When water stops flowing, pinch off end of coolant lines with hemostats, allowing at least 2-in. excess length.
- Cut coolant lines between hemostats and accumulator and supply flask.
- Thermally seal cut ends of coolant lines.
- Remove fixturing from coolant pump bellows.
- Remove fixture from gear box and cap gear box.
- Install ventricles (see Section A.2.3).
- Install shaft housing and replace insulation and protective Silastic around shaft ferrule.

- Sterilize pump assembly and flexible shaft core.
- Fill pump and shaft housing with sterilized sperm or other specified oil.
- Install flexible shaft core.
- Install nose cone on end of shaft.
- Place blood pump assembly in sealed sterile polyethylene bag.

A.2.3 PUMP VENTRICLE INSTALLATION AND SEATING

A.2.3.1 Ventricle Installation

As received, the blood pump has the coolant lines and insulation in place. The flexible shaft is not installed and the gear box is capped. All operations must be carried out in a clean laminar flow hood or equivalent dust free environment. Persons handling blood pump should be wearing clean, lint-free gloves. Only lint-free cleaning materials should be used, and all tools should be cleaned prior to use. Proceed with the following:

- Using appropriate punch and anvil, drive out three locking pins to separate right pusher cup skirt from its dome.
- Leak check ventricles at 1 psig.
- Install right ventricle to pusher cup dome by inserting attachment ring on ventricle into groove on dome. Be sure ventricle is properly positioned to produce the proper final positioning of the suction and discharge ports.
- Clamp right ventricle to pusher cup dome with right skirt. Make sure alignment pin on skirt engages notch in dome.
- Install retaining pins and drive into place with punch.
- Repeat all of the above steps for the left ventricle.
- Leak check ventricles again at 1 psig.
- Screw assembled pusher cups with ventricles into their appropriate positions on the blood pump mechanism.
- Fit ventricles onto blood pump housing, making sure the rolling diaphragms are not twisted or distorted.
- Secure ventricles with wire or appropriate clamps. Feed a wire through slot in ventricles being careful not to puncture ventricle internally. Make two wraps with single wire.
- Apply Silastic to seal ventricles to insulation on blood pump body and to seal insulation over coolant system discharge check valve.
- Allow Silastic to cure.
- Leak check ventricle to housing joint with 1 psig through gear box on blood pump.

A.2.3.2 Ventricle Seating

A.2.3.2.1 Dry Blood Pump with Pinned Ventricles

- Blood pump connected to mock circulation and bearing block.
- Vacuum line with gauge hooked up to bearing block.
- Fill ventricles with water (mock open to atmosphere).
- Start blood pump running at 300 rpm.
- Put 4-in. Hg vacuum on blood pump.
- Increase speed to 900 rpm.

A.2.3.2.2 Oil-Filled Blood Pump with Pinned Ventricles

- Blood pump connected to mock circulation.
- Connect tubing to drilled out screw in bearing block; put hemostat on line (line for oil drain).
- Fill ventricles with water.
- Connect flex shaft to bearing block (rotate so large pusher cup in).
- Place end of oil drain tubing in beaker and remove hemostat.
- Pressurize ventricles to 100 mm-Hg.
- Close oil drain line with hemostat or replace drilled out screw with seal screw when flow stops.
- Remove ventricle pressure.
- Open mock circulation to atmosphere.

A.2.3.2.3 Dry Blood Pump with Free-Type Ventricles

- Blood pump connected to mock circulation and bearing block.
- Vacuum line with gauge hooked up to bearing block.
- Fill ventricles with water.
- Start blood pump running at 300 rpm.
- Put 4-in. Hg vacuum on blood pump.
- Pressurize ventricles to 100 mm-Hg.
- Increase speed to 900 rpm.
- Remove ventricle pressure.
- Open mock circulation to atmosphere.

A.2.3.2.4 Oil-Filled Blood Pump with Free-Type Ventricles

- Blood pump connected to mock circulation.
- Connect tubing to drilled out screw in bearing block. Put hemostat on line (line for oil drain).
- Fill ventricles with water.
- Connect flex shaft to bearing block.
- Place oil drain line in beaker and remove hemostat.
- Start blood pump running at 300 rpm.
- Pressurize ventricles to 100 mm-Hg.
- Increase speed to 700 rpm, 900 rpm in 20 min.
- Close oil drain line with hemostat or replace drilled out screw with seal screw when flow stops.
- Remove ventricle pressure.
- Open mock circulation to atmosphere.

A.2.3.2.5 Oil-Filled ΔV Blood Pump with Pinned Ventricles

- Fill blood pump through compliance chamber - leak check prior to filling at 1/2 psig.
- Complete fill under vacuum
 - Install and fill flex shaft
 - Put nose cone on flex shaft
 - Approximately 360 ml oil in blood pump at this time.
- Attach blood pump to mock circulation (MCL) and install bleed (drilled out) screw in bearing block with hemostat on line.
- Fill ventricles with water and connect flex shaft to bearing block. Rotate so large pusher cup in all the way. Remove hemostat from bleed line.
- Pressurize ventricles to 100 mm-Hg. When oil has stopped draining, remove bleed screw and replace with seal screw.
- Remove ventricle pressure and open MCL to atmosphere.
- If diaphragm is collapsed with large pusher cup in, add small amount of oil (5 ml) to blood pump by syringe at bearing block. Exert gas pressure of 50 mm-Hg in ventricles before doing this.
- Repeat above until diaphragm is working by rotating flex shaft by hand.
- Run at 900 rpm with 120 mm-Hg pressure on left ventricle and check operation of diaphragm.
- Blood pump with diaphragm working properly will have approximately 313 ml oil; this will depend on ventricles used.

A.2.3.2.6 Oil Filled ΔV Blood Pump with Free-Type Ventricles

- Proceed as per A.2.3.2.5 through removing hemostat on drain line.
- Start blood pump running at 300 rpm.
- Pressurize ventricles to 100 mm-Hg and increase speed to 700 rpm, to 900 rpm in 20 min.
- Repeat as per A.2.3.2.5 when oil has stopped draining.

Note: Oil-filled blood pump with compliance chamber takes 315 ml.
Oil-filled blood pump without compliance chamber takes 250 ml.

A.2.4 THERMAL CONVERTER PROCEDURES

A.2.4.1 Electrical Thermal Converter Preparation

- Remove eight (8) screws holding hot finger assembly to the thermal converter (T/CNV).
- Remove hot finger assembly from T/CNV.
- Using the insertion arbor, remove the inner liner assembly from the hot finger assembly.
- Store the inner liner assembly with the insertion arbor in the glove box.
- Check T/CNV to see if it is primed with oil. (If not primed, fill T/CNV with 120 ml oil.)
- Connect right angle gear box to T/CNV.
- Install buffer space fill valve.
- Check displacer of T/CNV for tightness (rotate clockwise). (Note: Do not move displacer up or down.)
- Clean displacer with lint-free cloth and grain alcohol.
- Connect line cord of auxiliary cart to 115 VAC outlet.
- Switch "Inverter No. 1" supply to "Both" position.
- Switch "Inverter No. 2" supply to "2" position.
- Turn "Key Switch" to 12 VDC.
- Measure excitation voltage on B&F conditioner units. (Note: Set for 7.5 VDC.)
- Connect pressure transducers to B&F conditioners.
- Check transducers for balance and calibration.
- Remove transducer cables from auxiliary cart connectors.
- Install pressure transducers in T/CNV. (Note: Record serial numbers and locations.)

A.2.4.2 Electrical Thermal Converter Pretest

- Connect transducer cables from auxiliary cart to glove box. (Note: Check cables for labels.)
- Connect extension cables inside glove box. (Note: Check labels.)
- Measure excitation voltage on B&F conditioner units. (Note: Set for 7.5 VDC.)
- Place T/CNV inside glove box.
- Connect transducer extension cables to transducers.
- Balance transducers.
- Connect water lines from auxiliary cart to glove box. (Note: Check labels.)
- Connect water extension lines inside glove box and join together.
- Start auxiliary coolant loop and set temperature control to preheat water to 120°F.
- Connect heater wires from auxiliary cart to glove box. (Note: Check labels.)
- Connect heater wire extension in glove box. (Note: Check labels.)
- Connect T/C cables from auxiliary cart to glove box.
- Connect T/C extension cables inside glove box and to T/CNV.
- Measure T/C at the auxiliary cart end with the L&N potentiometer. (Note: Reconnect T/C cables.)
- Calibrate all transducers according to data charts.
- Place all tools in glove box. Complete tool checklist.
- Check argon supply. (Note: All bottles should be full.)

A.2.4.3 Electrical Thermal Converter Assembly

- Place T/CNV and hot finger assembly inside glove box.
- Complete tool check list.
- Close glove box. (Note: Check that none of the cables get trapped in door.) Insert gloves into glove box and install port covers.
- Open outlet valve of glove box.
- Set argon supply regulator for 200 psi with bottles #1 and #2 on line.
- Open argon system valve.
- Open argon purge valve.
- Adjust throttling valve at manifold for flow rate of 250 SCFH.

- Purge system for 15 minutes. (Note: At the 10 minute point, remove the glove cover plates.)
- Open valve to glove box pressure control gage. Set control point for 0.5 in water pressure.
- Close glove box argon purge valve and then close outlet valve.
- Check that glove box pressure control system is operating properly at 0.5-in. H₂O.
- Assemblers take their respective positions at glove box.
- Connect argon buffer space line to T/CNV.
- Open buffer space vent cap. Remove 4-40 screw from speed control.
- Open buffer space seal screw and purge buffer space for 5 minutes.
- Connect argon line to insertion arbor and purge inner line for 2 minutes.
- Disconnect argon line from insertion arbor and purge hot finger for 1 minute.
- Shut off buffer space argon valve (glove box).
- Replace buffer space vent cap. Replace 4-40 screw in speed control.
- Remove argon buffer line.
- Insert "Inner Liner Assembly" in hot finger assembly. (Note: Axis near horizontal.)
- Remove insertion arbor and place in holder. (Note: Place screws in plastic tray.)
- Check that "O" rings are properly installed in T/CNV.
- Install the hot finger assembly on the T/CNV. (Note: Check that "O" rings are properly seated.) Seal-off plug should be at fly-wheel end of T/CNV.
- Install eight (8) screws to secure assembly. Tighten uniformly.
- Connect argon buffer line.
- Pressurize buffer to 10 psig.
- Depressurize glove box and open door.

A.2.4.4 Electrical Thermal Converter Startup

- Mount T/CNV on motor starter plate.
- Connect thermo probe to controller and apply heat to crankcase.
- Control heat to crankcase at 101°F ±0.5°.
- Turn off auxiliary coolant loop pump.
- Connect coolant lines to T/CNV.

- Start auxiliary coolant loop pump.
- Route pressure transducer leads carefully and connect to transducers.
- Connect heater lines to T/CNV.
- Start heating T/CNV at 33 watts input power.
- Set temperature alarm for 1300°F.
- Align T/CNV and starter motor. Engage motor coupling with flywheel shaft.
- Mount "Motor Starter Assembly" at test position. (Note: Secure with clamps and locating pins.)
- Purge argon lines and T/Conv. 25 times from 50 to 15 psi.
- When alarm goes off at 1300°F, cycle argon pressure three times from 25 to 170 psig, leaving buffer pressure at 170 psig.
- Reset temperature alarm to 1400°F.
- When temperature reaches 1400°F, start T/CNV.
- On motor control set:
 - Switch in "Speed" position.
 - Dial to zero.
 - Rotary switch in "CW" position.
- Turn "Controller" power on. Rotate dial slowly until T/CNV starts. Disengage motor from T/CNV and adjust load. The rpm at flywheel should be 1800 rpm (approximately).
- Record pertinent data.
- If thermal converter power seems lower than expected, reduce temperature to 1300°F.

A.2.4.5 Electrical Thermal Converter Shutdown

- Stall T/CNV by using speed control or other means.
- Turn power "Off" crankcase heater.
- Reduce power to T/CNV heater. (Note: Temperature at top of thin wall should never exceed 1500°F and should not fall below 1300°F with pressure in excess of 80 psig in the buffer volume space.)
- Remove seal cap from buffer fill port.
- Install buffer fill valve. Connect buffer argon fill line. (Note: When installing fill line, a small pressure argon flow should be maintained.)
- Set buffer pressure regulator to 180 psi.
- Open buffer fill valve on T/CNV.

- Bleed the buffer volume pressure out of T/CNV with the buffer regulator valve to 25 psi.
- Close the buffer fill valve on T/CNV. Close argon tank valve and bleed off high pressure argon gas from lines and manifold.
- If additional testing will be made the next day, maintain T/CNV heater at 10 watts. Otherwise, turn power off T/CNV heater and skip the next two steps.*
- *Control T/CNV hot finger temperature at 1200°F with coolant system and 10 watts heater input. Set temperature alarm at 1250°F. Turn off coolant system heater.
- *T/CNV is now in an initial start-up state. (Note: Do not proceed to the next step unless system is to be shut down.)
- Remove buffer fill line.
- Remove buffer fill valve and install buffer seal cap.
- Shut coolant system "Off" after hot finger temperature falls below 300°F.
- Disconnect coolant lines.
- Disconnect heater and instrumentation cables.
- Remove insulation material from crankcase.
- Remove heater tape and thermo probe.
- Place T/CNV in plastic bag and store in upright position.

A.2.5 IMPLANTATION PROTOCOLS

A.2.5.1 First Stage Implantation

A.2.5.1.1 Calf Pre-Operative Workup

This is prepared by the animal experimenters.

A.2.5.1.2 Field Assembly of Pump

(See Section A.2.2.3). This is done prior to implantation.

A.2.5.1.3 Blood Pump Implantation

Preoperative:

- Hemotologic workup on calf.
- Confine calf to cage for 8 to 10 hours and train on the tread mill.
- Keep calf N.P.O. ("nothing by mouth") 24 hours, except for water.

- Neomycin 2000 mg six (6) hours preoperative.
- Lab packs.
- Coupling mechanism is attached to pump; both are filled with oil. Coolant lines are filled with water and shunted. The nose cone (fairing on free end of coupling mechanism) and thermal insulation are in place.

Operation:

- Induce with Brevane and 2 mg. Atropine and place calf on left side.
- Intubate and connect calf to respirator.
- Insert urinary catheter if female.
- Prep and drape and insert pressure lines in left leg.
- Expose the right jugular vein and carotid artery.
- Make incision through the right paramedian abdominal muscle to the peritoneum to receive the electric motor.
- Midsternal split incision and dissect mediastinum.
- Tunnel through diaphragm, under skin to incision made to receive the electric motor.
- Dissect superior vena cava and inferior vena cava and place tourniquet.
- Open pericardium, dissect aorta, pulmonary artery, place tourniquet and preclot grafts.
- Heparinize with standard dose, 5 mg/kg.
- Place bypass lines (arterial line in carotid artery).
- Start partial bypass very slowly, not more than 0.5 L/min. Stay on partial bypass for at least 5 minutes to get adequate mixture of blood and priming solution.
- Total bypass. Ligate hemiazygos vein and excise ventricle.
- Suture both atrial cuffs.
- Suture pulmonary arterial graft.
- Suture aortic graft and check for leaks.
- Pass shaft through the diaphragm and place the blood pump into the thorax.
- Flush ventricles with carbon dioxide.
- Connect electric motor to drive shaft.
- Recharge blood pump lubricating oil.
- Clip coolant shunt to electric motor.

- Tunnel and pass power leads and motor coolant lines through skin and hand to power control operator.
- Connect two atrial plus aortic and pulmonary arterial quick connects.
- Prime ventricles with saline and start pumping.
- Stop bypass and place chest drains.
- Check for leaks, remove bypass lines, and start protamine sulfate; 1.5 x heparin dose. Add 60 MEq. KCI intravenously.
- Close chest, carotid artery, and ligate jugular vein.
- Place calf in cage.
- Initiate vacuum on chest drains.
- Initiate recovery procedure, normothermia and lung ventilation as dictated by blood gases.
- Intensive post-operative care.

A.2.5.1.4 Post Operative Care

This is prepared by the animal experimenters.

A.2.5.1.5 Post-Operative Equipment Monitoring

This is prepared by the animal experimenters. However, the following parameters are desired:

- Real-time motor torque.
- Real-time motor speed.
- Real-time blood flow rate.
- Real-time blood pressure.
- Motor temperature.

A.2.5.2 Second Stage Implantation

A.2.5.2.1 Calf Pre-Operative Workup

This is prepared by the animal experimenters.

A.2.5.2.2 Thermal Converter Check-Out and Sterilization

This work is done prior to implantation. (See Section A.2.1).

A.2.5.2.3 Thermal Converter Implantation

The criteria for implantation is for stable condition using available physiological parameters. The animal is eating and drinking with no O₂ support and standing, a minimum of 48 hours.

Preoperative Procedure:

- The thermal converter has been sterilized and is running in a sterile chamber with the auxiliary coolant system operating, and a starter motor connected to the fly-wheel end.
- Anesthetize and intubate, and surgically prep the animal.

Operation:

- Incise through former abdominal incision and expose the electric motor.
- Examine site and prepare for thermal converter as required.
- Heparinize.
- Remove operating thermal converter, starting cradle and auxiliary lines from the sterile chamber and attach to sterile sling.
- Remove output shaft closure from thermal converter and maneuver cradle and sling to place converter over abdomen alongside incision.
- Remove blood pump electric motor from incision and place on abdomen.
- Peel back Silastic from flexible shaft ferrule and unscrew ferrule, maintaining connection between blood pump motor and shaft manually.
- Stop blood pump motor and stall converter with dummy load. Remove blood pump motor from flexible shaft.
- Connect thermal converter to flexible shaft with auxiliary coolant lines and starter motor connected to converter.
- Start thermal converter with auxiliaries connected by energizing starter motor.
- With sterile oil-filled syringe, engage bleeding port on gear box and charge with sperm oil and remove air creating mild vacuum. Cap self-sealing bleeding port.
- Check thermal converter for normal, stable operation.
- Monitor blood pressures and thermal converter temperatures.
- Remove starter shaft and cap off fitting.
- Check thermal converter for normal operation.
- Disconnect instrumentation leads from auxiliary cart, attach protective covering over leads.

- Remove the thermal converter from cradle and place thermal converter in abdominal cavity, and route instrumentation lines through existing skin tunnel.
- Close abdominal incision and re-connect instrumentation leads to auxiliary cart.

A.2.5.2.4 Post-Operating Care and Monitoring

This is prepared by the animal experimenters; however, monitor hot finger temperature and working volume pressure as follows:

- Monitor the thermal converter hot finger temperature every 60 minutes. The danger level for this reading is 1500°F at bottom of hot finger. Reduce the power supplied to the T/CNV heaters. The normal operating range for the hot finger temperature is 1400° to 1500°. Take a recording of the working volume pressure as outlined in the next step.
- Every TBD minutes or more often as provided for above, provide a two second minimum burst of buffer pressure taken at 200 mm/sec. The converter mean speed can be computed by counting the number of pressure cycles in two seconds. The normal speed is between TBD and TBD cycles in a two second interval. The mean pressure is obtained by averaging the high and low points of the cycle and should be between TBD and TBD.
- Alarm Circuits - The temperature measuring circuitry on the auxiliary cart is fitted with an alarm and control switch, which will sound and remove power to the heaters in the event the temperature exceeds normal operating conditions, or the thermocouple circuit fails. In the event the alarm sounds, read the temperature being measured and reduce the power input to the heaters and record setting. An additional alarm is fitted to the cart which will sound in the event of failure of the 110 volt line. In the event of a power failure, the emergency system will be activated. This system will continue to operate the auxiliary cart until the reset button is pressed to transfer the system back to main line power.

A.2.5.2.5 Equipment Recovery Procedures

A.2.5.2.5.1 Thermal Converter Recovery Procedure

Preparation:

Calf has been terminated, and heparinized to delay clotting.

Recovery:

- Open incision along skin tunnel and remove instrumentation leads; remove umbilical cord intact.

- Open abdominal incision exposing thermal converter.
- Withdraw operating converter.
- Remove closure from starter shaft coupling.
- Using nylon rod, apply pressure on flywheel shaft until T/CNV stalls.
- Withdraw converter from flexible shaft.
- Cut lines free from animal.
- Remove T/CNV from calf.
- Replace closure over output shaft.
- Wipe off converter surface.
- Turn power off.
- Connect fill valve to buffer volume.
- Monitor T/Converter temperature until 1300°F is reached. At this point, reduce buffer pressure to 50 psig.
- Thoroughly clean external surfaces of converter.
- Disassemble and inspect thermal converter as required.
- Converter is ready for next implant.

A.2.5.2.5.2 Blood Pump Recovery Procedure

Preparation:

Calf has been terminated and is lying on his back. Converter has been removed as discussed in Converter Recovery Procedure. Suitable tools are on hand, as is a dust-free environment, such as a laminar flow hood or equivalent.

Recovery:

- Immediately following removal of thermal converter, cap the converter end of the drive shaft to prevent contamination of the shaft and blood pump mechanism.
- Re-open thoracic incision.
- Excise quick connects.
- Remove blood pump and coupling mechanism from thoracic cavity.
- Uncouple quick connects.
- Thoroughly clean external surfaces of blood pump assembly.
- Remove cap from drive shaft and drain sperm oil from pump mechanism.
- Drain coolant system.

- Remove insulation from shaft coupling at blood pump end and from coolant line connections.
- Place blood pump assembly in suitable dust-free environment.
- Remove flexible shaft and coolant shunt from blood pump assembly.
- Remove pusher cups and ventricles from blood pump.
- Install bellows retainers.
- Visually inspect blood pump mechanism for damage or contamination.
- Remove shaft core and inspect for damage or contamination.
- Reassemble pusher cups (without ventricles) to blood pump mechanism.
- Pack blood pump mechanism in approved shipping container and return to Westinghouse for refurbishment, if indicated.
- Blood pump mechanism is ready for re-assembly if no damage or contamination has been observed.

A.3.0 COMPONENT RESEARCH AND DEVELOPMENT

For convenience, component research and development efforts have been classified under five general headings associated with major components or functions, of the nuclear-powered system. These are: the Stirling engine; the thermal insulation; the coupling mechanism; the blood pump mechanism; and the waste heat rejection system. These efforts were a mixture of analysis and experiments into the various detail functions and parts of the major components. The experimental approach was emphasized.

A.3.1 STIRLING ENGINE

The efforts were aimed, largely, at detail problems which engine experience indicated could prevent reliable long-term operation, adequately controlled function, or structural integrity from being achieved. These were: hot finger (engine cylinder) material investigations; high temperature joining of dissimilar materials; engine control system development; magnetic seals and coupling investigations; lubrication system development; and development work on a bellows seal for the engine power piston.

A.3.1.1 Hot Finger Materials Investigation

Almost all the effort in hot finger materials was done on Udimet 700, except for a small literature type analysis search for alternate materials to replace Udimet 700.

Udimet 700

Udimet 700 was developed for use as a high temperature gas turbine blade material. It has the following nominal composition: Cr 15%; Co 15%; Mo 5%; Al 5%; and Ni 55%. Most of the material used was obtained from Special Metals, Inc. The exception was a few bars of powder metallurgically prepared material obtained from Federal Mogul Corp. or Homogeneous Metals, Inc.

The use of Udimet 700 in the artificial heart thermal converter differs from its use in turbines in several respects. First, the dimension of the 4 mil thick cylinder is, by an order of magnitude, smaller than dimensions generally encountered in turbine applications. A second difference between this application and turbines is the operating environment. The atmosphere surrounding the Udimet 700 in turbines is very corrosive, whereas in the artificial heart thermal converter, it is bounded by vacuum and high purity argon gas. The absence of a corrosive atmosphere may have a beneficial effect on the life of the hot cylinder.

The dimensional difference is regarded as the most important as it requires much greater care in preparation, fabrication, and inspection to produce equipment with the capabilities approaching that which can be projected from bulk property data. It is also equally important to know what deviation from bulk property levels can be expected. In this connection investigations into grain size and structure, machining methods, inspection methods, and creep-to-rupture were carried out.

Bulk Properties

Two bulk material properties of great interest to the engine designer are: (1) creep data as a function of stress and temperature and (2) thermal conductivity. The bulk creep properties of Udimet 700 wrought bar stock, as obtained from Specialty Metals, is shown in Figure A.3-1. The test results are correlated using the Larson-Miller parameter (a standard method for predicting the stress rupture behavior of materials at high temperature). The parameter, as derived from the Arrhenius equation, combines temperature and time under stress as follows:

$$P = T (20 + \log t)$$

where:

P = parameter

T = temperature, °R

t = time, hours

A relationship between the value of the parameter P and the stress level required for rupture has been established experimentally. Values reported

at 1400°F for powder metallurgy samples and wrought bar stock compare as follows:

Temperature	Stress	Creep To Rupture	Supplier	Nature of Material
1400°F	85,000 psi	21 hours	Homogeneous Metal	Powder Metallurgy
1400°F	85,000 psi	37 hours	Federal Mogul	Powder Metallurgy
1400°F	85,000 psi	45 hours	Special Metals	Wrought Bar

Figure A.3-1. Rupture Stress Versus Larson-Miller Parameter for Udimet 700

The values for the thermal conductivity of Udimet 700 used in the design of the thermal converter were those published by Special Metals Corp. Later, it was found that the conductivity values published by both the International Nickel Corp. and the Department of Defense (in the Aerospace Structural Metals Handbook) were 35 percent higher than those reported by Special Metals. If true, these higher conductivities would imply 35 percent higher thermal conduction losses in the engine, severely reducing the efficiency potential of the engine. To resolve the question raised by these conflicting reports, a decision was

made to have the thermal conductivity of Udimet 700 measured. A company specializing in primary thermal conductivity measurements, Dynatech, Inc. of Cambridge, Massachusetts, was contacted to perform these measurements on a test specimen fabricated from a 1-1/4 inch diameter wrought bar purchased from Special Metals Corporation.

Figure A.3-2 shows the thermal conductivities reported by Special Metals and International Nickel and also the Dynatech data. As can be seen, the Dynatech data agrees closely with the Special Metals Curve used in our design studies. Further, in the opinion of experts at Dynatech, the non-linear shape of the International Nickel curve is in total disagreement with the theoretically expected, as well as the experimentally verified, characteristic of all nickel-base superalloys. Thus, both the data and the opinion of the experts confirm the validity of the conductivity values used in the engine design.

Udimet Thin Wall Investigations

Different material forming methods, heat treatments, and machining methods were investigated to improve and determine the creep to rupture properties of Udimet 700 in thin wall use. Samples of the various differently treated Udimet 700 were prepared and creep to rupture tests carried out.

Heat Treatment Variations

Treatment No. 1

1095°C		8 hours		Air Cool
1130°C		4 hours		Rapid Cool
870°C		8 hours		Air Cool
983°C		4 hours		Air Cool
650°C		24 hours		Air Cool
760°C		8 hours		Air Cool

Comparison of the grain structure of two wrought bars after this heat treatment and as received are given in Figures A.3-3 through A.3-6.

Figure A.3-2. Thermal Conductivity of Udimet 700

Figure A.3-3. Udimet Bar No. 1 As-Received (100 X)

Figure A.3-4. Bar No. 1 After Heat Treat (100 X)

Figure A.3-5. Udimet Bar No. 2 As-Received (100 X)

Figure A.3-6. Bar No. 2 After Heat Treat (100 X)

Treatment No. 2

1095°C 8 hours Air Cool
1130°C 4 hours Rapid Cool

A powder metallurgically prepared bar from Federal Mogul Corp. was subjected to Treatment No. 2. These bars are much more uniform in their composition than the cast and swaged bars. They are fully dense and exhibit a much smaller grain size than the cast bars. Comparison of the grain structure of the powder metallurgical bars as-received and after heat treatment are given in Figures A.3-7 and A.3-8. It will be noted that even after the heat treatment the grain size remains small.

Treatment No. 3

1175°C 4 hours Air Cool
1080°C 4 hours Air Cool
884°C 24 hours Air Cool

Treatment No. 4

1100°C 5 hours Air Cool
870°C 5 hours Air Cool
982°C 4 hours Air Cool

A comparison in the grain structures of wrought Udimet 700 after Heat Treatments No. 3 and No. 4 are given in Figures A.3-9 and A.3-10, respectively.

Heat Treatment No.4 was that used on the nuclear system engine. Also, a bar of powder metallurgically prepared Udimet 700 from Homogeneous Metals, Inc. was subjected to Heat Treatment No.4. The resulting grain structure is shown in Figure A.3-11.

As a result of experience, it was found that Heat Treatment No. 4 sometimes produced grains larger in dimension than the 0.004-in. engine thin wall and one further heat treatment, No. 5, was investigated (Figure A.3-12).

Figure A.3-7. Powdered Udimet As-Received (100 X)

Figure A.3-8. Powdered Udimet After Heat Treat (100 X)

Figure A.3-9. Heat Treatment No. 3 (100 X)

Figure A.3-10. Heat Treatment No. 4 (100 X)

Figure A.3-11. Heat Treatment No. 4 (100 X)

Figure A.3-12. Heat Treatment No. 5 (100 X)

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 to 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 Heat Treatment No. 5 (1050°C for 10 hours air cool; 870°C for 5 hours air cool). Heat Treatment No. 5 results in a grain structure as previously shown in Figure A.3-12.

Creep-to-Rupture Testing - a test facility for testing thin wall cylinders creep-to-rupture time was constructed. The test facility consists of a tubular furnace through which dried argon is passed as a protective atmosphere. The inside of hot finger samples are pressurized directly from a tank containing very high purity argon.

Figure A.3-13 is a schematic of the high temperature stress-rupture test setup. The apparatus includes an electric furnace with automatic temperature regulation supplied by a feedback control system. Temperature fluctuations can be limited to $\pm 2.5^{\circ}\text{C}$. The stress-rupture specimen is introduced from the right end in Figure A.3-13. A constant flow of argon gas is introduced at the left end to prevent corrosion on the exterior of the hot sample. The argon is first fed through a copper catalyst to remove residual oxygen and then through a molecular sieve to reduce the dew point of the gas to below -40°C . The sample is first heated to temperature, then charged to a predetermined pressure with argon to prevent internal corrosion. The gas charge is sealed in the sample for the duration of the test. Pressure fluctuations of $\pm 1\%$ due to room temperature fluctuations have been observed.

Figure A.3-13. Schematic of High Temperature Stress Rupture Approach

Pressure and temperature are recorded simultaneously on a multi-channel recorder. A loss in sample pressure is indicative of the rupture of the specimen. The time of the failure is obtainable from the recorder's data trace.

Initially the medium outside the test specimen was argon containing 4 percent nitrogen. The furnace was later altered to produce a vacuum on the outside of the specimen to better match actual engine operating conditions. Test specimens either 1 in. ID or 1.35 in. ID and 2.0 to 2.5 in. long were fabricated from these materials. A thin-wall region of nominally 4 mils wall thickness was machined over a length of 1.5 in. The cylinders were fabricated either by single-point machining with silicon-carbide tools or by grinding. Commonly in the first step, a rough cylinder was machined with a wall thickness of 1/16 in. This cylinder was heat treated at 1100°C for 5 hours. After this heat treatment, the inside was final machined and honed. The cylinder was then put on an arbor and the thin wall machined from the outside. Before closing off the cylinder by welding, a final heat treatment was applied at 982°C for 4 hours. All heat treatments were done in dried welding argon and were the same for swaged and powder metallurgical specimens. The common heat treatment will be recognized as No. 4. A typical specimen is shown in Figure A.3-14.

Figure A.3-14. Thin-Walled Udimet 700 Stress-Rupture Specimen

The cylinders were tested at two different temperatures, 750°C and 800°C, at various internal pressures. The internal pressure was chosen to give a particular creep-to-rupture life based on creep-rupture life according to the Larsen-Miller parameter as given by Special Metals Corporation. Due to variations in the thin wall from the nominal 0.004 inch, the predicted life varies from specimen to specimen.

The initial tests compared two (2) samples each of cast and swaged Udimet 700 (S) and powder metallurgically prepared Udimet 700 (P/M). For these tests purified welding argon was used external to the specimen. The results are as follows:

<u>Material</u>	<u>Temperature (C°)</u>	<u>Pressure (psi)</u>	<u>Predicted Life Hours</u>	<u>Measured Life Hours</u>
S	750	680	80	60
S	800	520	190	110
P/M	750	680	100	15
P/M	800	520	100	0

The predicted life is that calculated from thick section creep data and an approximate measurement of test specimen wall thickness. The test results indicate that cast and swaged material has better creep rupture life properties than powder metallurgy material. Scanning electron microscope pictures of the outer surface of the ruptured area, including the crack, were taken of all cylinders. Machining marks were clearly visible. Perpendicular to the machining marks were corrosion marks. Since the density of these marks depend on the length of exposure to the furnace atmosphere during testing, they were clearly visible in samples S750 and S800 which had long exposures. They were less visible in P/M 750 and only visible at high magnification in P/M 800 which had a very short exposure to the argon atmosphere. Samples S800 and P/M 750 also reveal that the corrosion marks are associated with plastic deformation. The corrosion marks are most dense in the neighborhood of cracks. Away from areas where plastic strain has occurred, no corrosion marks are observed. A mass

spectroscopic analysis of the purified welding argon showed that it was a mixture of 5 percent nitrogen and the remainder argon. No other gases were found, even as traces. It was concluded that the nitrogen was the cause of the corrosion observed.

As can be observed, sample P/M 800 failed quickly. Figure A.3-15 shows a micrograph of the inside of the hot cylinder. No corrosion is present due to the use of high purity argon. Honing marks are clearly visible, especially one that ends at the crack. Upon further investigation with the non-dispersive energy x-ray analyzer (see Figure A.3-16), the presence of silicon was confirmed. The silicon is due to a particle of the honing stone that has become embedded in the thin wall. Subsequently, the silicon-carbide particle became the crack nucleation site for fracture. The silicon-carbide particle can, in turn, explain the low creep rupture life of specimen P/M 800. A similar situation was present at the bottom of a scratch in specimen S800. A non-dispersive energy x-ray analysis also revealed silicon at the bottom of the scratch, which can be due to the carbide tool. In sample S800, the crack nucleation associated with the silicon-carbide particle was not quite as obvious as with sample P/M 800.

0.1 mm
4 mils

Figure A.3-15. SEM Micrograph of the Inside of Specimen P/M 800

Figure A.3-16. Enlargement of the Hole Shown in Figure A.3-15

Left: Non-dispersive x-ray spectrum of Udimet 700 away from hole (dots) and spectrum of material inside hole (bars).

Lower Right: Map of the same area as above using the silicon peak of the non-dispersive x-ray spectrum.

Because of the corrosion observed on the early specimens, the creep-rupture test apparatus was modified to provide a high vacuum environment external to the creep-rupture specimens. Glass-to-ceramic seals were made that allowed the evacuation of the furnace down to 10^{-6} torr at 750°C.

The emphasis of the creep-to-rupture investigation was shifted from comparisons between swaged and power metallurgy materials to possible improvements in the machining procedure and to instrumentation for better control of the thin wall. The machining procedure comparisons were between preparation of specimens by single-point machining followed by honing or preparation by grinding. During these early tests it became apparent that changes in the 0.004 in. wall thickness of half a mil (0.0005-in.) have a tremendous influence on the life of the cylinder. An air gage manufacturer was therefore contacted and a gage designed specifically for the measurement of the wall thickness of the hot cylinders. A picture of the gage is shown in Figure A.3-17. It consists of a spindle over which the hot cylinder fits snugly. The spindle contains an air jet. An arm brings another air jet directly over the first one so that the wall is between the two jets. With this device, the wall thickness can be measured reliably to +0.0001-in.

The purchase of the new thin-wall gaging system made it possible to accurately measure wall thickness for the first time. The gage was used to measure the wall thickness of previous creep-rupture specimens and this information was in turn used to recalculate actual stress levels and predicted lifetimes. (Previously, the nominal wall thickness had been used in these calculations.) The re-evaluated data are compiled in the next following unnumbered table. In this table the wall thickness given is the one measured at the thinnest portion of the wall.

For the second series of tests, only wrought Udimet 700 manufactured by Specialty Metals Corp. was used for the fabrication of the test cylinders. All the tests were performed in vacuum of less than 10^{-5} torr. This is in contrast to the first series of tests that were performed under a protective argon atmosphere. No oxidation of the test specimen is observed in vacuum. The air gaging system allowed measurement of the thin wall to within 0.1 mil. With knowledge of the exact dimensions of the wall, the test pressures were adjusted to give the stresses which result in a predicted life of 100 hours. These tests are summarized in Table A.3-1.

Figure A.3-17. Thin-Wall Thickness Gaging System

Creep Rupture Tests of Udimet 700 Cylinders

Specimen Number	Test Temperature (°C)	Minimum Wall Thickness (mils)	Hoop Stress (ksi)	Predicted Life (Heavy Sections in Air) (hours)	Actual Life (hours)
<u>Wrought and Swaged U700 Tested in Argon + 4% Nitrogen</u>					
1	750	3.6	94.5	12.5	60
2	800	5.2	50.0	560.0	110
<u>Powder Metallurgical U700 Tested in Argon + 4% Nitrogen</u>					
3	750	4.6	74.0	190.0	15
4	800	4.1	63.5	80.0	0

The first group of specimens of Table A.3-1 consisted of 1 in. diameter test cylinders that were conventionally machined with single-point tools and honed on the inside. The following comments apply to this group:

- Specimen No. 5 tested at 750°C was fabricated before the arrival of the air gage, and therefore its wall thickness could not be measured before testing. Examination of the outside of the specimen in the scanning electron microscope (SEM) showed a single crack. The crack could not be associated with a particular defect in the wall although on both the inside and the outside the remains of embedded alumina particles could be found. The alumina particles are assumed to stem from the machining tools used.
- Specimen No. 6 tested at 800°C was measured carefully and pressurized for a nominal life of 100 hours. It achieved 50 percent of this goal which can be considered satisfactory. Observation in the SEM showed a widely spread crack area, an indication that no defect in the wall was responsible for failure.
- Specimen No. 7 tested at 750°C and designed for 1000 hours life showed a very short actual life. An examination in the SEM revealed a very porous structure on the inside of the thin wall in the crack area. Energy dispersive analysis of X-rays showed the presence of chlorine which is difficult to explain. It seems that a material defect caused the crack initiation and in turn is responsible for the short life.

TABLE A.3-1

UDIMET 700 CREEP RUPTURE TEST SUMMARY

Specimen Number	Test Temperature (°C)	Wall Thickness		Pressure (psi)	Hoop Stress (ksi)	Designed Life Heavy Sections in Air (hours)	Actual Life (hours)
		Low (mils)	High (mils)				
Wrought and Swaged U700 1 in. Diameter Samples, <u>Single-Point Machined, Heat Treatment No. 4</u>							
5	750	3.5*	0	680	97.5	8	34
6	800	3.7	(+0.9)	460	62	100	49
7	750	4.0	(+0.6)	480	60	1000	2.5**
8	750	3.5	(+0.4)	441	60	1000	159
9	800	2.9	(+1.1)	273	47	1000	46
Wrought and Swaged U700 1.35 in. Diameter Samples, <u>Single-Point Machined, Heat Treatment No. 4</u>							
10	750	3.3	(+0.9)	393	80	100	10.5
11	800	3.3	(+0.9)	303	62	100	0.2**
Wrought and Swaged U700 1.35 in. Diameter Samples, <u>Ground, Heat Treatment No. 4</u>							
12	750	3.2	(+1.7)	380	80	100	6***
13	800	3.5	(+0.6)	322	62	100	8
Wrought and Swaged U700 1.35 in. Diameter Samples, <u>Heat Treatment No. 5</u>							
14	750 ⁽¹⁾ ****	3.75	(+0.75)	444	80	100	23
15	750 ⁽²⁾ ****	3.25	(+1.00)	385	80	100	12

*Only measured after testing.

**Material defects or inclusion observed.

***Large eccentricity.

****Identical specimens; however,

(1) Ground, and

(2) Single-point machined.

The second group of specimens consisted of 1.350 in. diameter test cylinders that were conventionally machined with single-point tools and honed on the inside. Of these,

- Specimen No. 10 tested at 750°C and designed for 100 hours of life achieved only 10 percent of this goal. Observation in the SEM revealed a very small crack that could not be seen on the inside. No embedded particles could be found. It is therefore assumed that this is a representative test.
- Specimen No. 11 tested at 800°C showed an alumina particle associated with the crack. It is therefore assumed that the embedded particle was responsible for crack initiation and the short life observed.

The third group of specimens consisted of 1.350 in. diameter test cylinders that had been ground on both the inside and the outside.

- Specimen No. 12 tested at 750°C was designed for 100 hours life and achieved only six hours. This less than expected actual life might be due to the large eccentricity measured on this specimen. The crack area was spread very widely and no embedded particles were found.
- Specimen No. 13 tested at 800°C failed about the same as No. 12, and cast doubt on the hypothesis that the early failure of No. 12 was because of eccentricity.

The last group of specimen consisted of 1.350 in. diameter test cylinders subjected to Heat Treatment No. 5 instead of Heat Treatment No. 4. Both specimens were tested at 750°C at stress levels designed to yield 100-hours life (based on thick-section data); one was ground, the other machined. The finished dimensions of the ground specimen were closer to perfection than any previously tested thin-wall. The wall thickness was highly uniform, with a minimum thickness of 3.75 mils and a variation of 0.4 mils. The specimen ruptured after 23.3 hours under test. Figure A.3-18 shows the specimen after failure. Note

Figure A.3-18. Ground Udimet 700 Test Specimen After Creep-Rupture Test

that creep has been distributed over the entire surface with no "lobing," a result of the uniformity of the wall thickness. The machined specimen had a minimum wall thickness of 3.25-mils with a variation of 1.0-mils; clearly not as uniform as the ground specimen. This specimen failed after 11.6 hours under test.

The results compiled in Table A.3-1 for Specimens 8 through 15, excepting 11, are plotted in Figure A.3-19. The observed percentage of predicted life is shown as a function of wall thickness. The creep-rupture life is strongly dependent on the wall thickness. The creep-rupture life is not a function of the diameter of the test specimen or the machining procedure, within the accuracy of the measurement. To ensure the ten year life of the prototype heart engine, only seven percent of the predicted thick section life is conservatively required. A strict dimensional control of 0.004 $\begin{smallmatrix} +0.0004 \\ -0.0000 \end{smallmatrix}$ will guarantee at least three times the required seven percent of the thick section life, as can be seen from Figure A.3-19.

An area of concern is quality assurance of the Udimet 700 hot cylinders, because it has been possible to associate the failure of some short-lived creep-rupture specimens with mechanical flaws in the thin-wall. A search was initiated for a suitable method of inspection and non-destructive testing of the thin-wall. A number of non-destructive test techniques were tried with negative results as follows:

- A radiography (x-ray imaging) test was performed but no inhomogeneities were found.
- An eddycurrent test was performed with the same results.
- A fluorescent liquid penetrant test was performed and gave an indication of defects. However, optical examination of the sites where the "dots" appeared revealed a visible surface blemish in only one of the sites.

In summary, radiography and eddycurrent methods appear to be unsuitable for the location of minute surface defects. The fluorescent liquid penetrant does not appear quite satisfactory because of the shallow nature of the surface defects. At present, hermeticity testing and optical inspection of surfaces are the best available inspection techniques and refinement of the latter method is required.

Figure A.3-19. Percent of Thick-Section Udimet 700 Creep-Rupture Life Versus Minimum Wall Thickness

Ceramic Materials

The choice of materials for the hot finger of the artificial heart engine is severely limited by the two principal design requirements which the cylinder must fulfill:

1. It must maintain hermeticity for ten years while stressed by the working gas charge in the engine.
2. It must minimize conduction losses from the hot side to the cold side.

The material used successfully in present designs is Udimet 700, a nickel base superalloy. It has a creep rupture strength of 62 ksi or 440 MN/m^2 at 750°C and a thermal conductivity of $0.16 \text{ W/cm-}^\circ\text{K}$. A figure of merit for candidate hot finger materials can be defined as the ratio of creep rupture strength to thermal conductivity. For Udimet 700, one obtains $27.5 \frac{\text{MN-}^\circ\text{K}}{\text{mW}}$. This figure of merit can be used as a baseline for the evaluation of alternate hot finger materials.

Ceramic materials would intuitively seem to be well suited for the hot finger because they are frequently used in applications where low thermal conductivity is required. A close examination, however, reveals that the insulating ceramic materials are quite porous and, therefore, not suitable for the application at hand. A number of fully dense ceramic materials have been examined further and the relevant properties have been compiled in Table A.3-2.

For structural ceramics like Alumina, SiC, and Si_3N_4 , the strength properties are well known. While the rupture strengths are high, these materials also have very high thermal conductivity. The figure of merit shows they represent no advantage over Udimet 700. In contrast, the remaining non-metallic materials are characterized by extremely low thermal conductivity and significantly lower rupture strengths. While the figure of merit for these materials appears attractive, this is because the properties are given for room temperature. The creep properties of these materials at elevated temperatures are little known. Indeed, weakening due to temperature may rule these materials out for service at 750°C .

TABLE A.3-2
PROPERTIES OF POTENTIAL HOT FINGER MATERIALS

Material (Temperature)	Creep-Rupture Strength (MN/m ²)	Thermal Conductivity W/cm-°K	Strength/ Conductivity Ratio $\frac{\text{MN-°K}}{\text{mW}}$
Udimet 700 (750°C)	440	0.16	27.5
SiC (750°C)	400	0.50	8
Si ₃ N ₄ (750°C)	400	0.28	14
Alumina (750°C)	300	0.20	15
Quartz (25°C)	50	0.014	36
Glass-Ceramic (25°C)	50	0.014	36
Zirconia (25°C)	80	0.020	40

To investigate the quartz tubing more closely, two test cylinders 1 in. in diameter with a 0.040 in. wall were manufactured with one end closed by a flat quartz disc. The cylinders were tested at room temperature and ruptured at a hoop stress of 17 MN/m², significantly lower than the values reported in the literature. It is difficult to see how substantial improvements over Udimet 700 can ever be realized with quartz. The prognosis for glass-ceramic and Zirconia is the same. Industrial sources are pessimistic as to their suitability, and it was impossible to procure closed-end sample cylinders for evaluation.

3.1.2 Metals Joining Investigations

Nickel 200 to Udimet 700

The hot finger of the engine has a Nickel 200 to Udimet 700 joint on the hot side which is microbrazed. A metallurgical investigation was conducted to determine to what extent the brazing filler metal diffused into the parent metals. A sample joint, shown in Figure A.3-20, was prepared under conditions equivalent to those used on the actual hot finger assembly. Following the completion of the braze, the joint was metallurgically investigated with the aid of the photomicrographs shown in Figures A.3-21 and A.3-22. As these

Figure A.3-20. Sample Braze Joint

Figure A.3-21. 35 Power Micrograph of Nickel-Udimet

Figure A.3-22. 270 Power Micrograph of Nickel-Udimet

indicate, the microbrazed diffusion into the Nickel 200 is pronounced, but the part is not critical in terms of stress. The diffusion into the Udimet 700 side is minimal, and would probably pose no long range problems.

Monel 400 to Udimet 700

Completion of the nuclear engine hot finger was delayed by a problem in accomplishing the braze joint between the Monel 400 flange on the ambient temperature end of the finger and the Udimet 700 outer wall. This braze had to be carried out at a higher temperature than that used in the previous bench model because the Udimet 700 heat treating procedures selected necessitated a final heat treatment at 982°C following brazing and final machining. The braze in question uses Microbrazed 10 alloy. Initial samples showed that the phosphorous in the braze metal was alloying with the copper content of the Monel 400, creating a low temperature eutectic alloy which diffused away from the braze region. The diffusion effect left voids at the contact surface which could cause a loss of hermeticity to appear with time. This porosity could have been catastrophic for the vacuum integrity of the permanently sealed vacuum insulation package.

The approach to this problem was to deposit a barrier layer on the Monel 400 flange to prevent diffusion of the phosphorous. Stainless steel was tried but did not work. Cobalt was tried and found to work, but the difference in thermal expansion between the substrate and the cobalt was found to cause cracking if too thick a layer was applied. The solution to the problem was to use a thinner cobalt barrier layer in conjunction with a thin molybdenum layer.

Titanium-to-Titanium and Titanium-to-Udimet 700

The intention of using titanium alloys extensively in a prototype heart engine necessitated the development of joining techniques for these alloys.

Joints of titanium using silver and silver-copper eutectic are fully described in the literature. Silver is an attractive material since the compounds that it forms with titanium are quite ductile and strong. The brazing temperature of 980°C, however, is well above the transus temperature of titanium. When titanium is heated to about 950°C, the transus temperature, an alpha-beta structure develops upon cooling. This structure may be undesirable for some

applications. Although the silver-copper eutectic flows well at 800°C, well below the transus temperature, certain copper-titanium compounds are brittle and may result in weak joints.

Gapasil-9, a product of Western Gold and Platinum, consisting of an alloy of gallium, palladium and silver, melts at 900°C and is compatible with titanium alloys. Several test samples were made and brazed with Gapasil-9. Base pieces consisted of titanium-to-titanium and titanium-to-Udimet 700. Good metallurgical bonds have been made in both instances and the method is recommended for joining these alloys. Figure A.3-23 shows the bond of Ti-5 Al-2.5V to itself. Figure A.3-24 presents a joint of titanium to Udimet 700. In both cases, the reaction or diffusion zone is somewhat harder than the base material. This is indicated by the somewhat shorter indentures made by the Knoop hardness test probe in the diffusion zone as compared with Knoop imprints in the base material.

Bronze and Titanium

Bronze bushings in titanium alloy structures look attractive for use in a prototype engine rhombic drive mechanism. Prior attempts to join bronze bushings to structures such as the connecting link by using plastic cements or soft solders were all unsatisfactory. The expansion coefficients of titanium and bronze are so different that braze joints attempted by conventional techniques resulted in a failed joint. The bronze member, on cooling from braze temperature, would pull away from the titanium housing, producing a crack in the braze joint. This condition was eliminated, however, by the use of a coining, or restraint, technique. A tapered hole was made in each bushing and a tapered titanium pin was set in place. In this manner, the bronze bushing was prevented from pulling away from the titanium housing on cooling from braze temperature because of the tight fitting titanium pin in the center of the bronze bushing.

Pure Copper and Pure Iron

A structure consisting of alternate layers of pure copper and pure iron is needed as a portion of the magnetic circuit for a magnetic fluid piston seal

Figure A.3-23. Titanium Alloy/Titanium Alloy Braze Joint, 400X

Figure A.3-24. Titanium Alloy/Udimet 700 Braze Joint, 400X

prototype. In order to develop and maintain dimensional tolerances, a non-liquidus bonding technique was selected for joining the copper and iron. Weight (pressure) during bonding is necessary to bring varying surfaces into intimate contact. No extrusion of material as a result of pressure is necessary or desirable. Prefired rings of copper, 0.002-in. thick, and Armco iron, 0.013-in. thick, were stacked on a jig. Pressure was applied as a dead weight. The assembly was vacuum fired for 1-1/2 hours at 1040°C. A portion of the ten-member laminate is shown in Figure A.3-25 (40X). Figure A.3-26 is an enlargement (400X) of a copper-iron interface. A good joint interface is indicated.

Hardcoating of Titanium and Udimet 700

Experimental work on coating Ti-6Al-4V and Udimet 700 with electroless nickel to produce hard wear-resistant surfaces was successful.

The titanium alloy, as 6 mm diameter rod, was carefully degreased and coated for about two hours and heated at 450°C for thirty minutes in vacuum at 10^{-5} torr to develop maximum hardness according to accepted practice. A representative sample was taken for metallographic examination and hardness measurements. The uniform thickness of the coating ($39 \mu\text{m} \pm 6$ percent) and its excellent bond to the base metal are shown in Figure A.3-27. A few radial cracks were observed in the coating, one of which is visible in the photomicrograph. There is no indication that these cracks affected the bond of the electroless nickel to the titanium alloy. Knoop microhardness values of $1390 \text{ kg/sq-mm} \pm 8$ percent were measured, which is approximately equivalent to a Rockwell C hardness of 73. These results indicate that optimum conditions were achieved for the deposition and hardenability of the electroless nickel.

A second sample of the titanium alloy, coated for only one hour, was heated at 700°C for one hour in a vacuum at 10^{-5} torr. The thickness of the coating was $15 \mu\text{m} \pm 14$ percent. The Knoop microhardness value of $753 \text{ kg/sq-mm} \pm 20$ percent, approximately equivalent to Rockwell C hardness of 61, is slightly lower than expected. This may have been due to diffusion.

Figure A.3-25. Fe-Cu Laminated Pole Face Structure (40 X)

Fe

Cu

Fe

Figure A.3-26. Fe-Cu Laminated Pole Face Structure (400 X)

Figure A.3-27. Electroless Nickel on 6Al-4V Titanium Alloy

A disk of Udimet 700 was coated for two and a quarter hours and heated in air at 400°C for one hour. The flat surfaces were ground to remove the electroless nickel. The disk was again coated and heat-treated under the same conditions. As a result, the flat surfaces of the disk were coated with a single layer, but the cylindrical surface was coated with a double layer. In addition, a portion of the coating on one flat surface was ground off to determine if grinding would adversely affect adherence of the electroless nickel to the Udimet 700. Figure A.3-28 shows the uniform thickness of the double layers which individually measure $55 \mu\text{m} \pm 5$ percent thick. The cracks in these layers do not affect either the bond to the Udimet 700 or to the adjacent layer. Grinding did not affect the bond of electroless nickel. The Knoop microhardness value of $1000 \text{ kg/sq-mm} \pm 6$ percent is consistent with values cited in the literature. It is approximately equivalent to a Rockwell C hardness of 70.

These results indicate that optimal conditions had been achieved for deposition and hardenability of electroless nickel. As a result, these coatings are recommended for displacer rods and the displacer ring wear surfaces on future engines.

300 ×

Figure A.3-28, Electroless Nickel on Udimet 700

Welding of Stainless Steel Mesh

Techniques were developed 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. Figure A.3-29 illustrates the seam weld and end triple weld; and Figure A.3-30 shows the seam weld in greater detail.

Regenerator Liner Fabrication

An advanced fabrication technique for regenerator liners was investigated. It consists of using 0.5 mil thick Inconel X-750 foil wrapped into a cylinder with an overlapped seam. The seam would be ultrasonically welded. Liners have previously been machined from bar stock with a final honing operation to reduce the wall to 2 mils thickness. This is not only costly but results in a thicker wall than required to sustain the cyclic negative buckling pressure of 1.0 psi placed across it by the regenerator pressure drop.

Figure A.3-29. Oil Filter Assembly Weld Specimen

Figure A.3-30. Seam Weld of Mesh Pleat

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

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

$$P = 0.807 \frac{Et^{2.5}}{Lr^{1.5}} \left(\frac{1}{1 - \nu^2} \right)^{3/4}$$

where:

P = buckling pressure, psi

E = elastic modulus, psi

ν = Poisson ratio

r = cylinder radius, inches

t = foil thickness, inches

L = cylinder length between stiff round supports, inches

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

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

Titanium-to-Aluminum

Forming a good titanium-to-aluminum joint proved difficult. A successful technique was to clad titanium (Ti-5Al-2.5Sn) with aluminum (1100) by vacuum melting to form a hermetic joint. Then the aluminum was joined to aluminum by vacuum brazing.

A.3.1.3 Engine Control System Development

The principles of the "short-circuit" engine control have been covered previously in Volume I, Section 3.2.2. The discharge pressure of an engine-driven oil pump actuates a short-circuit valve which connects the working space of the engine with the buffer space. Opening of the valve results in an increased, but controlled internal leakage loss, which effectively reduces the output shaft torque at a slightly lower heat consumption per revolution. Thus, if the speed control system is designed with the proper gain, the heat input will remain constant, since the off-speed conditions then exactly compensate for the off-consumption of heat per revolution.

For the "short-circuit" system the oil pump must be capable of meeting demanding control requirements since the speed control valve regulates the thermodynamic output power of the engine in response to oil pressure changes. In this regard, the oil pump is the key component because its output pressure is the absolute speed reference for the engine. All external factors which might affect engine speed are automatically compensated by the oil pressure/speed feedback. The pump pressure must therefore be extremely stable with time or the set point (nominal) engine speed will drift. Consequently, an adequate break-in period for both engine and pump is required. The break-in must reduce pump scroll and engine bearing wear to near zero.

Control System Analysis

The differential equations describing the operation of the speed control system were worked out. The characteristic function which determines stability shows a third order system with very interesting properties. The important results of this analysis follow and some of the non-obvious implications are discussed.

The following legend will help in the discussion where all lower case letters represent changes from equilibrium (or the nominal value), and capitals denote the set point values.

LEGEND	
q = oil flow rate change (in ³ /sec)	p ₁ = bellows oil pressure change (lb/in ²)
p = pump oil pressure change (lb/in ²)	k = valve spring constant (lb/in)
ρ = oil density (lb/in ³)	L = oil line length (in.)
g = gravity (386-in/sec ²)	I = oil line inductance
A = valve bellows area (in ²)	$\frac{\rho L}{A g} = \left(\frac{\text{lb-sec}^2}{\text{in}^5} \right)$
K ₁ = engine characteristic (in.-lb)/(rad/sec)	C = pump gain psi/(rad/sec)
K ₂ = controller characteristic (in ³)	t = engine torque change (in-lb)
$\bar{K}_2$ = valve characteristic (lb)	t _L = load torque change (in-lb)
$\bar{K}_2 = K_2 k / A$	J = system inertia reflected at output shaft (in-lb-sec ²)
R = oil line resistance (lb-sec/in ⁵)	ω = change in engine speed (rad/sec)
s = complex Laplace operator	T _L = constant load torque
f = valve spring force change (lb)	
x ₁ = valve poppet motion (in.)	

As can be seen, many variables must be taken into account. K₁ is the slope of the torque speed curve for the engine at the operating point with the control valve fixed. As speed increases, torque will decrease for a constant control valve position. K₂ is the linearized effect of the controller. As bellows pressure, p₁, increases, the valve opens and engine torque decreases proportionately from the operating point. Values for K₁ and K₂ can be experimentally determined on a working engine to improve theoretical estimates. $\bar{K}_2$ is solely a function of engine characteristics and valve seat geometry.

Seven simultaneous differential equations can be written and combined to form the Laplace transformed block diagram in Figure A.3-31. The figure shows the set point speed setting, f , as the system input reference; where f is the force exerted by the valve spring. This figure is of only academic interest because once the speed is set it is never changed; f is zero and the load torque, t_L , is the external variable affecting engine speed. Figure A.3-32 is the primary, or functionally useful, block diagram relating output speed to the input blood pump load torque. As can be seen, the speed control valve acts purely in the feedback path in this mode along with the oil pump. The system rotary inertia J is in the open loop or controller mode with its direct influence. Since the engine flywheel is geared up 3 to 1, its reflected inertia is magnified nine times and far exceeds the inertia of other system components.

It can be shown for the system in Figure A.3-32 that the steady state speed error ω_{ss} for a constant load torque error of magnitude T_L is given by:

$$\omega_{ss} = \frac{-T_L}{\left(K_1 + \frac{AC}{k} \bar{K}_2\right)}$$

Since K_1 is small, it can be neglected and the valve quantity (A/k) should be made as large as possible to decrease the steady state error, given C and $\bar{K}_2$.

For Figure A.3-32 it can also be shown that a necessary condition for stability is that:

$$k > \frac{A \left(C\bar{K}_2 I - AK_1 R^2 \right)}{JR - K_1 I}$$

where K_1 and $\bar{K}_2$ are both positive.

Heart engine values approximate:

$$C = 2.0$$

$$J = 0.05 \text{ (flywheel)}$$

$$A = 0.075$$

$$\left. \begin{array}{l} I = 0.05 \\ R = 1500 \end{array} \right\} 0.093 \text{ dia. x 4-in. long oil tubing}$$

$$k > \frac{6.4 - 1690}{7.5 - 0.005}$$

SET POINT SPEED INPUT

Figure A.3-31. Generalized Proportional Control System

LOAD TORQUE INPUT

Figure A.3-32. Control System Block Diagram for Blood Pump Torque Input

Estimated values are:

$$K_1 \approx 0.01$$

$$K_2 \approx 64$$

The following comments are pertinent:

1. The control system is stable for all values of k since the numerator is negative.
2. The ratio A/k may be made as large as practical in the design of the valve to decrease the steady state error. This, however, places an added burden on oil pump stability because a small oil pressure drift will cause a proportionately large speed drift. Use of too large A , however, is prohibited as the oil pump cannot both deliver instantaneously high flow rates and still supply the bearings. Sufficient gear pump capacity for the desired gain A/k under transient conditions must be considered.
3. If too large an oil line tubing diameter is used, I and R will become smaller and k must then increase as the cube of the tubing diameter, thereby limiting the stable gain.
4. Note that if too small a tubing diameter is also employed, I and R will become quite large; in which case, neglecting $K_1 I$ gives:

$$k > \frac{A^2 K_1 R}{J}$$

where R increases inversely with the 4th power of the tubing diameter. Response time will also increase as less oil will flow into the bellows per unit time.

5. One may conclude from 3 and 4, above, that an optimum range of tubing diameter exists where k/A can be minimized, thus minimizing steady state error according to the speed error equation.

"Short-Circuit" Control Valve Development

The initial control valve underwent three major modifications during the course of development to the valve used as the implantable electrically heated engine control. These will be designated as Mods 0, 1, 2, and 3.

Mod 0

The assembled Mod 0 valve is shown in Figure A.3-33 along with a cross-section drawing of the valve. The speed control system utilizes the valve to increase engine power, when the oil pump pressure decreases (speed decreases slightly),

Figure A.3-33. Control Valve Mod 0

by closing the valve. Upon a decrease in load torque, the engine will speed up, oil pressure increases and the valve opens. The speed band from full load to no load is called the proportional band and is theoretically a function of the bellows area, spring rate, valve seat geometry and pump output (psi/rpm). On a practical basis, hysteresis effects must be kept within acceptable limits because they increase the proportional band. Laboratory tests were conducted to obtain the actual spring rate of the assembly, effective bellows area, poppet positive versus oil pressure, flow rate versus poppet position and oil pressure, and valve leakage.

Valve test data showing how gaseous flow rate through the valve varies with speed control oil pressure is shown in Figure A.3-34. An important result of the data is a plot of flow rate through the valve, Q , as oil pressure, P , is varied. The significance here is that increased Q is related to decreased engine power and increased pressure is linearly related to engine speed. While gradually increasing oil pressure, flow rate was measured with a flowmeter. At a theoretically estimated 1000 cc/min flow (anticipated engine power reduction of 3 watts), the oil pressure was reduced and reduction in flow noted. Figure A.3-34 shows this path transversed by the use of arrows on the solid path. This "envelope" was found to enclose all other paths taken and is therefore the boundary, or "worst condition" of flow control. A typical loop is shown by the dotted path. The hysteresis loop may be transversed from one point to another as speed changes. An engine speed scale is shown for convenience. Oil pump test data was used for this scale determination based on an actual pump gain of 0.2 psi/rpm.

A 4 percent power loss line is shown for reference near 200 cc/min of valve flow. This is seen to intersect the hysteresis loop at 115 psi. Relative to maximum oil pressure, this corresponds to a speed change of about 75 rpm. This hysteresis effect is significant because the theoretical proportional band is only 30 rpm. This is verified by the small hysteresis noted when the valve is fully open. Hysteresis is much more pronounced for a nearly closed valve and is of greater significance here as full engine power capability without a significant speed reduction is desirable. The primary factors affecting hysteresis when the valve is just opening are:

Figure A.3-34. Short-Circuit Control Valve Performance

- Poppet shaft seal friction.
- Poppet/seat misalignment and friction.
- Nonlinear valve opening versus flow characteristic.

The last factor stated above is a major contributing factor in creating the large hysteresis evident at low flows. This is due to the fact that large changes in poppet position do not result in large changes in flow. Figure A.3-35 is a plot of flow rate for various poppet openings as measured with a non-contact displacement transducer. When plotted on log-log coordinates, Q is seen to fit rather well with valve opening raised to the 4th power. Isothermal laminar flow calculations predict a cubic relationship which does hold at very small valve openings. The velocity profile over most of the range is not fully developed due to the short conical seat length.

The Mod 0 valve was tested only under simulated conditions because some internal epoxy joints developed leaks while being installed on an engine. The valve was disassembled and the previous epoxied parts reworked to braze and soldered joints for better reliability. The valve was tested on a bench model engine with the following results:

- When actuated, the valve was capable of varying engine power, as predicted.
- The sinusoidal working space pressure variation tended to lift the poppet off the valve seat at a 10 Hz rate when it should have been closed, thereby seriously reducing engine power.
- Valve hysteresis and the effective unity mechanical advantage between oil bellows force and poppet forces were not sufficient to seat the poppet reliably nor actuate the valve smoothly.
- Friction and leakage associated with the rulon seal are not easily overcome in this design.

Mod 1

From the knowledge gained with the Mod 0 valve and through experimental verification of a poppet motion vibration analysis, a Mod 1 valve configuration was designed to eliminate the Mod 0 shortcomings. In Mod 1, the oil pressure bellows applies its force to a crank arm, thereby rotating a lead screw to which is attached the valve poppet. The poppet, in turn, moves linearly in response to

Figure A.3-35. Flow Through Short-Circuit Control Valve Versus Poppet Opening

oil Pressure (engine speed). A specially designed anti-backlash nut remains stationary in the valve housing. Its helix angle is designed such that the lead screw rotation locks under the sinusoidal working space pressure force on the poppet. There are no relative motion seals in the new design. This was accomplished by referencing the oil pressure to engine buffer space pressure rather than to the crankcase pressure, as did the Mod 0 design. Although the buffer space pressure varies $\pm 1\%$ due to the piston motion, it is smoothed out 5 to 1 by using a restriction in the buffer line tubing leading to the valve. This flow resistance times the internal volume of the valve (which acts like electrical capacitance) effectively filters out the ac fluctuation of 2 psi within the valve body, but freely passes the dc flow into the engine buffer space for control purposes. This is highly desirable for another reason; the flow, and therefore valve rate of response, is slowed down by the high frequency filtering; thereby not permitting the valve to react to the normal speed fluctuations occurring in the engine at 120 rpm due to the finite flywheel size and blood pump loading. If the control valve quickly matched this rate of response, energy would be dissipated two times per second to control engine speed. Full 100 percent engine output power could never be achieved. This design permits the flywheel to independently deliver or absorb transient energy as demanded by the pump or engine.

The various parts are illustrated in Figures A.3-36 through A.3-39. Figure A.3-36 shows the antibacklash nut and lead screw. By counter-rotating the nut halves and then locking, backlash is removed. Backlash of less than 10 micro-inches was measured. Figure A.3-37 shows the bellows subassembly and connecting rod. A tension spring was chosen for low spring rate with freedom from buckling. Figure A.3-38 shows the main valve parts positioned as they belong in the body, and Figure A.3-39 shows a completed valve.

The set-point speed adjustment is accomplished by turning the adjustment screw "stop" of the tension spring. By allowing the buffer space pressure to push on the piston and stretch the spring further, the set-point engine speed is reduced.

The lead screw actuator valve (Mod 1) was bench tested and met its design specifications.

Figure A.3-36. Lead Screw Assembly

Figure A.3-37. Bellows Assembly

Figure A.3-38. Parts Layout

Figure A.3-39. Completed Valve

Figure A.3-40 gives hysteresis and gain test results showing argon flow versus oil pressure. The second abscissa scale gives equivalent engine speed for various oil pressures. The results of two tests are shown, one for a differential pressure of 25 psi, the other for a 5 psi differential.

The hysteresis loop was obtained by gradually increasing the oil pressure and recording the short-circuit flow rate up to the maximum valve needed for full engine power reduction, and then reducing the oil pressure until near zero flow was again obtained. As can be seen, typical hysteresis is less than one psi, or approximately 3 rpm. The "gain" of the device is also large as noted by only a 30 rpm speed band from full to near zero flow at a 25 psi differential. This is accomplished with a 6 psi pressure change. The engine speed range anticipated is therefore only 30 rpm from full load to the minimum load condition. The poppet leakage was found to be only 100 cc/min (STP), which corresponds to approximately 0.6 percent engine production, or 30 mW. The radial play designed into the lead screw thread (while maintaining zero backlash) allows the poppet to gently align itself with its seat, achieving a tight seal without having to crank the poppet closed tightly. This feature was instrumental in achieving the low hysteresis levels.

A test fixture simulating the oil passageway resistances and inductances in the engine, Figure A.3-41, was made in order to impose a variable frequency sinusoidal oil pressure into the control valve to determine its transient response characteristics. A motor driven piston is employed to produce a sinusoidal nitrogen pressure on the horizontal oil surface which terminates inside a section of clear plastic tubing (1/16-in. ID), leading to the oil filter and control valve. The observed displacement of the oil level in the tubing, which is proportional to the actual bellows or poppet motion, was recorded. Variations in flow rate output of the valve were monitored with a flow meter.

Since the control valve should be designed not to respond to the 2HZ transient blood pump load, a laminar flow oil restriction was placed in series with the circuit to increase damping, and thereby attenuation. Figure A.3-42 shows the theoretically calculated relative poppet amplitude versus imposed frequency for various values of total flow restriction.

Figure A.3-40. Hysteresis Data on Control Valve Mod No. 1

Figure A.3-41. Control System Stability Test Apparatus

Figure A.3-42. Control System Transient Characteristics

This restriction includes that due to the oil filter resistance, in-line tubing resistance, and the restriction resistance (merely a squeezed down 1/2 in. length of 1/16 in. stainless tubing epoxied inside the oil line fitting. The valve mechanism acts as a second order system, but with the highly overdamped condition needed to slow the response time; it degenerates to a purely first order one with an exponential step response. The time constant, τ , existing for various total line resistances is shown in the plots. Since 95 percent output exists after 3 time constants' duration, the system specification of 10 second response time will be met for $\tau = 3.3$ seconds or less. The restriction used in valve Mod No. 1 gives an 8-second response time (dotted line in the figure) and as is seen, the blood pump load is attenuated approximately 97 percent at 2 Hz. The experimental setup confirmed the plots in Figure A.3-42 at both low and high rates of flow.

The oil line restrictor also makes the valve insensitive to the buffer space pressure variation of ± 2 psi which occurs at the engine speed of 10 Hz. As far as the valve bellows is concerned, its differential pressure is that which exists between the oil and buffer space pressures only. Therefore, the same frequency response plots are valid. This was verified experimentally. A buffer line restrictor is not needed to damp out such a large pressure fluctuation. The 10 Hz buffer space frequency attenuation shown in Figure A.3-42 is approximately 100 to 1; this was also confirmed by the testing.

The Mod 1 control valve was tested in closed loop on IVBM Engine No. 2 using the in vitro electrically heated hot side. Basic control characteristics were as follows: the speed control system was stable without the use of a flow restrictor; the system gain was, however, low. There was no noticeable hysteresis in valve performance. This is partly attributed to the pulsating working space pressure which loads and unloads the lead screw. Also, valve leakage in the just-closed position showed no reduction in peak engine power. The transient response to the control system was good. A step input change in torque demand from 12 in-oz (maximum load) to 2 in-oz (minimum load) resulted in only two damped oscillations. The response time was approximately three seconds, which was longer than anticipated.

The following tests were conducted without the presence of an oil flow restrictor. Figure A.3-43 shows representative engine power* data obtained on the Stirling engine, particularly indicating the speed control band. The valve set point was arbitrarily chosen to be fully closed at 620 rpm. Three watts of power dissipation occurred in a 90 rpm band. Due to the large speed increase to 710 rpm, it was calculated that the increase in bearing and other losses was about 1.5 watts. The control valve was therefore only dissipating 1.5 watts, not 3. These test results indicated a substantially lower than anticipated overall system gain. Hypothesized reasons at the time of testing for the reduced gain were the following:

- A low estimate in required valve flow rate in cc's per minute per watt of power dissipation.
- Possible engine pressure drop.
- Large pressure drops in fittings and tubing on engine and control valve.
- A lower than designed buffer charge pressure (160 psig instead of 175 psig).
- Lubrication system had only a 0.1 psi/rpm gain versus a design 0.2 psi/rpm gain.

To investigate these possibilities further, the setup shown in Figure A.3-44 was made. The engine buffer space port is capped and flow enters the engine from a tank through its charge port. The engine piston was removed to allow flow to reach the working space fittings. Figure A.3-45 shows the results of measuring pressure drop across the engine and valve as a function of valve poppet opening. A differential pressure across the system of 25 psi, (which is the equivalent rms pressure across the piston with the engine running) was used. The dotted curve shown is the actual data obtained. The apparent poppet position on the abscissa was calculated using the known pressure input to the bellows. However, due to known pressure drops across the 0.047 in. ID x 3 in. long buffer space tubing on the valve, the valve body is at a slightly higher pressure than that indicated on the buffer space pressure gage. This tends to close the poppet. By subtracting this closure, the true curve (solid) is obtained.

*Engine not fully warmed up.

Figure A.3-43. Control System Engine Power Versus Speed

Figure A.3-44. Pressure Drop Test Schematic

Analysis of this data shows that at a poppet opening of 3.2 mils one obtains a flow rate of 15,000 cc per minute STP through the system. This size valve opening on Figure A.3-43 intersects the maximum speed band point of 710 rpm. This flow rate theoretically dissipates about 1.5 watts. Additional bearing losses at 710 rpm are estimated also to be 1.5 watts. The total dissipation of 3 watts precisely matches the speed band data. Consequently, it can be estimated, with a good degree of certainty, that a flow rate of 30,000 cc per minute STP is required to dissipate 3 watts in the engine. Previous estimates were low at approximately 10,000 cc/min.

A flow rate test of the engine along with fittings in place but without the control valve mounted, was conducted to determine and isolate its flow resistance. Results are plotted as Curve B on Figure A.3-46. Preliminary laminar flow calculations indicated a small pressure drop even with 0.047 in. I.D. tubing employed. However, the contrary was true. Curve B which represents an initial engine configuration shows a very large pressure drop across the engine fittings at high rates of flow. This fact helped explain the lack of control valve flow increase after disassembling it and reworking the poppet (making it significantly shorter); theory had indicated a four-fold increase in gain. Curve A shows the effect of increasing the diameter of the working space tubing. This will be discussed shortly.

Figure A.3-45. Engine with Valve Pressure Drop

Figure A.3-46. Engine Pressure Drop Data

A relation was developed for the nearly turbulent flow conditions in the valve to accurately predict its flow rate characteristics. The following equation also predicts the apparent lack of flow rate increase with a new shortened poppet. It also correlates well with tubing flow rate data.

$$Q = C_D \pi D x \sin\left(\frac{\phi}{2}\right) \sqrt{\frac{2gP_0}{\rho_0} \cdot \frac{P_1 - P_2}{P_1 + P_2}}$$

where:

Q = Flow rate in valve (in^3/sec)

C_D = Discharge coefficient (typically 0.6)

D = Poppet diameter (inches)

x = Poppet motion (inches)

ϕ = Poppet included angle (degrees)

P_0 = Atmospheric pressure (14.7 psia)

ρ_0 = Atmospheric gas density (lb/in^3)

g = Gravity ($384 \text{ in}/\text{sec}^2$)

P_1 = Upstream pressure (psia)

P_2 = Downstream pressure (psia)

$P_1 - P_2 = \Delta P$

Valid for $\frac{\Delta P}{P} \ll 1$

According to the above formula, valve flow rate should vary linearly with poppet position at high flow rates. This was found to be true within experimental error as indicated by the solid curve's initial slope in Figure A.3-45. Furthermore, tests were conducted to verify the square root dependence of the pressure function to acceptable accuracy. This is demonstrated by the 0.50 slopes typically shown by Figure A.3-46 A and B data. In the above model, the cross-sectional area available to the flow is:

$$A = \pi D x \sin\left(\frac{\phi}{2}\right)$$

and, as one would expect, the formula is also applicable to predicting tubing flow rates and pressure drop when cross-sectional area is the pertinent parameter. In fact, one might generalize that predominant pressure drops are attributable to the smallest cross-sections in the flow path and are relatively independent of

tubing length. Tubing pressure drop tests were conducted which correlated well with the above formula. With a high degree of certainty, it was felt that the 0.038 in. ID working space tubing fitting, in particular, was limiting the engine flow—not to mention the effect of the standard 0.047 in. ID tubing also in line. This was verified by the Figure A.3-46 Curve B data.

In the other test, the engine working space port was capped and a special 1/16 in. ID fitting using 1/16 in. tubing was installed into the pressure transducer port of the working space. Curve A of Figure A.3-46 is that data. The significant increase in flow to 30,000 cc/min with a low pressure drop is apparent. This characteristic without constriction is required for proper control system performance. Similar data was taken on other size and length tubing, with and without right angle bends. The data conclusively showed that the 0.038 in. ID x 1/4 in. long tubing section initially on the engine working space fitting limited the flow. A revision to 0.070 in. ID tubing, 6 in. long, gave only a 1.0 psi pressure drop at a flow of 30,000 cc/min STP through the engine.

The foregoing tubing flow rate tests indicated that the initial control valve buffer and working space tubing and fittings of 0.047 in ID and 6 in. total length would be unacceptable above 15,000 cc/min flow. This was verified by testing the control valve at high flow rates using the Figure A.3-44 setup. The results are shown in Figure A.3-47. Similar to Figure A.3-45, the dotted curve is the actual data obtained which is based on the apparent poppet position. The solid curve is the true data. Again, flow is severely limited, even at a full 25 psi differential. Consequently, the valve poppet must open substantially more (to provide 30,000 cc/min flow) than which it was capable for a 30 rpm speed band. Therefore, 0.070 in. ID tubing must be used to minimize the buffer space pressure drop. It will be added that the valve body, when installed on the engine, does not see a dc pressure shift due to any buffer space tubing pressure drop. This is due to the 10 Hz cyclic pressure variation across the poppet. The control bellows cannot respond at this rate when a flow restrictor is in the oil line. Consequently, the valve body average pressure is the same as the buffer space pressure and proper operation results.

The data presented herein as well as data of Q versus ΔP for 0.070 in. ID tubing and associated fittings correlate excellently with the turbulent flow model.

Figure A.3-47. Pressure Drop Data - Valve With Fittings

Mod 2

The Mod 2 valve was designed to provide a much smaller speed band even at the reduced oil and engine charge pressures. To increase flow rate gain, the lead screw pitch was coarsened from 16 to 12 threads per inch and the poppet diameter was increased from 0.156 to 0.312 in.

A stiffer set point spring was used to provide increased bias force. The originally expected oil pump gain of 0.2 psi per rpm was assumed reduced to about 0.1 psi/rpm at 70 psi output pressure. This effect, and the stiffer spring in Mod 2, causes a reduction in gain and increase in the proportional speed band. To compensate, the poppet angle was increased from 60° to 80° to restore the design gain.

Figure A.3-49 shows the theoretical performance of the Mod 2 valve compared to Mod 1 based on the flow rate equation presented earlier. To obtain 3 watts of dissipation, a flow rate of up to 30,000 cc/min (STP) must be accommodated for a speed band of 60 rpm. Figure A.3-48 shows the experimental speed band data obtained on engine No. 2. The performance of the Mod 2 and Mod 1 control valves is shown. The band obtained for the Mod 2 valve was 50 rpm, in excellent agreement with what was expected. The set point speeds shown in this comparison are different, but their choice was arbitrary. As noted, the slope of the curve for the second valve diminishes somewhat near the closed position. This is attributed to 1.5 mils of poppet misalignment inherent in the assembly. An attempt to correct the misalignment in this particular valve was not quite successful, and some residual misalignment, higher friction forces, valve "sticking," and increased speed band are the result.

To obtain characteristic engine data for control system performance analysis, the power speed curve of in vitro electrically heated engine No. 1 was obtained at constant temperature. As noted in Figure A.3-50, the power out continues to increase with speed. The data were taken while the unit was being run-in with a new piston ring. Thus, the control valve must not only dissipate its nominal power rating, but must also dissipate the extra transient engine power available from thermal storage. The data obtained was over a short period of time when engine temperature remained constant. If left running beyond 620 rpm, the steady state heater

Figure A.3-48. Theoretical Flow Rate Data on Control Valve Characteristics

Figures A.3-49. Experimental Data on Control Valves #1 and #2

temperature decreased as well as the output power to nearly correspond with that obtained at 620 rpm. This "extra" power dissipation need of about 0.6 watts in a 90 rpm band partially explains the inadequacy of the Mod 1 valve design. This effect was less in the Mod 2 valve which exhibits a narrower speed band.

Figure A.3-50. Typical Change in Engine Output Power Versus Speed at Constant Temperature

Testing of the Mod 2 valve on in vitro electrically heated engine No.2 provided data on amplitude and phase shift of the pressure wave with the valve open. Figure A.3-51 is a plot of engine power versus phase shift relative to the closed valve position. The data was obtained from oscilloscope pictures and was averaged to minimize spread.

Figure A.3-51. Phase Shift Data

The phase shift results obtained verify the large flow rates existing in the valve, based on computed values. The consequences of designing this capability into the valve with regard to working space tubing size and poppet diameter were discussed previously.

The mean value of the pressure wave was also found to shift down, toward the buffer space pressure as the valve opens. This is probably due to unequal flow resistance past the valve poppet in the two directions. This "pumping down" effect is desirable over "pumping up" since bearing loads are reduced. However, the reduction in bearing losses also tends to increase the speed control band. This is of little concern since the speed control band of 50 rpm seems to be entirely satisfactory from a systems standpoint.

Running the engine with the nuclear heat source indicated that an increase in hot finger temperature results when the valve is controlling. Full engine power is therefore always instantly available in the 50 rpm speed band.

Mod 3

The Mod 3 valves incorporated several improvements over the Mod 2 valves:

1. A double-lead "V-thread" screw and anti-backlash nut replaced the square thread design. Radial play is to a desired 0.0002 TIR while axial backlash is set to 50 millionths of an inch. No lapping compound was used since it could degrade long-term performance.
2. An additional adjustment screw, which is sealed by an O-ring, has been added internally to the oil bellows. It serves two functions. First, when screwed all the way in (3 turns), the valve is forced open and the engine stalls. At the same time, the screw acts as a stop thereby not allowing the valve to be forced tightly closed during engine startup. This avoids possible valve overloading. By turning the screw fully outward while starting the engine, the valve is permitted to control the engine.
3. The valve seat is integrally machined with its body and lapped, using a dummy poppet which is accurately cut on the same setup as the valve poppet. This has resulted in excellent alignment and near "zero" leakage in the OFF position.
4. An increase in the stiffness of the set point spring was incorporated to allow a higher charge pressure to be used with the same oil pressure for additional flexibility in selecting system operating conditions. The speed band is expected to increase about 10 rpm due to this change, but is satisfactory for blood pump matching.

The Mod 3 valve configuration is illustrated in Volume I, Section 3.2.2.

Mod 3 valve was thoroughly tested on the in vitro electrically heated engine No. 1. Figure A.3-52 shows the engine response to positive (Figure A.3-52a) and negative (Figure A.3-52b) 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 A.3-52b, 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 A.3-52b 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 A.3-53a and A.3-53b are the results of larger negative step inputs at reduced engine power. In Figure A.3-53a, the step size was increased 1 in-oz from Figure A.3-53b, resulting in 1-1/2 oscillations. Theoretically, dropping the torque to zero load should not produce any noticeable increase in oscillation. Yet, Figure A.3-53b 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 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. Slightly larger diameter tubing should be used on any new valve. Dead volume may be controlled by using a shorter length.

Figure A.3-52. 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.

2 WATTS OUTPUT

Figure A.3-53. Control System Response at Low Engine Speed

Figure A.3-54 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 A.3-54 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 an improved gear pump. Pump pressure was typically 75 psi instead of 60 psi at 600 rpm using Tellus 25 oil.

Phase shift data was obtained using an engine with Mod 3 control valve installed. The data spread was reduced significantly compared to the averaged curve data presented in Figure A.3-51 for the Mod 2 valve. The Mod 3 sinusoidal type curve obtained agrees with theory and is shown in Figure A.3-55. The maximum phase lead of 17.5° agrees well with an analysis worked out for turbulent flow valve characteristics.

One of the Mod 3 control valves failed in service. It was found that one convolution of the electrodeposited bellows was cracked near its OD over 20 percent of the circumference. Metallurgical examination revealed a defect along the inner surface of the defective convolution, in the same relative location as the crack. Figure A.3-56 is a photograph of this defect at 1000X magnification. The defect, according to the manufacturer (Servometer Corp.) was probably due to the presence of dirt on the mandrel. This was a standard, stock bellows and an intermediate copper layer was not employed. As a result, the grain structure constituting most of the wall thickness was coarse. This structure may be detrimental to the fatigue strength of the material.

SHAFT POWER VS. SPEED

Figure A.3-54. Combined Control Valve and Engine Hysteresis

Figure A.3-55. IVBM Engine Short-Circuit Phase Shift Data

The control valve design is such that the bellows can adequately support the pressures encountered in-service and internal stops had been provided to limit bellows deflection. Therefore, it has been concluded that the stress concentration arising from the sharp radius and non-uniform wall at the defect caused the failure.

The following fabrication changes were insituted by Servometer for subsequent bellows:

- Specially machined materials with 100 percent inspection.
- Copper intermediate layer to promote a uniform fine-grain structure.
- Sulfur-free electrodeposited nickel.
- Strict control of bellows spring rate to help insure proper wall thickness.

Figure A.3-56. IYBM Control Valve Bellows Defect 1000X

Future Engine Control Schemes

Three possible control schemes were examined briefly for possible use on future engines. The first two of these involve what may be considered as external brakes or dissipation of engine power output and the third an improvement in the "short-circuit" control scheme.

Scheme 1:

The control system schematically depicted in Figure A.3-57 makes use of low losses (oil flow is supplied by a pump) in a valve to control the output of the engine. By increasing the system losses, the control will reduce output power and engine speed. When the engine is operating at design speed, the output pressure of pump I (4) is not sufficiently high to compress spring (2). If, however, a speed increase occurs, the output pressure from pump I will increase, pushing piston from position I to position II. In position I, output from pump II (5) is able

to flow freely (see arrow) to the sump (6), but when a speed increase moves the piston, the flow from pump II becomes more restricted in the small clearance space (1) of the piston. By restricting the flow, losses due to viscous friction go up and the result is both pumps slow down and with them the engine speed falls to its design value.

Figure A.3-57. Control Scheme No.1 - Flow Losses

Scheme 2:

The control scheme shown in Figure A.3-58 uses the flywheel motion to dissipate excess energy. When the shaft speed increases, it activates control (5) which opens valve I (6). Oil from the pump (7) flows into groove (2) and between the

A.3-78

flywheel (4) and the housing wall, producing viscous losses. As viscous losses increase, the engine speed drops. When the speed has dropped to its control point, the control shuts valve (6), diverting the excess flow back into the bearing lines.

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

Scheme 3:

Ambient temperature changes affect oil viscosity and, therefore, oil pump output pressure. Via the control valve, these changes would cause a speed change of the engine even at constant load. Test data was obtained on in vitro electrically heated engine No.2 with a Mod 3 "short-circuit" control during a 5-day continuous run period. In summary, a 2.7°F temperature increase resulted in a 15 rpm change in engine speed. An analysis of the speed control system was made and the following equation resulted, which accurately predicts the above result.

$$\Delta N = \frac{\epsilon P}{\frac{1}{K} + C} \Delta t$$

where:

Δt = temperature change, °F

ΔN = speed change, rpm

P = 70; oil pressure, psi

K = 10; valve gain, rpm/psi

C = 0.079; oil pump gain, psi/rpm

ϵ = 0.0143; oil viscosity sensitivity (percent/100)/°F

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

In performing the same type of analysis for the Mark I Prototype system design, it became apparent that the speed control valve could be designed to completely compensate for temperature induced changes in oil pressure since the buffer space pressure changes as well. In the present Mod 3 design, both effects adversely affect the speed band. A schematic of the new concept is shown in Figure A.3-59.

Figure A.3-59. Scheme No.3 - Temperature Compensated Speed Control Valve

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

complete flexibility of engine charge pressure and oil pressure, while eliminating set point drift due to ambient temperature changes.

Every effort has been made to hold down the size of the control valve of this concept. To this end, the set point spring was folded back through the use of a seesaw pivot which should be nearly friction free. This is because it transmits axial motion to the oil bellows and spring while undergoing pure rolling at its point of contact with the support flat. This pivot area, as well as the spring housing, is completely filled with oil. The overall control valve length is 3.75 in., including manifold, and fits within the axial length of the hot finger. A short speed adjusting screw is provided with access from the hot finger end.

It is presently felt that the potential improvement in overall system performance for other than just "fever conditions" would justify the use of this longer valve. The valve is roughly 1 in. longer than the non-temperature compensated Mod 3 valve. If the valve location is chosen parallel to the hot finger, the added length still fits within the overall Mark I engine envelope. Simplification of implantation procedures is an example of a potential benefit, since engine speed could be set without having to first reach a long duration thermal equilibrium in the calf.

3.1.4 Magnetic Seals Investigation

In an effort to help realize the Mark I Prototype System goals, component research and development efforts were undertaken on a magnetic coupling to transmit the output shaft power and magnetic fluid piston, displacer, and flywheel shaft seals.

Preliminary calculations showed very low friction loss with negligible blow-by loss if magnetic fluid piston and displacer seals are used in the Prototype engine, resulting in excess of 2 watts mechanical power savings. Magnetic fluid piston and displacer seals would eliminate the need for roll socks and their associated supporting oil pressures, permitting a fully pressurized crankcase to be employed. This would reduce bearing loads and further improve engine efficiency.

Magnetic fluid shaft seals to keep blood pump and engine oil out of the magnetic coupling housing are also presently contemplated. These shaft seals consume approximately one-fifth the power of present screw-type seals, with a power saving of not less than 0.1 watt per seal.

The use of a magnetic output coupling allows the Mark I Prototype Engine to be hermetically sealed from the external environment. This will enhance life and reliability.

Magnetic Coupling

The large transient torque requirements of the blood pump compared to its much lower average torque level imposes special requirements on a coupling system on the engine output shaft. The following major points must be considered:

- The magnetic coupling must not lose synchronism under any conditions.
- Use of a slip clutch as a transient torque-overload protector.
- Use of a slip clutch as a steady overload-torque protector.
- Startup capability of system after clutch slippage.

Various types of slip clutches, listed in Column 1 of Figure A.3-60, were considered. Figure A.3-61 demonstrates how the blood pump torque and engine torque requirements are combined in each design approach. It was decided that a slip clutch having a plain detent characteristic (Type B) would be the best choice since it adequately protects the magnetic coupling from all expected transient overloads.

Figure A.3-62 shows a preliminary layout of a combined magnetic coupling/slip clutch. The engine shaft is connected to the flywheel magnets through a yoke. The magnetic coupling transmits the torque through the membrane to the inner armature which is the outer raceway for the detent clutch. The detent housings are supported by a ball bearing and connected to the output shaft. Since the detent torque setting is less than the magnetic coupling torque capacity, upon its slippage the output shaft will rotate relative to the input shaft. A low drag torque will then be transmitted by the detent friction and spring tension. Under normal operation, no clutch slippage or ball bearing rotation will occur.

CLUTCH TYPE	MAGNETIC COUPLING & ENGINE STALL CHARACTERISTICS			
	FAST PULSE	MODERATE PULSE	STEADY OVERLOAD	COMPLEXITY
A				
CONSTANT TORQUE CLUTCH - NO DETENTS	NO SLIP NO STALL	NO SLIP STALLS	NO SLIP STALLS	SIMPLE
B				
DETENTS WITH LOWER SLIP TORQUE	NO SLIP NO STALL	NO SLIP NO STALL	NO SLIP STALLS	COMPLEX BUT STRAIGHT FORWARD DESIGN
C				
DETENTS WITH SLIP & DAMPING	DANGER OF SLIP NO STALL	NO SLIP NO STALL	NO SLIP NO STALL	COMPLEX AND REQUIRES DEVELOPMENT/ TESTING
D				
NO CLUTCH	NO SLIP NO STALL	SLIPS NO STALL	NO SLIP STALL	SIMPLEST

Figure A.3-60. Slip Clutch Designs

SLIP CLUTCH TRANSIENT BEHAVIOR INPUT-OUTPUT CHARACTERISTICS AT 2400 R.P.M.

Figure A.3-61. Magnetic Coupling/Slip Clutch Characteristics

Figure A.3-62. Baseline Magnetic Coupling

An analytical and experimental program was carried out to develop a magnetic coupling portion for the engine output link. Its purpose was to provide the magnetic-coupling-flywheel combination for the Mark I Prototype Engine. The required coupling torque specifications were 1.5 in-lb at 2,400 rpm. The program involved an initial design (Mod 0) and two major modifications (Mods 1 and 2) plus many minor changes to the Mods.

It was recognized at the onset that a great deal of experimental modification was to be expected to achieve an adequate coupling. Therefore the coupling test rig was designed to provide maximum versatility; not only were the magnets removable but the air gap was adjustable. The purposes of the rig were to evaluate:

- Static torque capability verification
- Torque versus angular rotation from which is obtained:
 - Spring constant
 - Energy absorption capability
- Torque versus air gap size
- Membrane eddycurrent loss determination

Mod 0 Magnetic Coupling

Figure A.3-63 is an end view of the Mod 0 coupling showing the 12-pole armature design on the inside surrounded by the 6-segment flywheel rotor portion which holds 12 samarium cobalt magnets. The blood pump flexible shaft would be attached to the output armature. A 7 mil thick titanium membrane, seen in Figure A.3-64, is located in the 20 mil air gap between armature and rotor. It hermetically seals the engine and will sustain up to 200 psi crankcase charge pressure. The transmission of torque and power across the air gap is subject to eddycurrent losses existing in the metallic membrane and windage losses on the surface of the rotating parts.

Figure A.3-65 shows the test fixture used to obtain coupling torque versus relative angular rotation. The same fixture was used to rotate the coupling at its design speed of 2400 rpm (and other speeds) for eddycurrent data. To do this, the drive motor (not shown) was calibrated and attached to the coupling shaft with flexible rubber tubing. After taking windage and other loss data, the

Figure A.3-63. Mod 0 Experimental Magnetic Coupling

Figure A.3-64. Titanium Membrane

<u>Item</u>	<u>Description</u>
1	Protractor - $\Delta\theta$ measurement
2	Weights - Torque input
3	Ball Bearings - Shaft supports
4	Torque Arm - Locked rotor

Figure A.3-65. Magnetic Coupling Test Fixture

membrane was inserted and the increase in power consumption recorded. Using the motor calibration curve, eddycurrent loss was determined. Basic test results are:

- Performance Data
 - 6.6 in-lb maximum torque
 - 400 mW eddycurrent loss at 2400 rpm
 - 140 in-lb/radian initial spring rate
 - 0.66 in-lb energy absorption capability
- Size and Weight
 - 20 mil air gap
 - 50 grams of magnets
 - 7 mil titanium membrane x 1.91 in. diameter
 - Pole faces 0.060 in. wide x 0.500 in. axial length

Figure A.3-66 shows plots of coupling torque data as a function of relative angular rotation for different pole-face widths. As the pole-face width was decreased from 0.100 in., peak torque increased until the maximum value of 6.6 in-lb was reached. For still smaller widths, torque decreased. At this optimum point, the magnet is operating near its maximum energy product and magnetic leakage is not yet prohibitive. Accurate analytical predictions are difficult to make because magnetic leakage changes with relative rotation between armature and rotor. The magnet "voltage" output is highly non-linear. Deviation from a sinusoidal torque curve becomes apparent at small pole face widths because of the increased effect of fringing. This is of little concern for the application because high maximum torque and energy absorption are the primary objectives. Figure A.3-67 shows this information. The "torque curve" versus pole face width was obtained by plotting only the peak torque in Figure A.3-66 for each pole face width. By integrating the area under each of the curves, the coupling stored energy (synchronous absorption energy capability) was found. This is presented in the "energy curve." It is the optimal pole face width for the experimental coupling and verifies that an optimal value for both torque and energy does exist for a given size air gap and axial length. Instead of reducing pole face width, its length could have been reduced with the same general result.

Figure A.3-66. Coupling Torque Data Versus Angular Displacement

Figure A.3-67. Peak Torque Data and Energy Storage Versus Pole Face Width

As the air gap size was reduced (by the addition of steel shims) with a constant pole face width, the peak torque of the coupling increased, as expected. This is shown in Figure A.3-68 for a 0.100 in. wide pole face. If the air gap were reduced to a very small value (several mils), the torque would increase further, probably peak out, and then reduce as the magnet energy and leakage factor pass the maximum energy point. Since this occurs at impracticably small air gaps for use with a membrane, the air gap was chosen to be as small as practical and the magnetic circuit pole face geometry optimized as previously described.

Mod 1 Magnetic Coupling

Extrapolating from the Mod 0 test data, it was thought that a redesigned Mod 1 coupling, Figure A.3-69, could be made to meet the required 1.5 in-lb torque at 2400 rpm while exhibiting only a 20 to 50 mW power loss. The design characteristics of the Mod 1 coupling are shown on Table A.3-3.

Figure A.3-68. Peak Torque Data Versus Radial Air Gap

The low magnet weight in comparison to the tungsten flywheel ring is made possible by an efficient magnetic circuit design. A speed of 2400 rpm compared to 1800 in the IVBM engine, significantly reduces the flywheel weight requirement (by 44 percent), while the eddycurrent loss is very low due to the pole face geometry and the thin, high resistivity (170×10^{-6} ohm-cm), titanium alloy membrane. The high torsional stiffness resulting from the narrow 0.08 in. pole face width gives a high natural frequency whereby blood pump load oscillations should be rapidly damped out by the flexible shaft.

Figure A.3-69. Mod 1 Experimental Magnetic Coupling Design

TABLE A.3-3

MOD 1 MAGNETIC COUPLING CHARACTERISTICS

Torque	1.5 in-lb at $\Delta\theta = 6^\circ$
Eddycurrent Loss	20 to 50 mW
External Surface Windage Loss	less than 35 mW
Speed	2400 rpm
Tungsten Flywheel Ring	150 gm (2.9 in. OD x 7/16 in. long)
Magnet Weight	30 gm
Membrane	0.005 in. titanium alloy (6Al - 4V) x 1-1/2 in. dia.)
Pole Faces	0.08 in. wide x 0.15 in. axial length
Natural Frequency	200 Hz with 10^{-5} (in-lb-sec ²) armature reflected inertia
Energy Absorption	0.2 in-lb
2 Hz Pulsating Blood Pump Load	less than 5 mW damping loss; 2° tran-
Transient Effects	sient angular rotation

A picture of the experimental Mod 1 coupling is shown in Figure A.3-70. The initial configuration employed four poles with on 4.5 gm magnet per pole. The peak torque held was 0.75 in-lb. The actual pole face dimensions were 0.080 in. wide by 0.160 in. long while the air gap used was 0.020 in.

Neither an increase nor decrease in pole face width affected the peak torque obtained, indicating that the Armco iron pole faces were magnetically saturated. A combination gauss meter and flux meter instrument was used to take magnetic field measurements. The air gap field was measured to be 21,000 gauss confirming that the iron was saturated and that no greater torque could be obtained in this configuration.

Rather than increase the torque of the coupling by significantly lengthening the pole faces (which would necessitate a thicker membrane and result in higher eddy-current losses), the number of poles was doubled and their length was increased by 0.020 in. The peak torque increased from 0.75 to 0.90 in-lb, and the iron was no longer saturated. However, the design goal of 1.5 in-lb was not reached.

Therefore, larger cross-sectional area magnets were procured and the coupling was modified to accept them. Four magnets were still employed, however, each weighs 7.7 gm and is associated with two poles which are located on the same magnet pole piece. A 10-turn coil of 3 mil wire was used for measuring the pole face flux density.

Figure A.3-70. Mod 1 Magnetic Coupling

The peak torque obtained using these larger magnets was 1.35 in-lb. Pole face width and length were 0.065 and 0.180 in., respectively. The radial air gap was 0.020 in. The measured pole face flux created was 17,000 gauss, nearly saturating the iron. Figure A.3-71 presents the data obtained showing torque as a function of relative angular rotation θ . In an effort to optimize the design, the pole face width was first increased and then decreased by 0.010 in. The peak torque reduced 3 percent in both instances indicating that a 0.065 in. wide pole face was the optimum value. Figure A.3-71 also shows the data obtained using the original smaller magnets with an 8-pole configuration for comparison.

Figure A.3-71. Torque Versus θ With Large and Small Magnets

Using the larger magnets, the coupling air gap was then reduced to increase the peak torque 11 percent from 1.35 to 1.50 in-lb. Iron shims were glued to the armature pole face tips to decrease the size of the air gap. Figure A.3-72 is the parametric data of torque versus θ for various size air gaps. The pole face width was 0.55 in., not the optimum value for a 20 mil air gap. Replotting the peak torque obtained as a function of air gap size, as shown in Figure A.3-73, indicates that the required 11 percent torque increase (above the 0.020 in. gap value) will result from using an air gap of 0.017 in. This clearance is satisfactory for the 6 mil thick membrane. Figure A.3-73 also shows the coupling energy storage capability without slippage. It is the computed area under each curve of Figure A.3-72 with θ in radians.

A revision to the magnetic circuit was made to reduce eddycurrent losses. In the just previous Mod 1 configuration, the magnet 0.50 in. length fitted axially into the 0.50 in. long tungsten flywheel ring. The close proximity of the magnet and its pole piece to the membrane flange was found to induce substantial eddycurrents in the flange. This condition was corrected by providing a large clearance (0.065 in.) between the magnet face and the membrane flange. This is shown in Figure A.3-74, which is a sectional view of the design change. This clearance was obtained by rotating the magnet 90°. Its 0.500 in. length now lies circumferentially and its 0.375 in. axial dimension fits inside the tungsten ring, thereby providing the above clearance. New pole faces have been made that fit this magnet orientation. Also, tests indicated that a 4 percent torque increase could be gained by using larger cross-section iron.

A 5 mil thick titanium alloy membrane 0.210 in. long was tested for eddycurrent loss with the 4-pole magnetic coupling configuration. Using the small magnets, the peak torque of this configuration again was 0.75 in-lb. The eddycurrent loss of 2400 rpm was 400 mW (much greater than expected). It was found that axial magnetic field leakage into the aluminum bearing support and membrane flange, due to the close proximity of the magnets and pole face structure, accounted for 295 mW of this total. An additional 45 mW was due to self-shortening of eddycurrents due to the presence of the end face of the membrane. The thickness of the right end face was reduced from 0.015 to 0.005 in. without any reduction in eddycurrent loss. However, removal of the end face and its replacement with a Plexiglas disc resulted in the elimination of the 45 mW self-shortening loss.

Figure A.3-72. Torque Versus θ and Radial Air Gaps

Figure A.3-73. Peak Torque and Energy Versus Radial Air Gap

Figure A.3-74. Magnetic Coupling Configuration

A plexiglas membrane flange as well as a plexiglas end face was made to support the 0.005 in. thick by 0.190 in. long cylindrical center portion of the membrane. The eddycurrent loss generated was 57 mW. This is the true loss associated with an open ended cylindrical shell.

The applicable loss formula for equally spaced poles around an open cylindrical membrane is the following:

$$P = \frac{\pi}{4} 10^{-17} W \omega^2 D^3 \frac{\delta}{\rho} B^2 \sum_{n=1}^{\infty} \frac{\sin^2 \left(\frac{n\pi t}{L} \right)}{n^2 + \frac{1}{4} \left(\frac{L}{W} \right)^2} \text{ watts}$$

where:

D = membrane diameter (cm)

W = pole face axial length (cm)

ω = angular velocity (rad/sec)

B = maximum field in air gap (gauss)

t = width of pole face (cm)

L = spacing between pole faces (cm)

ρ = resistivity of membrane (ohm-cm)

δ = membrane thickness (cm)

This formula of course gives no information about eddycurrent losses generated in the membrane flange or the magnetic coupling housing. In order to minimize engine length and flywheel housing diameter, housing clearances must be kept to a minimum which adversely affects eddycurrent loss due to magnetic leakage.

With regard to the flywheel OD, magnetic field leakage was very low (100 gauss) using the small magnets, but was large (800 gauss) with the larger magnets. By increasing the return ring thickness behind the magnets, from 0.070 to 0.125 in., and decreasing their axial length from 0.500 to 0.375 in. (by the 90° change in magnet position), leakage flux was not able to permeate the return ring. Consequently, the radial flux level at the OD of the flywheel was reduced to an acceptable 200 gauss level. The housing may now be positioned radially within 0.015 in. of the flywheel.

Magnetic field leakage measurements using the smaller magnets were obtained as a function of axial and radial distance around the coupling. The results of these gauss meter measurements are shown in Figure A.3-75 where the field density in gauss is plotted against radial position on the abscissa for each of nine axial locations. The peak field is seen to be located on either the top of the magnets or near the iron pole faces. The total eddycurrent loss obtained at 2400 rpm was correlated with these measurements in order to estimate the losses associated with the housing and membrane flange design. Since the differential eddycurrent loss is expected to be proportional to the volume of metal and proportional to the square of the product of field and velocity, a suitable parameter was chosen for correlating the data. It is plotted as a function of radius in Figure A.3-76. The integrated area under the curve is proportional to the total eddycurrent loss expected on each side of the housing. The upper curve is the loss obtained using an axial flange-to-magnet clearance of 0.015-in. while the lower curve is for a clearance of 0.065 in.

The formula derived below was employed to estimate housing eddycurrent losses.

$$dP \propto \frac{t}{\rho} B^2 V^2 dA$$

$$dA = 2\pi R dR \text{ and } V = \omega R$$

$$P = K 2\pi \omega^2 \frac{t}{\rho} \int_{R_1}^{R_2} B^2 R^3 dR \text{ watts}$$

where:

P = eddycurrent loss (watts)

t = flange or housing thickness (inches)

ρ = resistivity (ohm-inches)

B = normal magnetic field (gauss)

R = radius (inches)

ω = angular velocity (rad/sec)

K = 8.84×10^{-15} (watts-sec² ohm/gauss²-in⁴)

K was empirically derived using the power loss of 0.090 watt measured in a 0.065 in. thick titanium flange with a 0.015 in. axial clearance.

Figure A.3-75. Magnetic-Field Leakage Measurements

Figure A.3-76. Eddycurrent Loss Parameter Study

By increasing this clearance from 0.015 in. to 0.065 in., the eddycurrent loss would reduce to 12 mW. This is a computed value based on the area ratios in Figure A.3-76 and is the result obtained when the small magnets are used with a flange thickness of 0.065 in. This loss will be different where the large magnets are employed.

The high resistivity titanium alloy membrane located in the magnet coupling air gap is a thin-walled pressure vessel. It must contain the crankcase charge pressure. To minimize eddycurrent losses, it must also be as thin as practical without buckling. A parametric study was made for determining permissible buckling loads. Figure A.3-77 shows the results of this analysis, and these results were used in evaluating design alternatives for the pole-face geometry. For example, a longer pole-face length results in a reduction in buckling pressure and a thicker membrane must be employed. The design pressure used in these studies was 200 psi.

In order to verify the results shown on Figure A.3-77, a test membrane 5 mils thick and 0.190 in. long was fabricated and instrumented for deflections. Figure A.3-78 is a photograph of the membrane after buckling. A buckling pressure of 280 psi for a 5 mil, 0.190 in. long membrane was predicted and experimentally verified in this test within a few psi. Membrane maximum radial deflection was 0.6 mils and occurred at 125 psi.

A similar parametric study was made for longer membranes. The results show that for the anticipated 5.5 mil thick by 0.255 in. long membrane, a buckling pressure of 245 psi will result. This is a safety factor of 1.75 for the anticipated Mark I prototype engine crankcase charge pressure of 140 psi.

The flywheel, shown in Figure A.3-74, has a length of 0.50 in. and a diameter of 2.90-in. The internal structure of the flywheel that incorporates the magnetic circuit must be filled with low density, rigid material to give it a smooth disc shape. This is necessary to reduce wind turbulence. An aluminum disc was machined to the above dimensions for simulating windage loss at 2400 rpm. A torsion spring torque transducer was used to measure the input torque with a sensitivity of 0.0082 in-oz per degree of rotation. Windage loss amounted to 50 mW at 2400 rpm.

Figure A.3-77. Buckling Pressure Versus Wall Thickness and Length for Titanium-Alloy Membrane

Figure A.3-78. Structural Test Titanium Membrane After Buckling

Mod 2 Magnetic Coupling

It can be shown that the following formula can be used for predicting the peak torque of a magnetic coupling of similar pole-face geometry and whose iron pole-faces are near magnetic saturation. The proportionality constant is obtained experimentally.

$$T = KNLB^2 \frac{R}{g}$$

where:

T = peak torque (in-lb)

B = air gap flux density at zero torque ($\theta = 0^\circ$) (gauss)

L = pole-face length (inches)

g = radial air gap (inches)

R = air gap radius (inches)

N = number of poles

The Mod 2 magnetic coupling is that developed for possible use in a Mark I Prototype system design. The basic configuration is the same as Mod 1. However, eddycurrent tests show 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 eddy-current 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 axially positioned within 0.015 in. 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 in. thick end plate from 0.065 in. to within 0.015 in. 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 in. thick plate. A plate of this thickness results in only 12 mW of loss at 0.015 in. diameter.

The left end mounting flange-to-magnet gap was kept at 0.065 in. At a clearance of 0.035 in., eddycurrent loss was 50 mW for a titanium alloy test flange of 0.060 in. thickness. But when spaced at 0.065 in., 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 in. radial clearance with the flywheel OD. The magnetic circuit return rings were increased in thickness, which has reduced the radial magnetic field leakage to the 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 in. 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 in. 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 in. 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.

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 magnetic flywheel shaft seal is now obvious; namely, it is hermetic; and any compatible, containable gas can be used to pressure the flywheel housing. For example, a windage loss of only 56 mW would result if helium were employed. But this gas cannot be contained for a 10-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.

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:

$$\Delta\omega = \frac{A}{\pi I f}$$

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.

Figure A.3-79 is the final torsional characteristics of the Mod 2 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 coupling, the air gap will be left slightly undersize to compensate for magnet strength variations, tolerances, etc. This will result in excess torque. 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.

Figure A.3-79. Magnetic Coupling Torque Characteristics

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 Figures A.3-80a. Figure A.3-80b shows the sinusoidal torque input to be unidirectional. This was obtained by adding a D C current equal to the half amplitude current of the sinusoidal hysteresis brake. The drive motor (engine) must overcome this 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 A.3-79 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 A.3-79. Resonance reactions with either the engine or blood pump are not anticipated.

FIG. (a)

FIG. (b)

Figure A.3-80. Magnetic Coupling Experimental Test Stand

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 that includes a thin wall is shown in Figure A.3-81. The top photograph shows the left end mounting flange on the armature side which is 2.9 in. 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 in. to add minimally to engine length. The armature rotates inside the circular ring that is seen at the ID of the ribs. The rib ID is also the location of the 0.005 in. thin wall cylinder that is best seen in the bottom photograph. The bottom photograph is a view looking toward the right end plate of the coupling. A second ribbed structure, composed of 10 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 in. at 150 psi agreed well with the calculated value.

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 that still leads to a very complex analysis. Therefore, analysis of the problem was attempted using strain energy techniques that have led to a simple formula that 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 .

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

where:

P = external pressure

D = mean diameter

E = modulus of elasticity

I = mean moment of inertia

LEFT FLANGE

PRESSURIZING
FIXTURE

.0055" MEMBRANE

RIGHT END
FACE

2.9"

Figure A.3-81. Ribbed-Prototype Structure: Magnetic Coupling Membrane

Having tested the ribbed structure, a sandwich design was evolved which should provide even better performance. It is shown in Figure A.3-82 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 A.3-82 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 following table shows, 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 A.3-82)	18 gm
Ribbed (Figure A.3-81)	36 gm
Flat Plate	57 gm

The mounting flange axial length of 0.11 in. for the sandwich structure is ideal for incorporation of a screw viscosity seal on the output armature shaft (used for excluding blood pump oil from the armature housing). Adding armature length and an anticipated seal length of 0.12 in. permits the combined length to be incorporated inside the mounting flange. The armature shaft need only be supported; i.e., by the gearbox bearings.

Magnetic Fluid Piston Seals Development

As an approach to replacing rubbing piston rings in the Stirling engine, a brief exploration of magnetic fluid piston seals was carried out. The results did not give enough promise that the approach would work for the Stirling engine application to pursue it further because of potential blow-by problems.

Test Piston

A test piston and seal were designed and fabricated to the concept shown in Figure A.3-83. This is a "non-magnetic" design which means that the cylinder wall will be made of a non-magnetic material. The benefit here is that no

Figure A.3-82. Sandwich-Membrane Structure

Figure A.3-83. Magnetic Fluid Piston Seal

radial magnetic thrust loads are developed on the piston due to eccentricity of its motion. A 2 mil radial clearance around the piston is used. The flux produced by the magnets passes out through an iron end pole-face, down the 2 mil copper to 13 mil iron sandwich and back into the magnet through the other pole face. The flux fringing produced around the outer periphery of each 2 mil copper section creates a sufficient flux gradient in the cylinder gap to seal an estimated 1 psi per stage. Fifty such stages then would be capable of sealing the engine fluctuation of ± 45 psi with safety factor.

The centers of the two samarium cobalt magnets are hollow because the final design must be capable of sealing the displacer rod as well. A bellows life test apparatus was refitted with a new guide bushing for the piston rod and a new plexiglas head was made for tests of the magnetic seal. Thus, provision was made to optically observe the piston/cylinder seal interface during operation to verify the spatial relationships between individual magnetic fluid rings.

Since the vapor pressure of the fluid in the magnetic seal must be low to minimize its migration, silicone oil is proposed for an actual engine. In the test seal, Apiezon C oil was used since ferrite loaded silicone oil was not readily available. Figure A.3-84 shows a photograph of the piston. The 50 circumferential copper laminations are clearly visible. Each one is expected to form an O-ring-shaped magnetic-fluid seal. A flange is located at the bottom for screwing the piston onto the test fixture piston rod.

The following preliminary tests were conducted to evaluate the magnetic piston. The first test involved coating the piston with 400 gauss ester-base magnetic fluid. Figure A.3-85 shows some of the 50 "O-rings" of fluid formed. Each O-ring is 1 to 1.3 mils in height and 5 mils wide. The amount of magnetic fluid brushed on only slightly affected the O-ring dimensions. More importantly, however, the magnetic field along the axial direction of the piston was found to be non-uniform, being significantly weaker in the central region. Smaller O-ring cross sections exist here while larger ones are located at the ends. This is due to flux leakage through the surrounding air.

Figure A.3-84. Magnetic Fluid Piston

Figure A.3-85. Magnetic Fluid "O-Rings" On Test Piston

The piston was first tested in a 2 mil radial clearance Plexiglas cylinder and was capable of sustaining a static pressure of 11 psi using 400 gauss ester-base magnetic fluid. The installed width of the O-rings was 2 mils. Using this dimension, the theoretically calculated pressure was 12 psi, indicating correctness of the theory. A new cylinder was constructed with a 0.8 mil piston clearance. The measured static pressure increased to 45 psi, also in good agreement with calculations. This is the differential pressure seen by the piston in the engine, indicating that this magnitude of pressure can indeed be sealed statically, using a fringing effect piston.

Dynamic piston tests were also conducted to determine the degree of sealing or pressure capability as a function of piston speed. This data is necessary for designing an actual piston.

When the piston is reciprocated at low speeds and low differential pressures without the magnetic fluid (clearance seal), a reduction in pressure wave amplitude can be detected for the 2 mil clearance cylinder. This is shown in Figure A.3-86, an oscilloscope photograph of the pressure waveform with and without magnetic fluid. At higher speeds, there was no detectable amplitude difference between the two. However, the apparatus drive motor would not exceed 450 rpm with the clearance seal undergoing ± 10 psi differential, whereas it could reach a speed of 750 rpm with the fluid present. This is attributable to a dissipation of power due to a phase shift of the pressure wave, which results in a greater load on the motor. This will occur if the piston leaks even to a small degree (and is the same effect on which the engine speed control system is based). This effect can be seen before any reduction in pressure amplitude can be detected and can be used to detect piston leakage due to its high sensitivity.

A modification to the cylinder head was made to accept a magnetic pickoff in order to monitor the piston position and record phase shift as a means of determining blow-by data. Since the pickoff output depends on rate of change of flux, it was not sufficiently stable to determine piston position accurately. A switch was added to the cylinder lead to close at the peak piston position. This provided a more accurate reference for measurement.

Figure A.3-86. Oscilloscope Traces of Pressure Wave Form Above Piston With and Without Magnetic Fluid

Piston data with the 0.8 mil cylinder clearance was basically the same as with the 2 mil cylinder clearance. There was no difference in pressure amplitude during operation whether fluid was used or not at the same charge pressure, thus amplitude measurement could not be used to determine piston leakage. The maximum attainable motor speed without fluid was 650 rpm while sustaining ± 20 psi differential but increased to ± 26 psi with the use of fluid. This particular value of 26 psi was the reliable static pressure which the piston could hold with the piston rod displaced in its guide bushing clearance. This indicates that blow-by had just started at 26 psi limiting the speed to 650 rpm. Unfortunately, the test fixture guide bushing did not hold the piston as concentric as desirable for tests of this nature.

Two additional magnets which increased the magnet strength by 50 percent were added. This modification was made to increase the magnetic field in the central portion of the piston and to effect a better seal than had previously existed. This created fluid O-rings that were of uniform cross-section along the piston length. This corrected an earlier problem wherein smaller size O-rings formed at the center of the piston. Static pressure testing of a seal having a 1 mil radial gap in a plexiglas cylinder demonstrated a 30 psi pressure difference capability. A hermeticity test was performed and showed no leakage after 500 hours, at which time the test was intentionally discontinued.

Unfortunately, under dynamic conditions, unlike the static case, gas would occasionally blow-by and magnetic fluid would be lost. An analysis was made to determine the vapor pressure requirements of magnetic fluid for use in a magnetic piston seal. Previous analysis based on maximum oil evaporation rates was too conservative indicating the necessity of an extremely low vapor pressure oil like silicone or Apiezon C. A new analysis, based on the maximum amount of mass of vapor under equilibrium vapor pressure conditions that can possibly be pumped to the hot side of the engine per stroke, showed that the allowable vapor pressure for a 10-year life was 100 times greater than that of silicone oil. This opens up new possibilities for potential fluids since stronger magnetic saturation values can be obtained while maintaining a low viscosity. The silicone-base magnetic fluid has a viscosity of 400 cp while exhibiting a low 200 gauss magnetic saturation, but has a low vapor pressure of 10^{-9} torr. A magnetic fluid exhibiting a more desirable magnetic saturation of 400 gauss with the same viscosity and 10^{-7} torr vapor pressure is desirable.

Linear Magnetic Fluid Seal Concept

It is known that a magnetic fluid will be attracted into an increasing magnetic field gradient. Thus, by placing an axial field gradient between a reciprocating shaft to be sealed and its housing, any magnetic fluid within this gap will be forced back by the gradient. A seal incorporating this effect is shown in Figure A.3-87. The tapered pole-face creates an axial magnetic field gradient being strongest near the magnet where the pole-face gap is the smallest. Magnetic fluid at the top of the shaft is sealed under static conditions in the same manner as in the rotary shaft seal. When the shaft is moved downward, the shaft motion out of the magnetic fluid reservoir produces a thin film of magnetic fluid on the shaft. But the magnetic field gradient force exceeds the downward gravity force, the shaft acceleration forces, and any viscous forces, thereby not permitting the magnetic fluid to move down the shaft. By making the tapered pole-face length greater than the shaft stroke, the magnetic fluid is trapped.

This concept was tested by modifying a rotary shaft seal fixture as shown in Figure A.3-88. The fixture ball bearings were used as linear bushings to guide a reciprocating 0.160 in. dia. rod. The rod was driven by a string attached to an eccentric. The fixture could operate up to a stroke of 0.093 in. at 1800 rpm. The seal used was the same geometry as a rotary shaft seal. Even though this

type of seal is designed to hold back pressure for a rotating shaft, the pole-face air gap does possess a small axial magnetic field gradient, but only over a short distance. By keeping the stroke small, less than 0.06 in., the seal was run several times for 24 hours against 5 psi. There was no magnetic fluid leakage past the seal.

Figure A.3-87. Linear Magnetic Fluid Seal Concept

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

Figure A.3-88. Linear Magnetic Fluid Seal Feasibility Test Fixture

Magnetic Fluid Miscibility

Miscibility tests between an ester-based magnetic fluid and various liquids were conducted to determine whether diffusion effects might weaken the pressure capability of a magnetic fluid seal. The test apparatus in Figure A.3-89 was set up to make the determinations. Two magnets and pole faces are used to focus a magnetic field through a glass capillary tube of approximately 3/32 in. ID. The magnetic fluid was added to the tube, and Tellus oil was placed above it using a hypodermic syringe. The magnetic fluid at first was level with the bottom edge of the pole-face. As can be seen in Figure A.3-89, after approximately 800 hours, the magnetic fluid has pushed beyond the pole region. The Tellus oil is slowly diffusing into the magnetic fluid, weakening its magnetic saturation. After 1200 hours, the seal broke down. This result was predictable in that other tests involving mechanical mixing showed that this combination of fluids would not separate. The test was repeated using ethylene glycol and ester-based magnetic fluid. After 1300 hours, there was no evidence of weakening. Mechanical mixing tests have shown that these fluids are immiscible. Specimens of ester-based magnetic fluid that had been mixed with silicone oil flocculated to the point that they became a jellied mass. These preliminary experiments tested the methodology for evaluating the magnetic fluid-lubricating oil systems.

Magnetic Fluid Rotating Shaft Seals

In the Mark I Prototype engine to control windage losses, liquids must be excluded from the magnetic coupling/flywheel region. A design decision was made to accomplish this, if possible, on the engine oil side by a magnetic fluid rotating shaft seal. The main advantage of such a seal is extremely low power consumptions when properly designed. The use of high coercive force samarium cobalt magnets permit the use of such large radial air gaps that the viscous power loss in the coupling seal was anticipated to be on the order of milliwatts at 2400 rpm.

Theoretical Analysis

A search of the literature pertaining to magnetic fluid devices revealed that equations of motion and design equations were often presented without any

Figure A.3-89. Magnetic Fluid Miscibility Tests

statement of how these equations were derived and under what conditions they are valid. A decision was made to attempt to examine rigorously the formulation of magnetic fluid mechanics in non-conducting media in an effort to determine the validity of the design equations presented in the literature.

The Chu formulation of electromagnetism in moving media, as suggested by Penfield and Haus, Electrodynamics of Moving Media, was chosen as the basic set of constitutive equations. The equation of motion was developed for an inviscid magnetic fluid with zero elastic field and polarization field, and where the magnetic dipole moment strength is independent of the number of dipoles. In addition, the boundary conditions at the interface between two media were obtained with the inclusion of magnetic and surface tension forces. Bernoulli-type equations were found by integrating the momentum equations in the case of steady, irrotational flow.

The pressure difference across an idealized, static magnetic fluid seal was obtained for a geometrically simple case by neglecting surface tension and gravitational forces. The results agreed with published results of Rosensweig, "Ferrohydrodynamics," Encyclopedic Dictionary of Physics, pp. 111-117 (1970).

$$\Delta P = \mu_0 M_{\text{gap}} H_{\text{gap}}$$

where:

ΔP = pressure difference

μ_0 = magnetic permeability in free space

M_{gap} = magnitude of the magnetization

H_{gap} = magnetic field strength

In magnetic fluid devices, there is always relative fluid motion, usually in a narrow gap, so that viscous effects may be of importance. Since the theory outlined briefly above was developed for inviscid fluids, the rigorous inclusion of viscosity would necessitate a complete redevelopment of the magnetic stress equation. The simplest non-rigorous solution in this case would be to assume that the magnetic stress is the same for viscous and inviscid fluids, and to add an additional viscous term as follows:

$$\rho \frac{d\vec{u}}{dt} = -\nabla P + \underbrace{\mu_0 M \nabla H}_{\text{magnetic "forces"}} + \rho \vec{g} + \underbrace{\hat{\mu} \nabla^2 \vec{u}}_{\text{viscous "forces"}}$$

where:

- $\vec{u}$ = fluid velocity
- $\hat{\mu}$ = viscosity
- μ_0 = magnetic permeability in free space
- M = magnitude of magnetization
- H = magnetic field strength
- P = pressure
- $\vec{g}$ = gravity
- ρ = fluid density

The method of inclusion of viscosity above is a departure from the rigor of the inviscid magnetic fluid theory.

For the case of the rotary shaft seal, in which one has steady rotation of a concentric inner cylinder in a fixed outer cylinder, Figure A.3-90, the equations and boundary conditions show that the pressure difference across the gap is independent of the viscosity and is the same as that for static seal.

For a reciprocating piston seal, where one wall is fixed and another moves, it would appear that viscous effects cannot be neglected. In fact, a dimensionless number, A , defined as the ratio of magnetic forces to viscous forces can be shown to be:

$$A = \frac{\mu_0 M_s H L}{\hat{\mu} U}$$

where:

- M_s = the saturation magnetization
- H = the average magnetic field strength
- L = characteristic length
- U = characteristic velocity

Figure A.3-90. Rotary Magnetic Fluid Shaft Seal

In a subjective sense, one can say that the piston seal design should be oriented toward maximizing A so that magnetic forces are dominant. However, the "point" at which the seal would become workable is unknown. Preliminary calculations indicate that in the present piston seal design the magnetic and viscous forces are within the same order of magnitude.

In summary, analytical equations exist for the design of magnetic fluid devices. The design of a rotary shaft seal in particular appears well founded, the reciprocating seal less so. However, the electromagnetic theory has never been verified in moving media, so that these results are not known to be absolutely valid. Empirical evidence must ultimately be relied upon to prove the feasibility of designs.

Test Seals Performance - Series 1

Figure A.3-91 is a sketch of the initial shaft seals fabricated for testing. The primary flux path runs axially through the magnet, through the pole piece, through the magnetic shaft and back into the other end of the magnet through the other pole piece. The magnetic fluid is trapped in four discrete "O-ring"

stages, each capable of sustaining a portion of the differential pressure across the seal. The test setup is capable of monitoring differential pressure capability and power loss. The pole faces are designed to be removable for optimizing geometry and air gaps.

Figure A.3-91. Prototype Magnetic Fluid Shaft Seal

Figure A.3-92 is a photograph of the actual test seal using a samarium-cobalt washer magnet. The two pole faces at each end are shown displaced for clarity. Figure A.3-93 shows the test fixture open with the seal in the foreground. Some black magnetic fluid can be seen on the shaft. A calibrated motor, located in the housing, is used to rotate the shaft for viscous and eddycurrent loss determinations. The motor can be seen in another view in Figure A.3-94. Figure A.3-95 shows the test fixture with the two accurate pressure gages used to measure the differential pressure across the seal.

Figure A.3-92. Prototype Rotary Shaft Seal

Figure A.3-93. Rotary Shaft Seal Test Fixture - Seal End

Figure A.3-94. Rotary Shaft Seal Test Fixture - Motor End

Initially, a 2 mil radial gap, 4-pole design (2 poles, 7.5 mils wide per pole piece) was tested with 400 gauss fluid. The seal sustained a differential pressure of 40 psi without failure. Higher values were not tried due to the pressure limitation of a test gauge. The seal was, however, run at 2400 rpm at 15 psi. Eddycurrent loss was 40 mW.

A larger pressure gauge was placed in the setup and a 2-pole 2 mil radial gap design tested (15 mil wide pole face per washer). The highest maximum differential pressure sustained was 55 psi; typically, other test runs sustained only 25 psi. A 1-1/2 hour duration test was satisfactorily completed at 33 psi. Running the seal at 2400 rpm produced only 20 mW of eddycurrent loss. The large variation in pressure capability exhibited by the seal was attributed to filling difficulties caused by the 15 mil wide pole face structure where high fringing occurs on the two edges.

Preliminary tests were also run on a 2-pole design (15 mil wide pole face washers) with a 4 mil gap. The differential pressure sustained was 15 psi,

exactly the calculated value. Eddycurrent losses measured at 4800 rpm were 40 mW, indicating a 10 mW loss at 2400 rpm, the design speed.

Figure A.3-95. Instrumented Rotary Shaft Seal Test Fixture

A series of parametric tests varying pole face width were conducted using argon as the pressurizing medium to determine seal pressure differential capability. Figure A.3-96 gives the pertinent dimensions of the seal and washer magnet used for all tests. An air gap of 4 mils was used to minimize viscous losses. Figures A.3-97, A.3-98 and A.3-99 show the various test results obtained using 400 gauss ester-base magnetic fluid of 30 cp viscosity. Figure A.3-97 is a plot of the pressure capability of the seal as a function of pole-face width. The central solid curve is the average pressure capability whereas the dotted lines indicate the extremes in the data (three trials for each average value).

It is evident that pressure capability for the indicated geometry is maximized using a 7.5 mil pole-face width, although the spread in the data is also a maximum. This variability may be attributable to variations in radial air gap due to the relatively poor quality of the ball bearings used in the test fixture as well as to lack of refinement in loading technique.

Figure A.3-96. Rotary Test Seal Geometry

The pressure requirement of this seal in the heart engine is that it be capable of maintaining a 7 psi differential. The average pressure capability of 25 psi using a 7.5 mil pole-face is excessive since the flywheel housing, it is anticipated, will be charged through the seal. Eddycurrent loss is also of concern. Eddycurrent data is shown in Figure A.3-98. As can be seen, at the anticipated 2400 rpm of the flywheel shaft, eddycurrent loss for a 7.5 mil pole-face is about 20 mW. The spread in this data is approximately ± 10 mW due to the lack of sensitivity of the drive motor at such low power levels. Data was also taken at 4800 rpm to increase the dissipated power. The data correlates fairly well in that an expected four-fold increase in power consumption resulted.

Figure A.3-97. Shaft Seal Pressure Capability

Figure A.3-98, Eddycurrent Losses

Since it is desirable to charge the magnetic coupling housing via the engine crankcase, the inboard seal would have to be capable of blowing by at least one time then resealing to maintain the required pressure differential specification. Figure A.3-99 shows averaged data obtained for successive blow-bys. Again, the 7.5 mil pole-face would indicate the optimal width for maximum blow-by capability whether first, second or third times. The data indicates minimal percentage spread for the 7.5 mil pole-face width. Due to this spread, 7.5 mil pole-face blow-by No. 3 actually exceeded blow-by No. 2.

Figure A.3-99. Resealing Seal Data

Test Seals Performance - Series 2

These seals were basically similar to the first series but were designed to be used with tellus fluid of 180 gauss magnetic saturation. The pole-face air gap of 4 mils used with 400 gauss fluid was reduced to 2 mils. New samarium cobalt washer magnets were also procured. They are the same length as the ones previously used (0.093 in long), but their ID has been reduced for use with a 0.160 in. rather than a 0.188 in. diameter flywheel shaft. Pressure data obtained as a function of pole-face width is shown in Figure A.3-100. Argon was used to pressurize the seal. The results show that after the first blow-by, succeeding blow-bys sustained nearly the same ΔP level. The seal design goal is to sustain a 10 Hz fluctuating pressure wave of ± 7 psi.

The field resulting when using the optimum 16 mil wide pole-face was calculated to be 36,000 gauss in the air gap. The carbon steel pole-faces saturate at about 15,000 gauss; nevertheless, these high coercive force magnets continue to push additional flux through the circuit.

Armoco iron pole-faces were used for additional testing. Armoco iron saturates at 21,000 gauss. Limited ΔP data points using these Armco iron pole-faces and tellus magnetic fluid to pressurize the seal were obtained. This one-fluid seal simulates the condition in an engine using the magnetic fluid as a lubricant. The results, shown in Figure A.3-101, show that for the first blow-by, maximum ΔP also occurs using a 16 mil wide pole-face. Succeeding blow-bys, however, sustain reduced pressure differentials. This result was expected and is in contrast to the performance obtained using a gas wherein stages remain intact. During the first pole blow-by, magnetic fluid which is next to the seal fills the space between the poles. The original air is forced out of this space and past the second pole. Since the space is now filled with incompressible magnetic liquid, the two stages approach the performance of one. As all of the air is gradually displaced, the ΔP capability would be expected to asymptotically approach the value of one stage alone. Additional unreported data shows this to be the case.

Although the data is not sufficient to fully understand this effect, it appears that a substantial amount of trapped gas occurs with the 12 mil

Figure A.3-100. Magnetic Fluid Shaft Seal Pressure Capability With Argon

Figure A.3-101. Magnetic Fluid Shaft Seal Pressure Capability With Magnetic Fluid

wide pole-face. Succeeding blow-bys sustain nearly the same pressure. The data points plotted were averages of several values.

A pole-face geometry was investigated where a much thinner pole-face is used (0.025 in.) to reduce end leakage by capturing more of the field. Gauss measurements indicated that a portion of the field passed through the thinner pole-faces. The new pole-face is also tapered at its ID to provide the desired width. This geometry is advantageous when such large fields exist since the pole-face will be saturated over a considerably shorter length and more magnetomotive force will be available for the air gap. A pressure capability of 20 psi resulted on the first blow-by and any number of successive blow-bys resulted in a 10 psi differential. The calculated field is 48,000 gauss.

Test Seal Performance - Series 3

The tapered pole-face geometry was fabricated and tested. Figure A.3-102 shows the geometry of the seal and its test data. The magnetic fluid 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 magnetic fluid. For comparison, test data was taken of a 4340 engine shafting steel and for 1018 steel which is magnetically superior. As the data shows, there is actually improved performance using the 4340 steel shaft.

The data in Figure A.3-102 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 (one blow-by), 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.

The power consumption of the two-pole seal was measured to be about 10 mW using a spring dynamometer. 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

Figure A.3-102. Two-Pole Shaft Seal Pressure Capability

shaft. Data was also obtained for a four-pole seal configuration. The results are shown in Figure A.3-103. Note that the maximum pressure capability of the first blow-by is about the same as the 2-pole design. This seal is of doubtful value due to its added complexity. Furthermore, its only advantage over the 2-pole design is a factor of two reduction in the air gap magnetic field. This could promote magnetic fluid stability. Nevertheless, unless there is a problem with the stability of the present magnetic fluid, use of this seal is not indicated.

Test Seals Life

Life testing of shaft seals was carried out using a Pella 11 and Tellus 15 magnetic fluids. Both tests were discontinued intentionally. There was no evidence of seal failure through the time period of the tests.

Pella 11

Pressure Differential 10 psi

Time 1000 hours @ 7600 rpm

..... 3382 hours @ 2400 rpm

TOTAL 4382 hours

Tellus 15

Pressure Differential 16 psi

Time 3215 hours @ 2400 rpm

Pella 911

Pressure Differential 10 psi

Time 3192 hours @ 2400 rpm

Further Magnetic Fluids Investigations

For engine use the rotating magnetic fluid shaft seals must be effective against engine oil. Sealing of a liquid differs substantially from sealing a gas as mixing and diffusion of the two liquids seem to cause eventual failure. The

Figure A.3-103. Four-Pole Shaft Seal Pressure Capability

initial approach to a solution to this problem as implied in the piston seal development discussion was to identify those fluids which were immiscible in Tellus crankcase oil. A number of such liquids were identified and tested as potential magnetic fluid agents. Initial tests were in static seals. Later testing was in rotating seals. All rotating seal tests were failures. 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 magnetic fluid 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 magnetic fluid diffuse, are dragged away, and cannot be held from escaping the magnetic field, even under non-rotating shaft conditions. Eventually enough of the particles escape to ruin the sealing potential of the fluid in the magnetic field area.

The approach was then shifted to using the magnetic fluid as the engine lubricating oil. An investigation to characterize the properties of Tellus oil and Pella oil magnetic fluids in terms of engine use was undertaken.

Some of the necessary properties as a lubricant are:

- Must have suitable viscosity
- Must have long-term stability
- Compatible with rest of engine components, particularly roll sock piston seals
- Must not plug filters nor damage bearings nor lube pump

The seal life tests, previously reported, using Pella 11 and Tellus 15, were undertaken primarily to study long-term stability. Long-term tests have been carried out circulating these fluids through a closed loop containing a gear pump and two micron oil filters, and will be reported in Subsection A.3.1.5, "Lubrication System Development."

A mode of failure of a magnetic fluid is agglomeration of the colloidal suspension of the iron particles. The particles are coated for stabilization, the substance used is, most frequently, stearic acid.

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 Å and an overall diameter of 154 Å or less, agglomeration will not occur even in an infinitely strong magnetic field. With this in mind, Pella 11 magnetic fluid was procured with the smallest available particles, which are 100 Å. The same 100 Å particle size had been used previously to make the Tellus 15 ferrofluid. Figure A.3-104 shows an SEM photograph of this fluid. Typical coated particle size appears to be 150 Å or at about the critical limit; however, rigorous particle size measurements were not conducted.

Pella 11 oil is similar in composition to low viscosity Vitrea oils for roll-stock compatibility. The fluid was ordered as mentioned, with the smallest particle size available (nominally 100 Å) 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 magnetic fluid 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 A.3-105 shows the test data taken of magnetic fluid 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.

A magnetometer, shown in Figure A.3-106, was made prior to obtaining the Pella 11 magnetic fluid. It was 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 4000 turns of wire forming a coil and the leads are attached to an integrating voltmeter. Introduction of magnetic fluid into the non-magnetic

Figure A.3-104. TEM Photograph of Tellus Magnetic Fluid

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 magnetic fluid injection and is related to the magnetic saturation of the magnetic fluid (provided the field significantly exceeds the strength). The calibration curve for the device was obtained by introducing various known strength magnetic fluids. As the original Pella 11 magnetic fluid reduced in viscosity, its magnetic strength was determined using this device.

PELLA # 11 MAGNETIC FLUID
210 GAUSS

Figure A.3-105. Temperature-Viscosity Data for Pella 11 Magnetic Fluid

MAGNETOMETER CALIBRATION

Figure A.3-106. 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.

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

where:

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 magnetic fluid 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 even higher. Consequently, a shaft seal fixture was modified to accept a 0.160 in. diameter shaft that fit with a 0.26 mil radial clearance in a bearing in order to obtain shear rate data. A sensitive torsion spring dynamometer was used for measuring torque.

Pella 911 magnetic fluid shear rate data was taken to $255,000 \text{ sec}^{-1}$ and is applicable to the flywheel bearing which exhibits a shear rate of $133,000 \text{ sec}^{-1}$ with zero eccentricity and $213,000 \text{ sec}^{-1}$ with an eccentricity of 0.4. Figure A.3-107 is a plot of net shaft torque (directly proportional to viscosity) versus shear rate. The results for both mineral oil as the control and magnetic fluid are presented.

The reason for the "apparent" nonlinearity and slightly non-newtonian behavior at shear rates beyond 125,000-sec is probably due to heating of the oil in the bushing with an associated small decrease in viscosity. This effect is more clearly seen in Figure A.3-108 where normalized viscosity was computed from

Figure A.3-107. Magnetic Fluid Behavior as a Function of Shear Rate

Figure A.3-108. Viscosity of Magnetic Fluid and Mineral Oil at High Rates of Shear

data in Figure A.3-107 and plotted versus shear rate. The data for both lubricants is now seen to be identical, indicating that they behave the same. Verification of this result was the main purpose of these shear rate tests. The data was reproduced whether starting tests at low speeds and working up to high speeds, or vice-versa.

The shear rate ranges expected for various engine bearings are also pointed out in Figure A.3-108 for design ranges of eccentricities. Since it is clear that the magnetic fluid behaves exactly like Tellus IVBM oil, this lubricant is expected to perform properly in the Mark I engine. Furthermore, the newtonian behavior of the magnetic fluid has been verified and a Brooks viscometer can therefore be used at low rates of shear for valid viscosity measurements. This was also confirmed by making flow rate and pressure drop measurements on a capillary column.

To vie as a successful engine lubricant, the magnetic fluid must also demonstrate rollsock compatibility for the Mark I Prototype engine. Rollsock samples were immersed at 120°F in Tellus 25 oil as the control and Pella 11 magnetic fluid. 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 tests indicated that the magnetic fluid samples performed slightly better than the Tellus 25 samples. This is attributed to a lack of boundary and anti-foaming additives in the Pella 11 base oil.

A.3.1.5 Lubrication System Development

Major lubrication system development efforts were carried out in the areas of oil pumps, mechanical shaft seals, and gravity independent oil intakes. Lesser efforts were conducted in areas of magnetic fluid filtering and wear and crank-case oil churning loss assessment.

Magnetic Fluid Filtering and Wear

A requirement of the magnetic fluid for engine use is 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 endurance test was started for determining the following:

First, whether magnetic fluid can be filtered using the contemplated 2 micron mesh screening; and second, whether magnetic fluid particles cause oil pump wear. A closed circulating system was constructed whereby a hardened-gear pump discharged 60 psi oil into a 0.75 in. 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 Tellus 25 oil and uncut Pella 11 magnetic fluid.

Because the cross-sectional area of the filter is 16 times less than that of the Mark I Prototype filter and the pump flow rate 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 magnetic fluid particles.

When the 2 micron oil filter test fixture had accumulated 1260 hours using Pella 911 magnetic fluid, the pressure drop across the filter began increasing. The test was stopped and the filter element was inspected. Microscope analysis showed that extraneous dirt particles had clogged the element. There was no evidence of magnetic fluid clogging or deterioration. The setup had operated stably for this period of time, which is equivalent to 6300 hours of operation of the Mark I filter. There has also been no evidence of gear pump wear or pressure drift other than from ambient temperature changes. The test was then restarted.

The magnetic fluid filter/oil pump test fixture had accumulated 2026 hours when it was decided to stop the test in order to determine pump wear. Pump flow rate versus pressure drop was determined under identical conditions to those used before the life test. The data reproduced within 1 percent (within the measurement accuracy), indicating that no pump wear nor any increase in internal leakage had occurred. The filtering test was then resumed.

In an effort to see whether any of the magnetic fluid particles had been damaged by the oil pump, a sample of oil was withdrawn and prepared for TEM photographs. A special effort was made to use the same carbon substrate preparation technique as was used when the particle coatings could be seen as shown, previously, in Figure A.3-104.

We were unsuccessful in observing the coating although the particles can be seen. It is not known whether the sample and/or substrate preparation was at fault or whether the electron beam vaporized the coating. The magnetic fluid filtering test was continued through 3130 hours of operation. There was no change in pressure drop across the 2 micron filter.

Oil Pump Development

Substantial effort went into the development of an adequate engine lubrication oil circulating pump for the current nuclear system engine. In addition an oil pump for a future Mark I Prototype engine was designed, fabricated and tested. All pumps were of the spur-gear type.

Initial 2-Gear Pump (1/8 in. Pitch Diameter)

The initial 2-gear oil pump designed for the current nuclear system engine called for a zero leakage flow of 0.14 cc/sec ($0.0086 \text{ in}^3/\text{sec}$) and a flow of 0.1 cc/sec ($0.006 \text{ in}^3/\text{sec}$) against a head of 9.5 kg/cm^2 (135 psig) at 600 rpm. Its power requirement was estimated at 0.5 watt and its efficiency at 28 percent. The 20° tooth angle spur gears selected have a pitch diameter of 0.32 cm (0.126 in.) with 12 teeth per gear; the gears face width was 0.25 cm (0.10 in.). Figure A.3-109 shows the component parts of this initial gear pump.

Five of these gear pump assemblies were fabricated. A test setup was constructed wherein pump speed and output pressure could be accurately controlled, and input power and output flow could be accurately measured.

When first placed on test, all five pumps performed unsatisfactorily. The following table summarizes the results of this first test phase.

Figure A.3-109. IVBM Gear Pump

INITIAL GEAR PUMP PERFORMANCE SUMMARY

Pump Number	FIRST TEST PHASE		SECOND TEST PHASE	
	Peak Pressure, psig	Flow Rate @ 135 psig, in ³ /sec	Peak Pressure, psig	Flow Rate @ 135 psig, in ³ /sec
1	230	0.0034	>400	0.0048
2	120	--	350	--
3	85	--	175	--
4	65	--	125	--
5	110	--	190	--

A pressure of 135 psig at 600 rpm with a delivery of 0.006 in³/sec was the performance objective. None of the pumps met this goal; indeed, only one could achieve 135 psig with any flow at all. The problem common to all these assemblies was that the internal leakage of oil through clearances within the pump was far greater than that allowable. The volumetric pumping capability of all these pumps is 0.008 in³/sec at 600 rpm with no pressure head (hence no back leakage). At a pressure of 135 psig, only 25 percent leakage is tolerable. All five pumps exceeded this allowance.

There are three principle leakage paths within the pump; directly down the centerline of the pump through the meshing teeth, through the clearance between the tips of the gear teeth and the housing around the periphery of the pump, and through the clearance between the ends of the gear teeth and the front and back housing plates. The leakage through the first path is determined by the quality of the gears themselves. The back pressure through the pump actually helps to seal the gears at the meshing interface. It was not felt that this leakage path was the problem. The severity of the leakage through the other two paths is a cubic function of the applicable clearance dimension, and they were felt to be the cause of the problem.

An attempt was made to improve the performance of the five assemblies by reducing the width of the center plates enough to reduce the side leakage

clearance to less than 0.1 mil. Nothing could be done to reduce the tip clearance without completely remaking the center plates. The results of this first change are shown in the preceding table under "Second Test Phase." The maximum pressures attainable increased markedly for all the pumps, and the delivery of Pump #1 was almost good enough to be rated acceptable. In general, however, the pumps were still unsatisfactory. The conclusion was that the tip clearance would have to be reduced.

Mod 1 2 Gear Pump

Figure A.3-110 is a sketch showing the clearance between the tips of the gear teeth and the housing (scroll plate). An analysis was undertaken to quantify the effect of the tip clearance on the delivery of the pump, assuming one-dimensional Couette flow through the tip clearance. Based upon this analysis, a new pump housing was constructed for the same gears previously used in Pump #2. All clearance tolerances were decreased to 0.1 mil maximum. Additionally, the scroll plate was redesigned to permit a greater number of teeth to seal on the scroll. Initial tests on this reworked pump showed a delivery of $0.0075 \text{ in}^3/\text{sec}$ at 135 psig and 600 rpm.

Figure A.3-110. Schematic of Gear Pump/Housing Interface

Two additional gear pumps were modified to incorporate these design changes. Both pumps received 500 hours of run-in at 600 rpm with no change in performance noted. Figure A.3-111 shows the performance characteristics of Pump #3. The performance of the other pumps was very close to that shown in this figure.

Figure A.3-111. Experimental Results With IVBM Gear Pump #3

3-Gear Pump 1/8 in. Pitch Diameter

Figure A.3-112 illustrates the first version of a 3-gear oil pump. This pump development was undertaken in order to double the delivery capability of the oil pump while, ideally, incurring less than a 100 percent increase in pump power. In essence, the pump can be thought of as having two pairs of rotating gears. The input sides of both "sets" are connected as are the high pressure output sides. In the illustrated design, only two tubes connected to the pump while the manifolding is done internally. The first two pump assemblies failed

Figure A.3-112. Three-Gear Oil Pump for IVBM

when the teeth of the drive gear were stripped during run-in. Excessive power was available from the drive motor, and when the idler gear teeth seized in the housing, the high torque sheared the teeth.

Although the third pump attempted was successfully run in, the back leakage under pressure was excessive. A redesign of the housing was completed and the revised design was fabricated and tested. The changes made involved the internal manifolding circuit and the substitution of steel for brass as the material from which the scroll plate is made. Both changes were made to improve the internal leakage demonstrated by the first unit. Figure A.3-113 shows the power requirement and flow rate versus output pressure for this pump using Tellus 25 oil. While the internal leakage is more pronounced than that of the previous 2-gear pump, it was acceptable for use in the current engine. At 95 psi output pressure, the flow rate is $0.0135 \text{ in}^3/\text{sec}$, and the power loss is 0.45 watts.

Current nuclear system engine No. 2 was fitted with the 3-gear oil pump. During engine testing, the maximum pressure to which the internal 3-gear pump could pressurize the system was found to be 75 psi using Tellus 25 oil. The engine was run for several days in this mode while pressurized solely by the internal pump. During subsequent rework of the bearings in the engine mechanism, a leakage between internal pressure and suction lines was discovered inside one of the pillow blocks. This leakage path was caused by a drilled cross-hole breaking out the lubrication pressure space into the suction space. The hole had not been uncovered by radiography in previous inspections. It is possible that this leakage path may have been partly responsible for the low oil system pressures experienced.

Second 2-Gear Pump 3/16 in. Pitch Diameter

Two oil pumps were fabricated from inexpensive stock gears 3/16 in. long with pitch diameter (PD) of 3/16 in., diametral pitch of 64, and integral pinion shafts. The gears were ground to a face width of 0.110 in., and two support shafts, 0.093 in. diameter, were developed from the excess length on one side of the gear and the internal pinion shaft on the other. Additionally, several

thousands of an inch were ground off the outer diameter to assure its concentricity to the shafts. The housings were designed so that this pump could be retrofitted into the current nuclear engine system.

Figure A.3-113. Performance Data for 3-Gear Oil Pump

Figure A.3-114 shows the first of these pumps. This pump performed rather badly when placed on test. The radial clearance in its housing was somewhat higher than desired, the gear center spacing of 0.1875 in. was later proven to be too large, and inspection revealed some defects in the tooth profile.

Figure A.3-114. 3/16 in. Pitch Diameter Gear Pump

The result was that back-leakage through the pump was excessive. In an effort to improve performance, the side clearance between the housing and the gears was reduced in several increments, and finally eliminated altogether. The delivered volumetric flow of the pump increased somewhat in each instance but so did the input power. The wear marks apparent on the side plates in Figure A.3-113 resulted from running with zero side clearance.

The second pump fabricated using the stock gears but whose teeth were verified to have undamaged profiles, had a reduced radial clearance housing and reduced center-to-center distance between gears. This pump performed very well, as can be seen in Figure A.3-115. The delivery at zero back pressure was found to be $0.018 \text{ in}^3/\text{sec}$, somewhat less than the intended $0.021 \text{ in}^3/\text{sec}$. This was calculated to be due to the reduction made in the gear OD to make the OD of these stock gears concentric with their shafts. Precision gears would not need this rework and consequently would provide somewhat higher flow. The back-leakage through the pump was quite acceptable, with a flow of $0.017 \text{ in}^3/\text{sec}$ at 60 psig pressure. The input power was also quite low: 0.22 watt at 60 psig.

All the data in Figure A.3-115 was taken with Tellus 25 oil so that comparisons in performance could be made with earlier pumps. As can be seen, this new pump provides 30 percent more flow than the 3-gear pump while requiring less power. Used in the prototype lubrication system with a lower viscosity oil, this pump will still have adequate delivery while requiring even less input power, well within the design allowance of 0.25 watt. This second pump was retrofit into current engine No. 1, replacing the undercapacity 1/8 in. PD, 2-gear pump.

Mark I Prototype Gear Pump 3/16 in. Pitch Diameter

The developmental oil pump for the Mark I Prototype engine utilizes two spur gears. A pitch diameter of 3/16 in., a diametral pitch of 64, and a length of 0.110 in., was selected for the gears. The pump is designed for a volumetric delivery of $0.021 \text{ in}^3/\text{sec}$ of 25 centipoise oil at 60 psig. This design was fabricated and tested. The pump was the most efficient of any constructed on this program. Typical performance characteristics with Tellus 25 oil at 600 rpm and 60 psig were as follows:

Figure A.3-115. Gear Pump Performance Comparison

Delivery 0.0190 in³/sec
Input Power 0.250 watt
Efficiency 51 percent

Mechanical Shaft Seals

Testing and limited redesign of seals for the current nuclear system engine was carried out. A substantial amount of analysis and design optimization of screw-type seals for potential use on the Mark I Prototype Engine was carried out and is reported hereafter.

Current Engine Shaft Seals

The flywheel shaft seal for the current engine, as has been described in Section 3.0 of Volume I of this report, is a combination of screw thread viscosity pumping groove and cylindrical clearance seal. Testing of this shaft seal was successfully terminated following 650 hours of continuous operation against one atmosphere pressure gradient. Prior to this test, the same seal had been tested for over 700 hours against a pressure head of 12 in. of oil with no evidence of leakage. These tests demonstrated the feasibility of the seal concept.

The engine output screw-type shaft seal has been described in Section 3.0 of Volume I of this report. It consists of a left-hand and a right-hand screw thread viscosity pumping groove. The oils on either side of the seal are pumped inwards toward an interface across which no mixing or mass transfer should occur; the difference in viscosity of the two oils is accommodated by the number of threads cut into each half of the seal.

Initially over 1000 hours of testing were accumulated with Vitrea 21 oil on the engine side and sperm oil on the blood pump side. These initial tests were performed at room temperature with the two oil reservoirs at atmospheric pressure. Later a more severe test was configured with a positive oil pressure on the blood pump side oil reservoir equal to a 10 in. column of sperm oil. The apparatus is shown in Figure A.3-116. After 1300 hours of continuous operation the test was intentionally terminated.

Figure A.3-116. Thermal Converter/Blood Pump Seal Test Apparatus.

Following the test with positive sperm oil pressure, a test was begun with a negative sperm oil head (vacuum of 10 in.). This test ran for 840 hours with no fluid transfer. Over the next 48 hours, a steady transfer of fluid from the sperm oil side to the Vitrea 21 side occurred, depleting the sperm oil reservoir. The test was terminated and the fixture disassembled. No obvious cause for this phenomenon was apparent.

An assessment of the effectiveness and power penalties associated with a plastic, spring-loaded, lip seal on the output shaft was undertaken. A design to utilize this seal in the existing engine crankcase is shown in Figure A.3-117. The power penalty associated with this seal was measured to be 0.2 watt. A life equivalent to approximately 30 days of running at 600 rpm was demonstrated.

Mark I Prototype Engine Shaft Seals

Figure A.3-118 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:

$$\frac{Q}{\cos^2 \phi} = \frac{\omega D}{4} h \ell \left[1 + \frac{h_0}{h} \frac{\ell_0}{\ell} - \pi D \frac{\sin \phi}{\cos^3 \phi} \left(\frac{1}{\ell} \frac{h_0}{h} + \frac{2}{\pi D \sin \phi} \right) \right] \\ - \Delta P \frac{h^3 \ell}{12 \mu \pi D} \left[\frac{1}{N} + \frac{1}{(N+1)} \left(\frac{h_0}{h} \right)^3 \frac{\ell_0}{\ell} \left(\frac{\pi^2 D^2}{\ell_0^2 \cos^4 \phi} + 1 \right) \right]$$

or

$$Q = C_1 - C_2 \Delta P$$

Figure A.3-119 is the experimental data of flow versus pressure for an initial prototype seal. It is a straight line relationship of negative slope as Equation (1) predicts. For the following screw values:

$h = 0.0031$	$N = 0.2$ grooves
$h_0 = 0.0004$	$N + 1 = 1.2$ lands
$\ell = 0.0583$	$\mu = 7.71 \times 10^{-6} \frac{\text{lb-sec}}{\text{in}^2}$ viscosity
$\ell_0 = 0.0250$	$\omega = 251$ rad/sec
$D = 0.1560$	$\phi = 9.6^\circ$ or 12 threads/inch

APPLICATION		REVISIONS			
NEXT ASSY.	USED ON	LTR.	DESCRIPTION	DATE	APPROVED

SPACER

"LIP" SEAL
(BAL-SEAL)

GEAR BOX
LOCATING RING
AND SEAL
RESTRAINT

PHILIPS
ENGINE CRANKCASE

"WANL"
GEAR BOX

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES. TOLERANCES ON FRACTIONS $\pm 1/64$ TWO (2) PLACE DECIMALS $\pm .010$ THREE (3) PLACE DECIMALS $\pm .005$ ANGLES $\pm 0^{\circ}30'$	PROGRAM NO. <u>611</u>	PHILIPS LABORATORIES A DIVISION OF NORTH AMERICAN PHILIPS CORPORATION BRIARCLIFF MANOR, N. Y.
	DRAWN <u>G. CULHANE</u> <u>12.23.75</u>	
MATERIAL:	CHECKED	

Figure A.3-117. Output Shaft Seal Configured With Lip Seal

Figure A.3-118. Screw Seal Geometry

The flow rate of zero pressure is predicted to be $1.75 \times 10^{-3} \text{ in}^3/\text{sec}$. This is in good agreement with an experimental value of 1.5×10^{-3} seen in Figure A.3-119 at the ordinate intercept. Q was also measured to be independent of viscosity as expected.

When used as a seal rather than as a pump, the maximum sustained screw pressure is of most importance. When $Q = 0$ in the shaft seal equation, the maximum ΔP of the seal is determined. The data in Figure A.3-119 shows ΔP_{max} to be experimentally 13.5 psi when using 48 cp oil. An expression for ΔP_{max} is as follows:

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

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

Figure A.3-119. Screw Seal Data

model. Nevertheless, the model has provided much insight for continued improvement in the seal. For example, a second screw seal was designed 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 into the $\Delta P_{\max}$ equation, then differentiating with respect to ℓ_o and equating to zero, leads to the result that:

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

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

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

where:

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

Figure A.3-120 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 seal has $N = 2.0$ grooves, 28 threads/inch, a land and groove width of 0.018 in., and is only 0.125 in. long overall. Figure A.3-120 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_0 unchanged is only 18 mW.

Figure A.3-120. Seal Pressure Capability as a Function of Number of Grooves

A screw seal was designed to fit a 0.160 in. diameter flywheel shaft as an alternate to a magnetic fluid crankcase seal. Figure A.3-121 is a plot of pressure versus thread depth for this 0.125 in. long seal with particular emphasis on the effects of eccentricity. As expected, pressure capability is decreased as eccentricity approaches its limiting value of unity. With each point on the respective curves is associated an optimum groove and land width, and number of threads.

Figure A.3-121. Optimized Screw Seal Pressure Versus Thread Depth and Eccentricity

A seal of the following proportions, which are associated with optimal pressure capability with an eccentricity of 1, was fabricated:

Thread depth 0.003 in.
Land width 0.016 in.
Groove width 0.022 in.
Number of grooves 2.2

This screw seal has been tested and its performance verified. The screw seal was tested in a simulated oil-filled crankcase where ± 7 psi sinusoidal oil pressure was applied to one end of the seal. The other end was exposed to air in a simulated flywheel housing. It was found that for these simulated conditions (equivalent to a 90 percent filled Mark I engine crankcase), the screw seal pumped the flywheel housing down 23 in. of water column at 2400 rpm. This will have no effect on screw seal or engine performance since the screw is capable of completely sealing oil for up to a 16 psi differential.

To verify the leakage-free character of screw seals, an older screw seal was set up to pump oil directly into a pressure gauge to its maximum attainable pressure (12 psi). Oil was removed from the back of the seal to simulate the flywheel housing interface and the screw was positioned vertically to promote the drainage of any oil that might leak. After 12 hours, when the test was discontinued, there was no evidence of oil leakage past the seal nor did the pressure gauge reading change.

A study was also made of a screw for sealing blood pump oil out of the magnetic coupling housing.

Since ball bearings will support the screw seal shaft in the gear box, it is desirable to provide the required 4 psi pressure capability with as large a radial clearance as possible. The seal performance associated with a 0.125 in. long, 0.6 mil radial clearance screw is shown in Figure A.3-122. The results for both a 3/16 in. and 1/4 in. diameter shaft are presented for 33 cp oil. Shaft eccentricity used is unity giving the worst possible condition. As can be noted, a 3/16 in. diameter shaft will provide 6.3 psi pressure capability.

The optimum thread depth is 5.5 mils. The following table lists the optimum pressure attainable for a 3/16 in. shaft for other radial clearances.

<u>P (psi)</u>	<u>h_o (mils)</u>
15	0.4
6.3	0.6
3.4	0.8

Figure A.3-122. Blood Pump Screw Seal Performance for 3/16" and 1/4" Shaft Sizes

An additional design based on 1/4 in. diameter shaft, 0.250 in. long was optimized. This larger size (compared to the previous 0.125 in. long seal) will allow good sealing capability with larger radial clearances, larger eccentricities, and lower oil viscosities (a consequence of the higher oil temperature when used as the system coolant). Figure A.3-123 presents the theoretical performance characteristics of the seal showing in particular that an optimal thread depth exists for maximizing the pressure capability. The design point chosen is

The following are optimal thread proportions:

Maximum diametral clearance	 0.0016 in.
Maximum eccentricity ϵ_0	 0.5
Thread length	 0.250 in.
Major diameter	 0.250 in.
Number of threads	 3.0
Threads/inch	 14
Thread depth	 0.005 in.
Thread width	 0.314 in.
Groove width	 0.040 in.
Helix angle	 5.19°

Performance data:

Pressure capability @ 2400 rpm ϵ_0	 0.5
6.2 psi @ 120°F	 normal
5.6 psi @ 128°F	 fever
Power consumption	 14 mW

Lubrication Oil Pump Intake Development

An artificial heart system must operate independent of gravity orientation. One of the requirements for independence of gravity is an ability to supply lube oil to the oil pump inlet at all orientations. Development was carried out on two schemes. The first of these is the screen-wicking arrangement of the current nuclear system engine. The second is use of a magnetic inlet with magnetic fluid lube oil of the Mark I prototype engine design.

DIAMETER CLEARANCE = Δ
ECCENTRICITY = E_0
VISCOSITY = 12.5 c.p.
SPEED = 2400 R.P.M

Figure A.3-123. Output Shaft Screw Seal Pressure Capability

Capillary Wicking

Material selection and proof-of-principle tests of the capillary wicking system have been carried out. An analysis of candidate wire gauge mesh sizes (see Table A.3-4) led to the selection of 325 x 325, 0.0011 in. wire diameter stainless steel mesh for further study.

TABLE A.3-4

<u>Mesh</u>	<u>Wire Diameter Inches</u>	<u>Theoretical Wicking Height - Inches</u>	<u>Max. Flow Rate Attainable in³/sec (3-layer mesh)</u>
270 x 270	0.0011	5.0	0.053
270 x 270	0.0016	6.3	0.045
325 x 325	0.0011	6.6	0.066
325 x 325	0.0014	8.2	0.047
400 x 400	0.0011	9.4	0.061

The wicking height of Vitrea 21 oil in a 3-layer sample of the candidate mesh measured 5.75-in. compared with the theoretically predicted value of 6.6 in. This height would still permit a flow rate of 0.053 in³/sec through the IVBM intake system, more than six times the anticipated flow rate.

The test setup in Figure A.3-124 simulates the IVBM oil intake system and was used to measure the flow rate at which surface tension breakdown occurs. The intake wicks are immersed in oil, while the wicks exposed to air are housed in lucite at a height of 2.5 in. above the oil's free surface (simulating the maximum wick spacing in the IVBM system). The system was primed with oil and a Phase II oil pump used to circulate the oil. The flow rate which caused air ingestion into the exposed wick was measured. The candidate wick described above maintained an oil flow of 0.051 in³/sec without breakdown, confirming the design selection.

Several mechanisms by which the air bubbles could detrimentally affect the performance of the lubrication oil pump inlet wicking system were identified and systematically studied.

Figure A.3-124. Capillary Wick Test Setup

The presence of air bubbles in the crankcase could have the following effects:

- A clogging effect of crankcase bubbles collecting at the screened intakes, decreasing the flow rate, or
- A collection of micro-sized bubbles might cause a breakdown of the surface tension "seal" at the intake port allowing significant amounts of air from the crankcase to be ingested by the oil pump.

The possibilities concerning clogging due to bubbles and seal breakdown were studied in experiments that measured the resistance to flow directly across the wire mesh.

The basic equation for pressure drop across the mesh is:

$$\Delta P = -R_{\text{mesh}} Q$$

where:

$$\Delta P = P_2 - P_1$$

Q = volume flow rate

R_{mesh} = resistance to flow across the mesh

For a fixed cross-sectional area and oil:

$$R_{\text{mesh}} = f(\text{mesh})$$

so that R_{mesh} is expected to increase for tightly woven wire meshes. It is necessary to find R_{mesh} for different types of meshes to determine if the surface tension seal at an exposed mesh would break down due to a large pressure drop, and to determine if R_{mesh} increases significantly due to bubbles collecting at the upstream side of the mesh.

Figure A.3-125 is a schematic of the experimental set-up which was used. The total pressure difference between points #1 and #2 is the sum of the pressure drop in the tubing and the drop across the mesh. $P_1 - P_2$ can be measured directly by the U tube and heights of oil in the lines above points #1 and #2.

Figure A.3-125. Apparatus to Determine Flow Resistance of Intake Screens

Assuming linearity:

$$R_{\text{total}} = R_{\text{mesh}} + R_{\text{no mesh (tubing)}}$$

Thus, we can obtain the mesh resistance by calculating R_{total} (with a mesh) and subtracting the resistance without a mesh ($R_{\text{no mesh}}$) in the tubing above, where:

$$R_{\text{total}} = \frac{P_2 - P_1}{Q} \quad (\text{with mesh})$$

and

$$R_{\text{no mesh}} = \frac{P_2 - P_1}{Q} \quad (\text{no mesh})$$

Tests were conducted utilizing Shell Oil Tellus 29 both with the engine off (no bubbles and no heat) and with the engine running. Table A.3-5 lists some of the values of R_{mesh} calculated from this data.

In Table A.3-6 h_w is an experimental value for the head supported by the surface tension forces at the exposed mesh, and A is a calculated value based on the geometry of the lubrication system lines as well as on the value of h_w . The seal will hold when $R_{\text{mesh}} < A$.

Several conclusions were drawn from the experimental results:

- There is no clogging effect at the mesh, as seen in Figures A.3-126 and A.3-127. Rather, the resistance actually decreases as t increases due to an increase in oil temperature and a corresponding decrease in oil viscosity. During the first 30 minutes, this temperature effect dominates the resistance.
- The surface tension seal will hold for the 325 x 325 sintered mesh used in the engines, and should also hold for a 165 x 1400 1-layer mesh (Table A.3-6).
- The resistance between two different pieces of sintered mesh can differ to a large extent, as seen in Table A.3-5 for 325 x 325 mesh. Here, the percent change in the two values is:

$$100 \left(\frac{1.20 - 0.83}{1.20} \right) \approx 31\%$$

Thus, the overlap and orientation of the layers may alter the resistance of the sintered mesh.

- The resistance of the 400 x 400 and 325 x 325 meshes tested (Table A.3-5) are close in value if we use the result from Run 8 for the 325 x 325 mesh.

These tests demonstrate that clogging of the intake ports or loss of the surface tension seal in engine tests due to air bubbles is unlikely.

TABLE A.3-5
MESH RESISTANCE TEST RESULTS

Oil	Run No.	Date	Mesh	Engine	$R_{\text{mesh}} \left(\frac{\text{psi}}{\text{cm}^3/\text{sec}} \right)$
Tellus 29	8	01/30/75	325 x 325 sintered 3-layers	Off	1.20
Tellus 29	8	02/20/75	325 x 325 sintered 3-layers	Off	0.83
Tellus 29	7	02/20/75	325 x 325 1-layer	Off	0.23
Tellus 29	6	02/20/75	400 x 400 1-layer	Off	0.47
Tellus 29	9	02/20/75	450 x 2750	Off	11.45
Tellus 29	10	02/20/75	165 x 1400	Off	3.15
Tellus 29	5	02/20/75	400 x 400 sintered 3-layers	Just Started	1.27 at $t = 0$

Magnetic Oil Pump Intake

In contemplation of the use of Pella 911 magnetic fluid lubricant, work was initiated and completed on the development of a magnetic fluid intake for the oil pump. This type of intake will permit an engine to run with a partially filled crankcase in any orientation. It has these additional advantages:

- The sinusoidal pressure on the flywheel housing shaft seal is eliminated.

- Viscous churning losses are reduced.
- The oil will be pre-filtered before entering the pump.

TABLE A.3-6
SURFACE TENSION SEAL CALCULATIONS

Oil	Mesh	h_w (cm)	$R_{\text{mesh}} \left(\frac{\text{psi}}{\text{cm}^3/\text{sec}} \right)$	$A \left(\frac{\text{psi}}{\text{cm}^3/\text{sec}} \right)$
Tellus 29	325 x 325 1-layer	21.6	0.23	1.37
Tellus 29	400 x 400 1-layer	--	0.47	--
Tellus 29	325 x 325 sintered 3-layers	24.2	0.83	1.61
Tellus 29	400 x 400 sintered 3-layers	--	1.27	--
Tellus 29	165 x 1400 1-layer	48.2	3.15	3.92
Tellus 29	450 x 2750 1-layer	>91.	11.45	>8.0

Figure A.3-128a is a photograph of the intake while in operation. A gear pump is delivering ferrofluid lubricant through the tubing and is depositing it in the vicinity of the magnet. The magnetic field gradient generated by the pole faces is bending the free surface of the oil flow toward the apex of the pole face cone and is actually forcing the fluid to travel up-hill toward it. Figure A.3-128b is a schematic of the device showing the samarium cobalt magnet and conical pole pieces. This setup was run for several weeks in a closed oil loop. It has also been tested under shock load conditions. Theoretically, an acceleration of thousands of g's would be required to dislodge oil from the intake line, yet a differential pressure of only 12 in. of oil exists across it.

R_{TOTAL} VS TIME

RUN 5, 2/20/75 400 X 400 SINTERED MESH,
3 LAYERS, SECOND SET OF DATA POINTS

ENGINE STARTED AT $\Delta t = 0$

OIL, TELLUS 29

Figure A.3-126. Mesh Resistance Versus Time
(With Air Bubbles)

During engine operation, oil that is splashing in the crankcase will continuously keep the inlet filled. This is because oil which is splashed into the vicinity of the device is forced by the magnetic field to converge to the inlet. Magnetic field plots have been made in the free space surrounding the pole faces. These plots and experimental data indicate that free droplets of oil will be pulled into the inlet against gravity within a sphere of about 0.5 in. radius around the inlet.

The magnetic intake advantage of operation as a particulate filter is valid for both magnetic and non-magnetic particles. Non-magnetic particles even as dense

as lead are buoyed out of the magnetic fluid by the field gradient. Air bubbles are no exception. During operation, air bubbles and foam can be seen to float on the surface of the oil that surrounds the pole faces. This feature will not only enhance oil pump operation, but should increase bearing reliability by the removal of air. It will also improve long-term operation of the 2 micron mesh filter since large particles will be filtered out prior to entering.

R_{TOTAL} VS TIME

Figure A.3-127. Resistance Versus Time (With Air Bubbles but No Mesh)

(A)

(B)

Figure A.3-128. Magnetic Oil Pump Intake

Crankcase Oil Churning Losses

A test was performed on current nuclear system engine No. 2 to determine the crankcase loss due to oil churning. The data was taken to provide a base for estimating this loss on the Mark I prototype engine concepts. (Measurements of total input power were taken with and without a filled crankcase; the difference was divided by 0.8, and this result ascribed to churning.) Using Tellus 29 oil with a nearly 100 percent filled crankcase, the loss was 1.5 watts.

To determine what the loss would be with Tellus 21 oil, an assumption was made that 75 percent of the measured value was viscosity dependent, and the churning loss with Tellus 21 was calculated to be 0.83 watts.

The Mark I prototype rhombic drive mechanism concept under study has a stroke that is 80 percent of the current engine while it has the same rotational speed. Hence, the peak velocities of its members will be 80 percent of the current engine. In general, characteristic lengths and areas will be 80 percent of the current engine while clearances between parts will remain the same. Power loss for a rotating part is proportional to rotational speed (ω) squared, times a characteristic radius (r) to the fourth power. The power loss of a reciprocating part is proportional to velocity squared times area squared. In both these cases, the measured current engine loss would be scaled downward by the factor $(0.8)^4$ or 0.41. Therefore, a rough estimate of the churning losses in a Mark I prototype concept crankcase would be $0.41 (0.83) = 0.34$ watts, or in any case, less than one-half watt. This value was used in the mechanical loss estimates for this engine.

3.1.6 Bellows Piston Seal Development

Because of the very long unattended life required of a nuclear-powered artificial heart, there is a potential advantage in being able to seal the working space from the crankcase space of a Stirling engine with nearly perfect hermeticity. For this reason, a development aimed at providing such hermeticity through a metal bellows seal was carried out. The development was unsuccessful because limitations on bellows cycle life were not overcome.

A bellows tester was designed (see Figure A.3-129) and fabricated. Its basic specs were:

- Piston diameter = 3 cm
- Bellows free length = 2.5 cm
- Stroke = 0.8 cm
- Mean working pressure = 11 atm-ga
- Mean crankcase pressure = various 6.5 atm-ga to 9 atm-ga
- Bellows pressure differential = various 2 atm to 4.5 atm
- Speed = 600 rpm

Some problem studies in the design of the unit include:

- Shaft deflection in bearings with small clearances
- Bearing loads due to bellows spring rate
- Guide bushings performance
- Forces due to magnetic coupling

Relations were developed for bellows design which include Stirling engine working space dead volume and spring rate. The latter agrees well with an equation given by one of the bellows manufacturers. These relations are given in Table A.3-7. The various bellows parameters are indicated in Figure A.3-130.

The relations in Table A.3-7 were used to calculate a Stirling engine working space dead volume and approximate spring rates for proposed test bellows, as shown in Table A.3-8. The various configurations illustrated in Figure A.3-131 are itemized in Table A.3-8.

The bellows test unit is shown in Figure A.3-132. The apparatus was originally designed to be driven by an electric motor via a magnetic coupling. The use of such a coupling was to provide design data and operating experience with a configuration which is being considered as a potential engine/pump interface in future models. However, during discussions with the Indiana General Corp., the manufacturer of magnetic materials for couplings, it was found that when driving a pulsating load, such as that of the engine, magnetic couplings may be subject to "hunting." While investigating the "hunting" problem in more detail, it was decided to drive the apparatus directly by an integral electric motor.

Figure A.3-129. Bellows Tester Design

TABLE A.3-7
RELATIONS DEVELOPED FOR BELLWS DESIGN

VOLUME

$$V = \bar{V}_O + \pi \Delta R_O^2 \left[\left(\frac{R_2}{R_O} \right)^2 - \frac{1}{3} \left(1 + \frac{R_1}{R_O} + \left(\frac{R_1}{R_O} \right)^2 \right) \right]$$

INITIAL VOLUME

$$\bar{V}_O = \pi (\text{pitch}) R_O^2 \left[\left(\frac{R_2}{R_O} \right)^2 - \frac{1}{2} \left(1 + \left(\frac{R_1}{R_O} \right)^2 \right) \right]$$

SPRING RATE

$$K_{EQ} = \frac{\pi E \eta^2 \bar{R}_t^3}{24 N b^3}$$

$$\eta = \frac{b}{c}$$

where:

N = number of active convolutions

Δ = total bellows deflections

$\bar{R}$ = mean radius; i.e., $\frac{R_1 + R_O}{2}$

TABLE A.3-8
DEAD VOLUMES FOR VARIOUS CONFIGURATIONS

Configuration	V Max Outer	(cc) Inner	V Min Outer	(cc) Inner	Dead (cc) Space
A1	4.5	6.2	3.2	4.5	7.7
A2	4.5	4.6	3.2	3.5	6.7
B1	5.1	5.9	3.7	4.1	7.8
B2	5.1	5.0	3.7	3.9	7.6

Figure A.3-130. Bellows Parameters

Figure A.3-131. Bellows Configurations

Figure A.3-132. Bellows Seal Test Fixture

Seven bellows were tested. All were of electrodeposited nickel (Type A, Table A.3-8) fabricated by Seryometer, Inc. The test results are given in Table A.3-9.

TABLE A.3-9

Unit No.	Rated Differential Pressure Capability	Test Pressure Difference	Cycling Regime	Accumulated Cycles to Failure
1	65 psi	44 psi	100% compression	2.4×10^4
2	65 psi	44 psi	100% compression	0.6×10^4
3	65 psi	44 psi	85% compression 15% extension	2×10^5
4	44 psi	44 psi	85% compression 15% extension	4×10^5
5*	28 psi	28 psi	55% compression 45% extension	9×10^6 @ 0 psi 11×10^6 @ 12 psi 10×10^6 @ 28 psi
6	44 psi	44 psi	55% compression 45% extension	5×10^5
7	44 psi	44 psi	55% compression 45% extension	11×10^6
*Failure due to technician error				

As can be seen, none of the bellows assemblies lasted longer than 11×10^6 cycles at rated pressure (although in one instance the failure was due to human error in handling the test apparatus). It is felt that while quality defects in these bellows may have contributed to their premature failures, the basic cause was that their design was deficient. The manufacturer had been directed to meet the stroke, pressure and life requirements with as short a bellows as possible (minimum convolutions). This was necessary to minimize the large dead volume contributed by the bellows to the working space of the engine. (Indeed the dead volume of the actual test bellows was still too high to be acceptable for engine use.)

In parallel with the cycle-life tests, studies were made of alternate bellows designs and methods of filling the convolutions in an effort to reduce the dead volume to acceptable levels; however, this problem remained unsolved.

Because of the poor life performance of the test bellows combined with the unsolved problem of bellows dead volume, the development effort was discontinued.

3.1.7 "Y" Piston Ring

A novel geometry piston ring was evaluated on current nuclear system engine No. 2. The hope was that it would result in improved performance relative to the design specified Bal-seals. Figure A.3-133 is a photograph of this "Y" geometry ring installed on its piston. In this design, the legs of the "Y" are pressure activated and impose a minimal radial force on the cylinder wall. Pressure sealing capability is bi-directional. The test conducted on the engine showed that this seal preferentially pumped-up the working space so that the minimum working space pressure was nearly equal to the buffer space charge pressure. Consequently, the bearings were operating with significantly larger loads. The measured engine output power was less than that measured with the Bal-seal, while the compression ratio was unchanged.

Figure A.3-133. "Y" Piston Ring

To learn more about the pumping effect, the seal was installed in a reciprocating piston tester. The pressure waveform was monitored with a pressure transducer and oscilloscope. The photograph of the waveforms obtained is shown in Figures A.3-134 and A.3-135. In Figure A.3-134, a charge pressure below the piston of 45 psi is indicated by the straight line calibration at the bottom. It is evident how pumping has taken place toward the top of the cylinder. Figure A.3-135 shows the result of placing two 0.005 in. diameter holes in each lip of the "Y" cross-section. The downward leakage permitted by the orifices allows the D C flow due to pumping to flow back to the bottom of the piston. This has the desirable affect of shifting the peak pressure downward to the mean charge pressure. An undesirable consequence of these relatively large holes is that the pressure amplitude has significantly decreased.

Another possible technique for shifting the mean working space pressure downward (thus decreasing the bearing loads) would be to place a small slot in the cylinder wall near the top of the piston stroke. When the piston is at the top of its stroke, flow would bleed past the piston through the slot. Upon downward motion of the piston, the lip would again seal to the wall and the flow would cease. If the length and width of the slot were just sufficient to equalize the pumping effect, virtually no decrease in pressure amplitude would result. While this technique has been used in larger Stirling engines, its implementation in a miniature heart engine does not appear warranted.

Figure A.3-134. Piston Pressure Waveform

Figure A.3-135. Waveform With Bleed Holes

A.3.2 THERMAL INSULATION

Several component development tasks necessary to design, build, and insure long-term reliability of the thermal insulation for the thermal converter are reported in some detail in this Section A.3.2. Also included are the test results for the thermal protection insulation development work completed during the program.

A.3.2.1 High Temperature Vacuum Foil Insulation

The following component development tasks are presented in this section.

- Nickel Cup Fabrication and Development
- Guarded Heater Testing
- Helium Backfill Apparatus
- Coating and Plating Procedures
- Data System Program Listing

Nickel Cup Fabrication and Development

Characterization of Electroplated Nickel Used to Manufacture Insulation Foil Cups

This section contains information reported in full in Westinghouse Research Laboratory memo 76-106-MFOIL-MR, entitled, "Characterization of Electroformed Nickel Cups", by E. L. Kochka. The purpose of this investigation was to characterize the electroformed nickel cups prepared by Gar Electroforming Division to ascertain why they were susceptible to failure during chemical dissolution of the aluminum alloy mandrels.

Samples of electroformed nickel taken from defective cups in the first shipment from Gar Electroforming Division were examined by optical and scanning electron

microscopy. The outstanding features observed as possibly contributing to the structural instability of the cups were the broad nodules on the outer surface of the foil and a fast etching high cobalt phase at the inner surface. Ion microprobe analysis showed the high cobalt level to persist through the fast etching phase of the sample. Chemical attack of the high cobalt inner surface was detected on cups that pinholed during mandrel dissolution. Foil prepared by the Westinghouse Research Laboratory was fine grained, more resistant to etching, and free of cobalt. A sample of foil from a later shipment of cups from Gar was extremely fine-grained and indicative of the use of chemical additives in the plating bath. Although the above differences were noted, reasons given for the high failure rate of the Gar cups remain speculative.

Cups failed by cracking or disintegrating into several pieces during dissolution of the aluminum mandrel in concentrated caustic solution. Some cups were also severely pinholed. A scanning electron micrograph of a Gar electroformed nickel cup that cracked during mandrel dissolution is shown in Figure A.3-136. The front surface is nodular with deep crevices that provide paths for crack propagation. The back surface (the surface adjacent to the mandrel) shows only a replication of the machined mandrel surface. Figure A.3-137 shows a pinholed surface, and reveals a very unusual back surface that looks like deeply etched grain boundaries.

Metallographic cross sections of these foils exhibited a slightly discolored area at the inner surface. This area etched very rapidly compared to the rest of the sample to give the discontinuous, nodular phase seen in Figure A.3-138. Upon annealing, this phase readily diffused into the bulk of the sample (see Figure A.3-139). Traces of this phase were also noted on cross sections of the pinholed samples (as seen in Figure A.3-140), and it is evident that this phase was attacked perhaps upon prolonged exposure to the hot caustic solution used to dissolve the mandrel. Pinhole formation, however, appears to have developed from the outer surface as shown in Figure A.3-141 and suggests the possibility of entrapped soluble or reactive impurities or gas pockets in the coating.

(a) Outer Surface

(b) Surface Adjacent to Mandrel

Figure A.3-136. Scanning Electron Micrographs of an Electroformed Nickel Cup that Cracked During Mandrel Dissolution (560X) (Gar Cup No. MD21)

(a) Outer Surface

(b) Surface Adjacent to Mandrel

Figure A.3-137. Scanning Electron Micrographs of an Electroformed Nickel Cup Pinholed During Mandrel Dissolution (1560X) (Gar Cup No. MD41)

(a) 500X

(b) 1500X Oil Immersion

Figure A.3-138. Metallographic Cross Section of Gar Electroformed Nickel Foil with Apparent Second Phase at Surface Adjacent to Mandrel. (Etchant dil HCl-5% H_2O_2) (Gar Cup No. MD21)

(a) As-plated Samples

(b) Sample Heated in Vacuum, 4 hrs. at about 500°C

(c) Sample heated in Vacuum, 2 hrs. at about 1000°C

Figure A.3-139. Metallographic Cross-Sections of Gar Electroformed Nickel Foil Showing Effect of Vacuum Annealing. (500X, Etchant dil. HCl-5% H_2O_2) (Gar Cup No. MD42)

Figure A.3-140. Metallographic Cross-Section of Gar Electroformed Nickel Cup No. MD41 Showing Residual Traces of Apparent Second Phase (See Figure 137 for SEM of surfaces). (2000X, Etchant Electrolytic 5% H_2SO_4)

(a)

(b)

(c)

Figure A.3-141. Metallographic Cross-Sections of Gar Electroformed Nickel Cups No. MD41 Showing Possible Progression of Pinhole Formation from Outer to Inner Surface (500X Unetched)

At this point, an examination of samples of Westinghouse prepared foil and of samples from a later shipment from Gar was initiated to provide a comparison with the above material. The surfaces and cross sections of these foils are compared in Figure A.3-142 and A.3-143, where it can be seen that the Westinghouse prepared material is fine grained compared to the early Gar material, and also that the later Gar material is extremely fine grained and indicative of the possible use of an additive in their plating bath. The Westinghouse prepared foil was the most etch-resistant, and the early Gar material was the least etch-resistant (especially the inner phase).

In order to identify the inner phase and to further compare the various foils, samples were subjected to energy dispersive x-rays analysis (EDX) of the inner and outer surfaces. The elemental analyses in terms of peak height counts are shown in Table A.3-10. The most significant difference between the Westinghouse and Gar samples is in the cobalt analysis.

It is absent from the Research sample but present in the Gar samples and, furthermore, on the Gar samples that show a separate phase, it is significantly higher on the inner surface than it is on the outer surface. These data, however, are qualitative and good only to a first approximation of the relative amounts of the elements present.

Ion microprobe analysis was used to determine the distribution of cobalt through a cross section of the foil. A typical trace, shown in Figure A.3-144, confirmed that the high cobalt concentration persisted through the apparent second phase and then dropped abruptly to the lower concentration throughout the rest of the cross section.

Although it is not unusual for cobalt to be present in commercially electroformed nickel due to the presence of cobalt in the plating bath or in the nickel anodes, (it is not uncommon for commercial American nickel to contain up to 0.5 percent cobalt -- even pure electrolytic nickel may contain about 0.05 percent cobalt), the etching characteristic of the Gar foil is puzzling.

(a) SEM of Outer Surface, 560X

(b) Metallographic Cross Section, Etchant Electrolytic 5% H_2SO_4 (1000X)

Figure A.3-142. Appearance of Electroformed Nickel Foil Prepared at the Westinghouse Research Laboratories

(a) SEM of Outer Surface, 560X

(b) Metallographic Cross Section, Etchant dil. HCl-5% H₂O₂ (500X)

Figure A.3-143. Appearance of Gar Electroformed Nickel Foil From a Later Shipment of Cups (Gar Cup No. GA17C)

TABLE A.3-10

ENERGY DISPERSIVE X-RAY (EDX) ANALYSIS OF NICKEL FOIL SURFACES⁽¹⁾

Cup Identification	Westinghouse Research		Gar MD21		Gar MD41		Gar GA17C	
Surface ⁽²⁾	Inner	Outer	Inner	Outer	Inner	Outer	Inner	Outer
Element ⁽³⁾								
Aluminum	342	320	332	268	249	318	250	302
Sulfur	--	53	--	--	--	--	--	--
Chromium	--	--	41	--	--	--	--	--
Iron	--	117	--	--	53	--	--	22
Cobalt	--	--	175	65	133	24	127	179
Nickel	8934	8668	9072	9236	9169	9231	8702	8464

- (1) Data is semiquantitative. Number represent maximum peak heights as read from instrument counter.
- (2) Inner surface is surface adjacent to mandrel in plating operation.
- (3) Silicon was also found to be present in raised or indented areas on Gar MD21 with a maximum peak height value of 55.

Relative Concentration

(Nickel Trace Reduced by a Factor of Three)

Figure A.3-144. Typical Ion Microprobe Trace on a Cross-Section from Gar Electroformed Nickel Cup No. MD42

The cobalt content of the nickel deposit depends on the temperature, pH, and composition of the plating solution and on the current density during deposition. Of these variables, only current density has a large effect, and deposits made at low current density have a higher cobalt content than deposits made at higher current density. At a low pH, cobalt deposits simultaneously in cubic and hexagonal forms and the proportion of hexagonal close-packed cobalt increases as the pH increases. At bath temperatures above 70°C, however, only hexagonal close-packed cobalt is deposited. Cobalt deposits have a higher internal stress than nickel deposits, and high stress levels have been measured in cobalt-nickel alloy deposits. In preparing alloy deposits, the cobalt concentration in the deposits can be greater than in the bath because of an appreciably lower polarization of cobalt during deposition.*,** Thus, changing current density, pH, or temperature abruptly during the deposition to cause, in particular, a change in the mode of deposition from hexagonal close packed to cubic, could account for the etching differences in the foil. Detailed x-ray analysis could verify this assumption.

During the deposition of nickel, hydrogen is simultaneously discharged at the cathode. Most of the hydrogen ions are transformed into molecular, gaseous hydrogen, but some will codeposit or dissolve in the metal over the entire current density range. If nickel is deposited with a metal that forms a solid solution with it, e.g., cobalt, the increase in the number of lattice defects created relates to a large increase in hydrogen content.*** A high hydrogen content can also lead to the presence of hexagonally crystallized nickel along with the normal cubic nickel lattice and create strains. Hydrogen tends to concentrate preferentially at points where marked lattice strains exist. Recombination and expansion of molecular hydrogen during the exothermic caustic dissolution of the aluminum mandrel would undoubtedly rupture the thin foil.

* Brenner, A., Zentner, V., and Jennings, C.W., "Nickel and Its Alloys", Plating 39, 865-927 (1952).

** Brenner, A., Electrodeposition of Alloys, Vol. 1, Academic Press, 1963.

*** Raub, E. and Muller, K., Fundamentals of Metal Deposition, p. 136 ff. Elsevier Publishing Company, NY, 1967.

The incorporation of other impurities in the foil would also create strained areas or areas subject to local chemical attack. SEM/EDX analysis of samples always detected iron and/or silicon on what appeared to be blisters or bubbles. An accumulation of metal ion impurities and hydrogen could lead to the pin-holed coatings discussed above.

For purposes of comparison, the structure of the metal foils plated at the Westinghouse R&D Center were also examined.

Figure A.3-145 (a-f) is a group of scanning electron micrographs of the electroplated Ni cups. Figure A.3-145 (a and b) shows the as-electroplated structure of the outside and inside surfaces. A faint indication of the fiber structure noted on the inside surface, Figure A.3-145(b), can be seen in Figure A.3-145(a). The axis of the cup is indicated on the picture. The fiber structure follows the direction of the original machining marks of the aluminum mandrel. Figure A.3-145(c) shows the recrystallized grain structure after a 4-hour vacuum anneal at 870°C. The average grain size is 52 μm . Figure A.3-145 (d) shows the grain structure after a 25 hour anneal at 870°C. The average grain size is 56 μm . Figure A.3-145 (e) shows the foil after 96 hour anneal at 870°C with an average grain size of 53 μm . Figure A.3-145 (f) shows the grain size after a 980°C anneal for 100 hours. In this case, the grain size was 58 μm .

A set of 5 Ni-B electroless deposited foil cups was ordered from DuPont. Electroless deposition is an attractive process for forming cups. One feature is the possibility of utilizing reusable mandrels. Also, the Ni-B alloy is a wear-resistant alloy which is harder than electrodeposited nickel. This property could indicate the possibility of providing a stiffer cup with a higher modulus resulting in less tendency to sag and sinter together. The first step in the evaluation process was an attempt to dissolve the cups from the mandrels. Two attempts proved unsuccessful. The Ni-B cups cracked and broke apart. An Ni-B coated mandrel was sent to the Westinghouse Research Laboratories for etching using the etching procedure they employed for the electroplated nickel cups. This cup also cracked. At this point this evaluation was abandoned.

(a)
Outer Surface— As Deposited
(3600X)

Cup
Axis
→

(b)
Inner Surface— As Deposited
30 sec Etch - Aqua-Regia
(2700X)

Figure A.3-145. Scanning Electron Micrographs (SEM) Showing the Effect of Temperature on the Surface Microstructure of Electrodeposited Nickel Cups

(c)

Outer Surface

Annealed 4 hrs. at 870°C in 10^{-6} Torr vacuum
60 sec etch Aqua-Regia (100X)

(d)

Outer Surface

Annealed 25 hrs. at 870°C in 10^{-6} Torr vacuum
60 sec. etch Aqua-Regia (100X)

Figure A.3-145. (c-d) - Continued

(e)

Inner Surface

Anneal 96 hrs. at 870°C in 10^{-6} Torr vacuum
90 sec etch Aqua-Regia (100X)

(f)

Outer Surface

Annealed 100 hrs. at 980°C in 10^{-6} Torr vacuum
90 sec etch Aqua-Regia (100X)

Figure A.3-145. (e-f) - Continued

Spacer Technology

Plasma sprayed ZrO_2 spacers have been shown to be adherent on electroplated nickel. However, experimental data show the emissivity of the ZrO_2 sprayed cups is higher than the powder coated cups and, therefore, the heat loss from a plasma sprayed cup set is higher. However, the same tests indicate that the outer cup emissivity is extremely important in controlling the ultimate heat loss. This phenomena suggests that the plasma sprayed coating can be utilized on the interior hot cups where sintering is a problem and a very low emissivity coating such as gold plating and powdered ZrO_2 spacers could be utilized on the cooler exterior cups.

In an effort to optimize the performance of the standoffs with regard to both properties and ease of application, an extensive evaluation of the adherence and density of ZrO_2 spacers at various spacings and sizes was conducted. Various patterns of plasma sprayed ZrO_2 were applied using masking techniques. The resultant samples sprayed on 1 mil nickel foil were evaluated for adherence, density, and ability to prevent metal contact at 982°C (1800°F). The plasma spray process and metal preparation parameters that were utilized throughout the investigation are shown on Table A.3-11.

Random spraying, due to the elimination of both the mask and the surface preparation, would be a low cost method of applying minimum density standoffs. In an effort to determine if random spraying would produce acceptable results, a total of 12 samples at various process conditions were produced. It was concluded from the evaluation of these samples, detailed on Table A.3-12, that random spraying was not a feasible method of applying refractory standoffs to thin foil concentric cups due to poor adherence and excessive foil heating (foil wrinkling). In addition, it was felt that maintaining a uniform and reproducible coverage to achieve predictable thermal performance would be a serious problem.

TABLE A.3-11

PLASMA SPRAY AND METAL PREPARATION EQUIPMENT AND PROCESS PARAMETERS

A) PLASMA SPRAY

<u>Equipment</u>	<u>Materials</u>	<u>Process Parameters for ZrO₂</u>
Plasmadyne ⁽¹⁾ SP-1B-61M-R	ZrO ₂ -Metco ⁽²⁾ 201	Arc Gas: Argon-60 cfh at 50 psi
Hand Gun SG-1B	Al ₂ O ₃ -Plasmadyne 301-F	Powder Gas: Argon-7 cfh at 50 psi
Control SP-A-1000A		Powder Feed&Gear A at 30 setting
Power Supply PS-61M		Powder Injection: Internal
Power Feed 1000A		Arc Power: 21kW-60Amps, 35 Volts
Electrode SI-3-F&R		Gun-to-Substrate Dist. : Variable
		Traverse Speed: 12 inches/sec.

B) FOIL PREPARATION - 1 mil

Clean acetone; rinse methanol

Attach appropriate mask

Grit blast with Airbrasive⁽³⁾ Powder (No. 3) thru mask - S. S. White⁽³⁾ Airbrasive Unit F

Airblast to remove airbrasive

⁽¹⁾Plasmadyne; Santa Ana, California⁽²⁾Metco, Inc.; Westbury, Long Island, New York⁽³⁾S. S. White, Inc., New York, New York

TABLE A.3-12

RANDOM SPRAYED ZIRCONIA ON 1 MIL Ni FOIL

<u>No. of Passes</u>	<u>Gun-to- Substrate Dist. (In.)</u>	<u>Wt. ZrO₂ Applied (mg)</u>	<u>Thickness of ZrO₂ (mils)</u>	<u>Calculated Area Coverage (%)</u>	<u>Tape Adherence</u>	<u>Foil Appearance</u>	<u>Sintering Test 982°C-150psi</u>
1	3	56.9	0.5	13.8	Fair	Wrinkled	No Sintering
	4	9.1	0.4	2.8	None	Some Wrinkling	---
	5	3.8	0.3	1.5	None	Loose Powder	---
	6	1.6	0.2	1.0	None	Loose Powder	---
2	3	112.3	1.2	11.4	Fair	Wrinkled	No Sintering
	4	53.4	0.7	9.3	Poor	Some Wrinkling	---
	5	18.5	0.5	4.5	None	Loose Powder	---
	6	10.1	0.5	2.5	None	Loose Powder	---
3	3	103.8	0.9	14.0	Fair	Wrinkled	No Sintering
	4	33.9	0.5	8.2	Poor	Some Wrinkling	---
	5	29.9	0.5	7.3	None	Loose Powder	---
	6	8.8	0.4	2.7	None	Loose Powder	---

Square wire mesh stainless steel masks were next evaluated as an economical method of applying ZrO_2 standoffs of known size, spacing, and area coverage to thin foil concentric cups. A total of 10 masks (8 to 200 mesh, 2 to 39 mil openings, 34 to 50% open area, 3 to 56 mils between standoffs) and three process variations were used to produce 30 samples, as shown in Figure A.3-146. The evaluation of these samples, detailed on Table A.3-13, indicated that square wire mesh masks could be utilized to apply adherent and porous ZrO_2 standoffs if the high area coverage somewhat inherent in these masks can be tolerated. Sintering tests at 982°C indicated, as expected from the high coverage, that foil-to-foil sintering would not be a problem. The only problems that were encountered were a tendency for overspray due to the construction of the masks and a thermal deterioration of the fine wire mesh masks with the 120 and 200 mesh sizes. It was concluded (based on expected mask life, adherence, density, and minimum spacing) that 50 and 80 mesh masks were the optimum size. Figure A.3-147 shows the appearance of the ZrO_2 sprayed through these masks. Five cup systems using both ZrO_2 and Al_2O_3 were evaluated to determine the effect of material emittance. Al_2O_3 was chosen based on literature which indicated that Al_2O_3 had a lower emissivity in 3 to 7 micron range than ZrO_2 .

A total of 18 special masks having 10, 20, and 30 mil openings on staggered centers with area coverages of 5, 10, 20, 30, 40, and 50% were designed and produced by chemical milling of 5 mil stainless foil.

A considerable attempt was made to produce square holes since this would produce maximum mask integrity at the desired size and coverage. It was found impossible, however, to chemically etch 10 mil square holes in 5 mil stainless. It was, therefore, decided to produce round holes which have the geometries shown on Table A.3-14. Sintering tests on samples produced by spraying through these masks were used to select the optimum mask configuration needed to minimize coverage and maintain foil-to-foil separation.

A chemical milling technique was utilized to produce 3 mil thick stainless steel masks having 30 mil diameter openings on staggered centers arranged to

Figure A.3-146. Square Wire Mesh Stainless Steel Masks

TABLE A.3-13

SUMMARY OF WIRE MESH MASKS -
1 MIL Ni FOIL - FLAT - 2" x 12" - ZrO₂

Mesh Opening (mils)	No. of Passes	Calc. Area of Standoffs	Wt. Applied	Thickness	Calc. Spray Density	Adherence *			100 Hour Sintering Test at 982°C (1)	Remarks
						Tape	Abrasion	1t Bend		
8 93.7	1	49.5	0.7975	1.3	56.7	G	E	P	None	Mesh O. K. - No overspray apparent.
	2		1.3806	2.3	55.4	E	E	P	None	
	3		2.0188	3.3	56.5	G	P	P	None	
10 78.7	1	47.6	0.9217	1.7	52.1	G	E	G	None	Mesh O. K. - No overspray apparent.
	2		2.0078	3.0	64.3	E	E	G	None	
	3		3.1864	4.3	71.2	E	E	G	None	
12 66.1	1	46.5	1.0531	1.5	69.0	E	E	G	None	Mesh O. K. - No overspray apparent.
	2		2.1167	3.4	61.2	E	E	G	None	
	3		2.5223	3.9	63.6	E	E	G	None	
14 55.5	1	43.7	0.6095	1.6	40.0	E	E	G-E	None	Mesh O. K. - No overspray apparent.
	2		1.5720	2.2	74.8	E	E	G	None	
	3		1.7437	4.3	42.4	E	E	G	None	
18 39.4	1	40.1	0.5183	1.7	34.8	E	E	E	None	Mesh O. K. - Minimal, but visible overspray.
	2		1.2481	3.0	47.4	E	E	G	None	
	3		1.5897	4.1	44.2	E	E	G	None	
35 19.7	1	35.4	0.3389	1.2	36.5	E	E	E	None	Mesh O. K. Degree of overspray large at 2 and 3 passes. Little at 1. Pattern excellent. Adherence excellent.
	2		1.3568	2.3	76.2	E	E	E	None	
	3		1.8094	3.4	68.7	E	E	E	None	
50 16.7	1	33.6	0.6612	1.4	64.3	E+	E	E+	None	Mesh O. K. - Minimum, but visible overspray at 1. Con- siderable at 2 and 3.
	2		0.8518	1.5	77.3	E+	E	E+	None	
	3		1.1101	1.8	83.9	E+	E	E+	None	
80 7.0	1	32.9	0.4859	1.1	61.4	E	E	E	None	Mesh easily burned. Almost complete coverage due to overspray at 1, 2, and 3.
	2		1.0963	1.2	>100.0	E	E	E	None	
	3		Invalid	1.1	---	E	E	E	None	
120 4.9	1	33.3	0.1255	0.5	34.5	E	E	E	None	Mesh easily burned and clogged. Complete coverage.
	2		0.7265	1.3	77.0	E	E	E	None	
	3		1.3391	1.5	>100.0	E	E	G	None	
200 2.9	1	33.7	0.2215	0.7	42.9	P	P	G	None	Mesh easily burned. Coverage complete. Very powdery due to small openings.
	2		0.5843	1.2	>100.0	F	G	F	None	
	3		1.0746	1.5	>100.0	F	G	F	None	

*
G - Good
E - Excellent
F - Fair
P - Poor

(1) Sintering test-ceramic ring on stacked foils - load 150 psi.

A.3-223

Figure A.3-147. ZrO_2 Plasma Sprayed Through 50 and 80 Mesh Screens onto Nickel Foil Cups

TABLE A.3-14
 GEOMETRY OF CHEMICALLY MILLED MASKS -
 ROUND HOLES IN 5 MIL STAINLESS - STAGGERED CENTERS

<u>C. M. Mask No.</u>	<u>Size Opening (mils)</u>	<u>Open Area (%)</u>	<u>Holes per Sq. In.</u>	<u>Distance Between Holes (mils)</u>
1	30	5	70	90.0
2	30	10	142	53.1
3	30	20	283	30.0
4	30	36	425	18.3
5	30	40	566	12.3
6	30	50	708	7.5
7	20	5	159	60.0
8	20	10	318	35.4
9	20	20	637	20.0
10	20	30	955	12.2
11	20	40	1274	8.2
12	20	50	1592	5.0
13	10	5	637	30.0
14	10	10	1274	17.7
15	10	20	2548	10.0
16	10	30	3823	6.1
17	10	40	5096	4.1
18	10	50	6369	2.5

produce 5, 10, 20, 30, 40, and 50% open areas. These masks were used as described on Table A.3-15 in the application of ZrO_2 standoffs by the standard plasma spray technique at varying standoff densities to 1 mil thick nickel 200 foil. From the results also shown on Table A.3-15 and in Figure A.3-148 (a-f), it is apparent that from using these masks adherent and low density refractory standoffs to thin nickel foil with minimum overspray were produced. The masks protected the nickel foil from thermal degradation and were reusable. A simple sintering test (10 one square inch samples weighed with a 25 gm molybdenum cylinder resulting in 150 psi stress) at 982°C for 100 hours indicated that the 30 mil diameter standoffs prevented foil-to-foil contact sintering at 30% coverage (Mask No.4). This coverage for 30 mil diameter standoffs on staggered centers gives a calculated distance between standoffs of 18.3 mils and indicates that 20% coverage for 30 mil diameter standoffs (20.0 mil spacing and 10% coverage for 10 mil diameter standoffs (17.7 mil spacing) would also prevent foil-to-foil contact.

Plasma Spraying Cups Through A Wire Mesh Mask

ZrO_2 spacer material was applied by plasma spraying to the outer surfaces of a set of 5 nickel cups. To achieve an approximate 50% surface area coverage, a wire mesh screen was used as the plasma spraying mask. The as-plasma sprayed cups and ZrO_2 spacer patterns are shown in Figure A.3-149. Since plasma spraying had to be performed with the cups in place on the mandrel, it was not possible to accurately determine the weight of the applied ZrO_2 spacer material. Prior to plasma spraying, the surface of the nickel foil cups was grit-blasted using the same wire mesh screen as a mask that was to be used for plasma spraying. Adherence of the plasma-sprayed ZrO_2 was excellent.

TABLE A.3-15

CHEMICALLY-MILLED MASKS FOR REFRACTORY SPACER APPLICATION (30 MIL DIAMETER OPENINGS)

GENERAL

Mask Material - 3 mil SS - 4-1/2 in. x 12 in. mask area
Substrate Material - 1 mil nickel - 2 in. x 12 in. sprayed area
Spacer Material - Metco 201 - calcia stabilized ZrO₂

Application Parameters:

Air abrasive blast then mask with No. 3 powder
Spray Parameters: Plasmadyne (with argon electrodes)
Arc Gas: Argon 60 cfh at 50 psi
Feed Gas: Argon 7 cfh at 50 psi
Feed Setting: Gear A at 30 setting
Arc Power: 20 KW - 60 amps, 35 volts
Spray Distance: 4 inches
No. of Passes: 1, 2, or 3

EVALUATION PROCEDURE AND RESULTS

Density: Calculate from material density, area coverage, weight applied, and thickness
Adherence: Tape pull, scraping, and IT bend
Welding: 10-1 in. square foil tabs with 150 psi pressure 982°C-100 hr. $< 1 \times 10^{-5}$ Torr.

Mask No.	Dia. Opening (mils)	Area Cover. (%)	Spacer to Spacer Dist. (mils)	No. of Passes	Spacer Wt (gms)	Spacer Thick. (mils)	Calc. Spacer Density (%)	Adherence	Sintering After 100 Hr at 982°C
1	30	5	90.0	1	0.0183	0.8	20.9	excellent	yes - all 10
		5	90.0	2	0.0840	1.6	48.0	excellent	yes - all 10
		5	90.0	3	0.1110	1.7	59.8	excellent	yes - all 10
2	30	10	53.1	1	0.0646	0.7	42.2	excellent	yes - all 10
		10	53.1	2	0.1335	1.7	35.2	excellent	yes - all 10
		10	53.1	3	0.2241	1.7	60.3	excellent	yes 8 of 10
3	30	20	30.0	1	0.2041	1.8	25.2	excellent	yes 7 of 10
		20	30.0	2	0.4203	1.7	56.5	excellent	none
		20	30.0	3	0.4882	1.7	65.7	excellent	none
4	30	30	18.3	1	0.3699	1.7	33.2	excellent	none
		30	18.3	2	0.6712	1.7	60.2	excellent	none
		30	18.3	3	0.9666	2.3	64.1	excellent	none
5	30	40	12.3	1	0.4926	1.5	37.5	excellent	none
		40	12.3	2	0.9953	2.1	54.0	excellent	none
		40	12.3	3	1.1660	2.9	46.0	excellent	none
6	30	50	7.5	1	0.3066	1.2	23.4	excellent	none
		50	7.5	2	0.9677	2.6	34.0	excellent	none
		50	7.5	3	1.1340	2.6	39.9	excellent	none

(a)
Area Coverage - 5%

Spacer to Spacer Distance - 90 mils

(b)
Area Coverage 10%

Spacer to Spacer Distance - 53 mils

(c)
Area Coverage 20%

Spacer to Spacer Distance - 30 mils

Figure A.3-148. (a-c). 30 Mil Diameter Plasma Sprayed Dot Patterns
After 2 Passes (two coats of ZrO_2)

(d)
Area Coverage - 30%

Spacer to Spacer Distance - 18.3 mils

(e)
Area Coverage - 40%

Spacer to Spacer Distance - 12.3 mils

(f)
Area Coverage - 50%

Spacer to Spacer Distance - 7.5 mils

Figure A.3-148. (d-f). Continued

Figure A.3-149. Cups as Plasma Sprayed Through Wire Mesh Spacers

Guarded Heater Testing

A report entitled "An Investigation of High Performance Thermal Insulation Systems"* describes the guarded heater testing program and the test apparatus. During this earlier program, foil cup assemblies with up to 20 cups were evaluated and several system modifications of the test apparatus were made. Here, subsequent experimental results will be reported.

A 40 cup system was evaluated and life tested. In addition, the guarded heater test stand was used to evaluate the relative emittance of nickel foil cups with various surface treatments and characteristics.

Life Testing - 20 Cups Ni Package with -325 Mesh ZrO_2 Spacers

The thermal history of the 20 cup pack test is presented in Figure A.3-150. The equilibrium thermal performance of this package was evaluated for one month. The test was interrupted and the outer ten layers were removed to provide a ten pack for evaluation. This ten cup pack test was concluded after six days. The outer 10 cups were re-coated where necessary and replaced. The 20 cup pack extended life test was continued. For the next two weeks, thermal equilibrium data were obtained. The power was then set to maintain a 1600°F (870°C) temperature for 56 days. Power was periodically adjusted to maintain temperature to compensate for aging of the heater elements. All testing was done in a dynamic vacuum.

In Figure A.3-151, the heat loss as a function of temperature is plotted for the 20-cup pack. The plot shows that the initial heat loss values of the 20-cup pack before and after rebuild are quite reproducible. Also, the heat loss values for the 10-cup pack agree very well with the foil layer versus heat loss plots reported in the thermal insulation final WANL-M-FR-73-001, 1973. The reason for the change in thermal performance at 1600°F after the extended time at elevated temperature was determined when disassembled after four months of life testing.

*Svedberg, R.C., "An Investigation of High Performance Thermal Insulation Systems," COO-2239-001 (WANL-M-FR-74-004), August, 1974, Contract No. AT(11-1)-2239, Westinghouse Astronuclear Laboratory, Pittsburgh, Pennsylvania.

Figure A.3-150. Time-Temperature History 20 Cup Foil Life Test

Figure A.3-151. Effect of the Life Test Exposure on the Heat Loss of the 20 Cup Insulation Package as a Function of Temperature

Figure A.3-152. Guard Heater Hot Finger After Removal of the 20-Cup Pack Showing Where Sintering Occurred

Several points on the heater surface had sintered to the first cup (as shown in Figure A.3-152). The first and second cups were also sintered together at these points. In foils 3 through 6, there was some resistance when separating the cups. This sintering caused an increase in heat loss for the 56 cm² test system from 2.4 to 2.8 watts, an increase in heat loss of 0.4 watts or about 15% thermal performance degradation.

The heat loss resulting from this sintering between foil layers is equivalent to a loss of 3 to 4 foils or ~0.9 watts based on an 84 cm² hot finger for the thermal converter. An equivalent amount of sintering in a 40 or 60 layer package, i.e., ~4 foils being shorted would amount to only ~0.25 watts for a 40 layer package and ~0.17 watts for a 60 layer package. As the number of foil layers increase, the effect of sintering 3 or 4 layers becomes less significant. However, to prevent sintering, a new technique for applying ZrO₂ spacers was developed and used for the 40-cup pack test. For the 20-cup pack test, the nickel cups were coated with ZrO₂ on the outsides only, by brushing. The coverage was ~3.2 mg of ZrO₂ per cup. The dipping technique used for the 40 cup pack gave an average of 18 mg per cup and deposited ZrO₂ particles on both the inside and outside of the cups.

40-Cup Pack Assembly

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

An alcohol-ZrO₂ powder (-325 mesh) slurry was made by mixing 24 grams of ZrO₂ per 1000 ml of ethyl alcohol. This slurry was then placed into a glass tray. Periodically, the concentration of the slurry was checked by sampling the slurry and weighing the slurry before and after the alcohol has evaporated. Concentrations remained between 12-18 mg/ml. The difference between the as-mixed concentrations and that measured is due to the instant settling of the larger ZrO₂ particles. Figure A.3-153 shows a cup dipped into a slurry-filled tray. The cup was rolled in the slurry and allowed to drip dry for 15 to 30 seconds

Figure A.3-153. Dipping Technique Used to Coat Nickel Foil Cups for 40 Cup Pack

before drying in a hot air stream. The cup was handled only at the closed end. After drying, the cup was carefully picked up by the open end skirt and the top dipped and dried twice to cover the area used to handle the cups after the initial dipping. The 40 cups, after coating, are shown in Figure A.3-154. The cups were assembled by sequentially inserting the cups into the outer cup one at a time and weighing the package after each additional cup was added. The weight of the powdered spacer was then calculated by subtraction. Figure A.3-155 shows the 40 cup pack fully assembled.

The average weight of ZrO_2 on a cup was ~ 18 mg. Air films generated by air displacement when a cup was inserted, closed end down, were utilized to guide the cups into place. This minimized the probability of scratching or removing the spacer. Cups 1, 39 and 40 are shown in Figure A.3-156 after assembly and disassembly. These photographs indicate that no wiping away of the ZrO_2 spacer material occurred during assembly. (Cup 40 is uncoated on the exterior). The cups are positioned vertically onto the guarded test apparatus.

After 47 days of testing with no measurable thermal degradation, the 40-cup pack test in the vertical position was terminated as a result of a vacuum failure. The test results verify that spacer as applied by dipping in a slurry of ZrO_2 and ethyl alcohol did prevent sintering together of the cups. As a result, this coating procedure was used to coat the Stirling engine insulation systems.

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

Cup Nos.

1 - 6

14 - 20

27 - 35

7 - 13

21 - 26

36 - 40

Figure A.3-154. As-coated Nickel Foil Cups Prior to Assembly of the 40 Cup Pack Test

Figure A.3-155. Bottom View of 40 Cup Pack Test After Assembly

No. 1

No. 40

No. 39

Figure A.3-156. Several Cups after Disassembly of a Trial Assembly Showing that ZrO_2 was not Scratched or Removed during Assembly. (Cup 40 was not coated externally.)

Heat losses through the 40 cup system after the vacuum system malfunction showed a 0.3 W increase apparently due to the increased emissivity the foil cups resulting from the discoloration. Table A.3-16 lists the equilibrium heat loss values for the 40-cup pack tests before and after the vacuum system malfunction. These results are plotted in Figure A.3-157.

Also included in Table A.3-16 are the first two equilibrium data points for the 5 nickel cup pack test for evaluation of the plasma flame-sprayed ZrO_2 applied through the wire mesh. The heat loss through this 5-cup system is 2.3 watts greater than the average heat loss for a 5-cup pack. The results are also plotted in Figure A.3-157.

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

An additional 5 cups were plasma-sprayed with ZrO_2 through the wire mesh. These cups were designed to be added to the 5-cup set on test to provide a 10-cup pack test. Because there was insufficient clearance between cups in the first 5 cups on test, as evidenced by the tight fit, alternate cups from the 10-cup pack were assembled to form a 5-cup pack. This minimized the conductance contribution to the heat loss and, thus, enabled the effect of the ZrO_2 -sprayed mesh pattern on the effective emissivity of the nickel cups to be evaluated.

Guarded Heater Test System Calibration Procedures

Because of the difficulties inherent in thermal measurements at low heat loss levels, two standard insulation calibration samples were machined from Min-K-2002 to be used to evaluate the guarded test heaters and to provide criteria against which to evaluate individual heater performance. As guard heater designs were modified or heaters repaired, a set of heat loss measurements were made on the Min-K-2002 calibration packages. The heat loss through each Min-K-2002

TABLE A.3-16
EQUILIBRIUM THERMAL DATA FOR THE 40 NICKEL CUP PACK
AND THE 5 NICKEL CUP PACK*

<u>Insulation Configuration</u>	<u>Test System</u>	<u>Equilibrium Temperature (°C)</u>	<u>Heat Loss (watts)</u>
40 Cup Before Vacuum System Failure	No. 2 Vertical	780.7	0.86
		797.0	0.98
		861.3	1.52
40 After Vacuum System Failure	No. 2 Vertical	685.1	0.40
		774.4	0.98
		826.2	1.52
		835.0	1.52*
		865.5	1.83
5 Cup Plasma Sprayed	No. 1 Vertical	593.0	5.56
		822.0	10.25

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

Figure A.3-157. Total Heat Loss as a Function of Temperature for the Nickel Foil Cup Pack Tests

calibration package was identical for a given heater configuration. Figure A.3-158 shows the heat losses through MinK-2002 for the heater designs utilized in the program. These values are presented in tabular form in Table A.3-17. The first two heater designs have been previously analyzed*.

Table A.3-18 lists the changes in heater designs incorporated during the thermal insulation test program. The two major heater changes, which caused the largest difference in heat loss, were: Mark II change, where the guard heater was extended and Mark IV change where the stainless steel heater leveler was radially grooved. With the Mark IV configuration, the heat loss through the Min-K was about 2 watts greater than the calculated heat loss value at 870°C. The other modifications to the heaters resulted in a heat loss through the Min-K of about 0.5 watt below the calculated heat loss value. The spread in this thermal data implies an uncertainty in a 40 nickel cup insulation package of only ± 0.27 watt or 0.0048 watt/cm².

After the calibration procedure, the 40-cup pack and 5-cup pack were exchanged between the two guarded heater test stands to further evaluate the performance of the guarded heater apparatus. Table A.3-19 lists the equilibrium data recorded. The first three sets of data apply to the comparison between system No.1 and system No.2. The next five sets of data illustrate the effect of a spacer material on the outer cup on the heat loss of a 5-cup system.

Figure A.3-159 shows the heat loss as a function of temperature for the 40 cup pack and 5 cup ZrO₂ sprayed pack. This plot shows the difference in heat loss caused by the guarded heater test design. Figure A.3-160 is a plot of heat loss difference between system No. 1 and system No. 2 plotted as a function of the total measured heat loss for system No. 1. The data are shown for 780°C and include data points for Min-K 2002, the 40 cup pack and the 5 cup pack (cups 1, 2,3,4, and 5). Using this plot, the data collected for a given insulation for each guarded heater system design can be evaluated and compared directly.

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

Figure A.3-158. Total Heat Loss vs. Temperature-Min-K 2002 Calibration Test Runs

TABLE A.3-17
CALIBRATION DATA FROM GUARDED HEATER TEST

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

TABLE A.3-18
DESCRIPTION OF GUARDED INSULATION TEST SYSTEM HEATERS

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

TABLE A.3-19
EQUILIBRIUM THERMAL DATA FROM GUARDED HEATER TESTS

<u>5 Cup Set</u>	<u>Test System*</u>	<u>Equilibrium Temp. (°C)</u>	<u>Heat Loss (Watts)</u>
<u>Test Stand Evaluation:</u>			
1-2-3-4-5	No. 2 (MK IV)	736.0	8.10
		754.0	8.85
		792.0	10.75
1-2-3-4-5	No. 1 (MK III)	693.0	5.35
		753.6	7.016
		822.0	10.25
40 cup set	No. 1 (MK III)	667.0	0.00
		845.0	0.656
		865.0	0.822
		875.0	0.906
<u>Emissivity Evaluation:</u>			
2-4-6-8-10	No. 2 (MK IV)	641.0	4.42
		701.0	6.03
		750.0	7.60
		812.5	10.55
		847.2	11.95
2-4-6-8-9	No. 2 (MK IV)	667.3	6.04
		709.0	7.73
		791.0	11.80
2-4-6-8-10	No. 2 (MK IV)	694.0	6.03
		828.0	12.00
3-5-7-8-10	No. 2 (MK IV)	665.5	4.98
		771.5	8.52
		842.0	11.99
3-5-7-8-9	No. 2 (MK IV)	648.0	4.99
		727.0	8.563
		808.0	12.77

See Table A.3-18.

Figure A.3.2-159. Effect of Test System on Heat Loss of 40 and 5 Cup Packages

Figure A.3.2-160. Correction Curve to Rationalize Heat Losses Between the Mark III* and the IV* Heaters

*See Table A.3-18.

Figure A.3-161 shows the effect of interchanging cup No. 9 and cup No. 10 of the 5 cup pack test. Cup No. 9 was flame sprayed and cup No. 10 was as-plated with no spacer applied. The results of 5 separate runs show a significant decrease in heat loss attributable to the emissivity of the outer cup. As temperature increases, the difference in heat loss increases. At 750°C this is 1.8 watts difference, and at 800°C, 2.4 watts difference. These results explain why the heat loss for the 5 cup ZrO_2 spray coated system was higher than the 5 cup system with ZrO_2 (-325 mesh) powdered spacers. The emissivity of the cups was increased by the ZrO_2 coverage. This coverage is on about 50% of the cup surface. The studies also indicate that by further improving the emissivity of the cups, further decreases in heat losses can be achieved.

Cup Pack Investigation

The cup pack tests on the 5 cup 80 mesh plasma sprayed ZrO_2 cups were completed on test system No. 2 using a matte finished as-plated outer cup, and the 80 mesh sprayed outer cup. The 7 cup 30 mesh plasma sprayed pack was tested with the matte finished as-plated outer cup, the 30 mesh outer cup, and a gold plated outer cup. Table A.3-20 summarizes the data from test system No. 2. Table A.3-21 summarizes the data from test system No. 1.

Test System No. 1 - Figures A.3-162 and A.3-163 show the results of the evaluation of test system No. 1 by comparing heat loss data for Min-K-2002 and the 40 cup (with -325 mesh ZrO_2 powered spacers) insulation package.

The only difference now existing between test system No. 1 and test system No. 2 is the type of thermocouple being used. Test system No. 1 has a sheathed thermocouple and test system No. 2 has bare wire thermocouples in Al_2O_3 ceramic insulators. The sheathed thermocouples in test system No. 1 apparently provide a large heat leak from the guard into the measure heater system, thereby indicating a lower total heat loss at a given temperature. These data now permit a precise comparison between data taken on both test system No. 1 and test system No. 2. Subsequent to the calibration runs, the 5 cup pack with the 50 mesh plasma sprayed ZrO_2 pattern were evaluated on this system.

Figure A.3-161. Effect of the Emissivity of the Outer Cup on the Heat Loss of a 5 Cup Pack on System No. 2

TABLE A.3-20

TEST SYSTEM NO. 2 DATA SUMMARY

As-plated Ni Outer Cup 5 Cup 80 Mesh		80 Mesh Outer Cup Sprayed Outer Cup 5 Cup 80 Mesh		Gold Plated Outer Cup 7 Cup 30 Mesh		As-plated Ni Outer Cup 7 Cup 30 Mesh		30 Mesh Outer Cup Outer Cup 7 Cup 30 Mesh	
(°C)	Power, W	(°C)	Power, W	(°C)	Power, W	(°C)	Power, W	(°C)	Power, W
748	7.66	709	7.57	854	8.85	848	9.17	702	5.06
822.5	11.1	801	11.25	805	6.90	790	7.10	767	7.07
859.7	13.13	837	13.07	739	4.85	725	5.02	817.5	9.09

TABLE A.3-21

TEST SYSTEM NO. 1 MK IV DATA SUMMARY

Min-K-2002		
Temperature (°C)	Measured Power, W (53 cm ²)	Corrected Power, W (56 cm ²)
668	8.55	9.01
779	11.05	11.65
828.5	12.51	13.19
870.5	13.56	14.29
40 Cup Pack		
810	0.660	0.70
845	0.974	1.03
807	0.678	0.71
672	0	0

Figure A.3-162. Comparison of Min-K-2002 Calibration Insulation After Rebuilding Test System No. 1

Figure A.3-163. Comparison of 40 Cup Pack System After Rebuilding Test System No. 1

Test System No. 2 - Figures A.3-164 and A.3-165 show the results of 7 cup tests with outer cups showing different emittances and 5 cup test on 30 and 80 mesh spacer patterns with outer cups with different emittances. The technique for determining the emittances will be discussed in the next section. Figure A.3-164 shows that an as-plated Ni cup and a gold-plated nickel cup give improved heat loss performances when compared to an outer cup of 30 mesh plasma sprayed ZrO_2 . The emittances of the various cups are listed in Table A.3.2-22.

Figure A.3-165 shows that the heat losses of the 80 mesh spacer system are less than that of the 30 mesh spacer system. Again, the emittance results indicate that the 80 mesh system has a lower average emittance than the 30 mesh system. With the as-plated Ni outer cup, the heat loss for both systems is identical.

Emittance Measurements

The initial emittance measurements are listed in Table A.3-22. Measurements determined by two different techniques were utilized. Both techniques involved the use of an infra-red pyrometer* which was sensitive to infrared radiation between the wave lengths of 2.0 to 2.6 microns. The temperature of the outer cup was measured by sighting the pyrometer on graphite dots sprayed onto the outer cup as seen in Figure A.3-166. The first technique to measure emittance utilized the temperature of the outer cup, and solving the equation

$$\epsilon = \frac{q}{\sigma A (T_2^4 - T_1^4)}$$

where

q	=	equilibrium heat loss
A	=	area of the outer cup
T_1^4	=	temperature of the bell jar
T_2^4	=	temperature of the outer cup
σ	=	Steffon-Boltzman Constant ($5.67 \times 10^{-8} \text{ W/m}^2 \text{ -deg K}$)

* IRCON Model 300.

Figure A.3-164. Effects of Various Outer Cups on the Heat Losses of the 7 Cup Pack

Figure A.3-165. Effect of the Emissivity of the Outer Cup on the Heat Loss of a 5 Cup Pack on Test System No. 2

A.3-258

Figure A.3-166. Graphite Dot Sprayed Onto an Outer Cup for Emittance and Temperature Measurements

TABLE A.3.2-22.

PRELIMINARY EMITTANCE MEASUREMENTS ON
ELECTROPLATED OUTER CUP

Outer Surface Configuration	Avg. Outer Cup Temp. ($^{\circ}\text{K}$)	Emittance*	
		Pyrometer (2.0-2.6 μ)	Calculated (Total Hemispherical)
Matte Surface			
As-plated Ni Cup	590.8	0.28	0.19
5 Cup Set	624.5	0.28	0.27
System No. 1	640.8	0.28	0.28
80 Mesh ZrO_2	467.0	0.35	0.35
Pattern	508.0	0.35	0.37
5 Cup Set	536.6	0.41	0.43
System No. 1	543.4	0.41	0.40
Gold Plated Cup			
5 Cup Set	622.0	0.24	0.15
System No. 1			
Bright Surface	476.0	Below 0.2	0.05
40 Cup Pack	459.7	Extrapolated to	0.042
System No. 2		between 0.11-0.10	
50 Mesh ZrO_2	479.9	0.47	0.50 **
5 Cup Set	488.8	0.47	0.46 ***
System No. 2			

* Values good to ± 0.03 ** 5.5 $^{\circ}\text{C}$ out of equilibrium*** 2.8 $^{\circ}\text{C}$ out of equilibrium

Total hemispherical emittance for the electroplated nickel cups coated with the 0.030 inch diameter x 30 percent coverage plasma-sprayed ZrO_2 spacers falls within the range from 0.4 to 0.5. These values indicate that the ZrO_2 spacer is not transparent enough to permit the emittance of the nickel substrate to be predominant.

If the operating temperature of a unit is below $720^{\circ}C$, this increase in emittance should not be detrimental to the thermal performance, as demonstrated by the thermal performance of the IVBM No.2 insulation package which utilized this plasma-sprayed pattern.

Emittance measurements were performed on the electroplated nickel cups supplied by the commercial plating vendor, Gar Electroforming. Tests were conducted on both the as-plated surface and on plated surfaces which had been polished using $1\ \mu$ gamma alumina slurry. The as-plated and the as-polished surfaces had the same relative emittance (0.4). This can be attributed to the character of the as-plated deposit since the polishing did not significantly alter the surface topography but only affected surface highlights.

Helium Backfill System

A helium backfill system was designed to permit helium to be added to the multifoil insulation vacuum space to spoil the effectiveness of the insulation. This is necessary to prevent temperature excursions in the hot finger area of an isotope powered thermal converter. In the event the Stirling engine stalled while the engine was implanted, the energy normally used to produce work in the system could not be dissipated and would, thus, cause an increase in the temperature. The engine cylinder and components could mechanically fail under the temperature extreme and would not be reusable.

The helium backfill system was designed to pressurize the insulation vacuum to 120 ± 5 mm. At this pressure, the insulation would be spoiled and the excess heat would be conducted to the surface of the thermal converter.

Figure A.3-167 shows the layout drawing of the helium backfill system. Figure A.3-168 shows the component parts of the helium backfill system prior to assembly, while Figure A.3-169 shows the mechanism with the capillary tube in position prior to fitting the outer helium chamber over the mechanism. When the sealed capillary tube is severed, helium is admitted into the insulation volume. The interface between the scram gas and the insulation package consists of the welded end of the capillary tube. This does not compromise the pumpdown and outgassing characteristics of the insulation since the capillary tube does not add any significant volume to the system. The procedure for activating the helium backfill system is listed in Table A.3-23.

Procedures Developed for the Program

This section of the appendix lists the procedures developed during the program for:

- 1) Applying ZrO_2 Spacer Material to Nickel Foil Cups (Table A.3-24)
- 2) Etching Nickel Foil Cups from Aluminum Alloy Mandrels (Table A.3-25)
- 3) Component and Subassembly Leak-Testing Procedures (Table A.3-26)
- 4) Leak Testing Procedure Using UTI Mass Spectrometer (Table A.3-27)

Data Acquisition Program for Outgassing Evaluation

Table A.3-28 is a Fortran Program listing of the code developed to acquire spectral data using an H-P 2116B Minicomputer system to control a UTI 100 K quadrupole mass spectrometer.

The computer code enabled the computer to:

- control the scan function of the mass spectrometer
- determine spectral peaks and their locations
- record the maximum height of the spectral peaks

Figure A.3-167. Layout Drawing of the Helium Backfill System

Figure A.3-168. View of Helium Backfill System Component Parts

Figure A.3-169. Assembled Cutting Mechanism with Capillary Tube in Position

TABLE A.3.2-23

SCRAM ACTIVATION PROCEDURE

1. Tools required

- 2 Spanner Wrenches (as per Drawing 389D448)
- 1 pair diagonal cutting pliers
- 1 quarter-inch box end wrench

2. Activation Procedure

- A. Engage spanner wrenches in holes in scram body and scram nose cover.
- B. Turn nose cover counterclockwise to remove nose cover (direction looking axially at nose cover).
- C. Keep spanner wrench on scram body and use as a handle.
- D. Using cutting pliers, clip the security wire and remove from pin and bolts.
- E. Place the 1/4 inch box end wrench on the nut and rotate 4 full turns in the clockwise direction to cut the scram tubing.

TABLE A.3-24
PROCEDURE FOR APPLYING ZrO_2 SPACER MATERIAL
TO NICKEL FOIL CUPS

- 1) Weigh and record weight of cleaned nickel cup.
- 2) Dip each cup in ethyl alcohol- ZrO_2 slurry (0.48 g ZrO_2 (-325 mesh)/ 20 ml ethyl alcohol).
- 3) Prior to dipping each cup, stir the slurry solution to prevent settling of ZrO_2 particles.
- 4) Dry inside and outside surfaces of slurry coated cup using hot air gun.
- 5) Weigh and record weight of slurry coated cup.
- 6) Assemble cup onto appropriate hot finger. Where handling has removed ZrO_2 spacer material, use touch up brush and slurry solution to replace.

TABLE A.3-25

PROCEDURE FOR ETCHING NICKEL FOIL CUPS
FROM ALUMINUM ALLOY MANDRELS

- 1) Immerse nickel plated aluminum alloy mandrel in NaOH solution for 4-8 hours. 250 g NaOH/1000 ml H_2O , temperature of solution $\geq 150^\circ F$.
(Caution: Do not allow solution to cool on partially etched cups)
- 2) Rinse immediately in flowing tap water.
- 3) Immerse in 20% HNO_3 solution for 1-3 minutes or until nickel is bright.
- 4) Rinse immediately in flowing tap water followed by rinsing with either distilled or demineralized water.
- 5) Rinse with grain ethyl alcohol.
- 6) Hot air dry and store.

TABLE A.3-26

COMPONENT AND SUBASSEMBLY
LEAK TESTING PROCEDURES

- I. Calibrate Leak Detector with Calibrated Helium Leaks (AVS 2.1 - 1963)
- II. Individual Subassembly Tests
 - A. Mount subassembly and seal as required by each specific configuration
 - B. Do not use silicone vacuum grease on surfaces to be welded
 - C. Determine background leak rate
 - D. Spray outside of subassembly with He
 - E. If a leak is detected, reduce helium flow to pin point leak location.
- III. Final IVBM Component Test
 - A. Mount system in leak detector using pinchoff port
 - B. Spray system with He
 - C. If no leaks are found, proceed to mass spectrometer leak testing

TABLE A.3-27

LEAK TESTING PROCEDURE USING UTI MASS SPECTROMETER

- I. Calibrate mass spectrometer with calibrated helium leak
- II. Isolate mass spectrometer and IVBM insulation system from all pumps
 - A. Close 0.2 l/sec orifice valve, open 2.0 l/sec orifice valve
 - B. Spray outside of entire vacuum system with helium
 - C. Correct any leaks found in vacuum system
 - D. Spray outside of insulation packages with helium in following sequence
 1. Instrumentation feedthroughs
 2. Pinchoff port attachment
 3. Scram port attachment
 4. Final seal weld
 5. Outer container
 - E. Open 0.2 l/sec orifice
 - F. Close 2.0 l/sec orifice
 - G. Repeat Step D
 - H. Close mass spectrometer valve and valve in pumps
- III. Heat insulation system to $180 \pm 10^{\circ}\text{C}$ using internal heater and external heating tapes
- IV. Maintain $180 \pm 10^{\circ}\text{C}$ until H_2O (AMU*18) pressure begins to decrease, repeat Step II
- V. If leaks are found, cool system to room temperature, repeat Step II, and effect repairs.
- VI. Increase internal heater temperature to 50°C above maximum use temperature and continue bakeout until the system pressure decrease with time stabilizes.
- VII. Repeat Step II.

*Atomic Mass Unit

TABLE A.3-27 (Continued)

- VIII. If no leaks are found, isolate insulation system and mass spectrometer by valving out all pumps and in a static mode measure rate of total pressure rise with time and also measure the rate of helium pressure rise with time. Continue for 4 hours.
- IX. Repump the system to original pressure level measured at the end of Step VI. Bag entire insulation system in plastic bag and flood interior of bag with helium. Repeat Step VIII.
- X. If Step IX indicates a larger pressure increase with time than Step VIII, selected areas of the insulation system are bagged separately and Steps VIII and IX repeated.
- XI. If Step IX indicates no leak, remove plastic bag, isolate the insulation system for 100 hours, and measure the rate of residual helium accumulation and rate of pressure increase.
- XII. Repump system to pressure level measured in Step VI, bag in helium, return to static mode operation, and measure rate of total pressure and helium pressure increase.
- XIII. Calculate leak rates, compare values obtained in Steps XI and XII which will be used to assess static vacuum capability.

Calculated Pressure Rise	Estimated Static Vacuum Life (i. e. , time for system pressure to rise to 1×10^{-3} torr)
4×10^{-10} torr/sec	30 days
3×10^{-11} torr/sec	1 year
3×10^{-12} torr/sec	10 years

TABLE A.3-28

Mass Spectrometer Data Acquisition Program for UTI 100 C Mass Spectrometer
and Hewlett-Packard 2116B Mini-Computer

```

PROGRAM MASCN
DIMENSION PKHTS(50),GAIN(50)
DIMENSION DAT(35),PKHT(50),IAMU(50),RELI(50),PPRES (50),M4(50)
DIMENSION M5(50),                PKHTB(50)
10  WRITE(2,20)
20  FORMAT("                *****SCAN ROUTINE*****")///)
    WRITE(2,24)
24  FORMAT(/"DO YOU WANT ANY DATA NOW IN MEMORY? - 1=YES, 0=NO")
    READ(1,*) I9
30  WRITE(2,40)
40  FORMAT(/"NEED INSTRUCTIONS? - 1=YES, 0=NO")
    READ(1,*) K8
    IF (K8) 50,64,50
50  WRITE(2,51)
51  FORMAT(/"SET SWITCH REGISTER AS FOLLOWS."/
1    "SWITCH 0 - UP FOR MULTIPLIER GAIN RUN."/
2    "SWITCH 1 - UP TO PRINT SYSTEM ION CURRENT AND PEAK LOCATION"/
3    "SWITCH 2 - UP TO PRINT BACKGROUND ION CURRENT AND PEAK LOC."/
4    "SWITCH 3 - UP TO PRINT PARTIAL PRESSURE AND RELATIVE INTENSITY")
64  WRITE(2,65)
65  FORMAT(/"SET SWITCH REGISTER")
    PAUSE
    WRITE(2,66)
66  FORMAT (/ "ZERO CORRECTION? - 1=YES, 0=NO")
    READ(1,*) K4
    IF(I9) 415,60,415
60  WRITE(2,70)
70  FORMAT(/"NEW SCAN PARAMETERS? - 1=YES, 0=NO")
    READ(1,*) K7
    IF(K7) 80,121,80
80  WRITE(2,90)
90  FORMAT(/"ENTER TIME DELAY AND ENCODE VOLTMETER")
    READ(1,*) SEC,NCODE
    WRITE(2,100)
100  FORMAT (/ "ENTER SCAN START,READINGS/PEAK, SKIP INTERVAL")
    READ(1,*) NSTAR,K,ISPAC
    WRITE(2,120)
120  FORMAT (/ "ENTER AMU RANGE")
    READ(1,*) JB,JS
121  WRITE(2,122)
122  FORMAT (/ "NUMBER OF SCANS FOR AVERAGE?")
    READ(1,*) N7
    J4=JS-JB+1
    DO 129 J=1,J4
    PKHTB(J)=0
    PKHTS(J)=0
129  CONTINUE
130  WRITE(2,140)
140  FORMAT (/ "IF SYSTEM SCAN TYPE 1,-IF BACKGROUND TYPE 2")
    READ(1,*) K6
    GO TO(170,150),K6

```


TABLE A.3-28 (Continued)

```

150  WRITE(2,152)
152  FORMAT (/ "ENTER BKGD. MULT. GAIN-PRESSURE")
      READ(1,*) GMULB, BPRES
      N6=1
      GO TO 190
170  WRITE(2,180)
180  FORMAT (/ "ENTER SYST. MULT. GAIN-PRESSURE")
      READ(1,*) GMULS, SPRES
      N6=1
190  ISTAR=NSTAR
      J1=JB
      J=1
      I=0
200  I=I+1
      CALL RMPVT(ISTAR, IMASS)
      CALL DELAY(SEC)
      CALL READ(NCODE, DATA)
210  DAT(I)=DATA
      ISTAR=IMASS
      GO TO(200,220), I
220  GO TO (230,260), K6
230  IF(DAT(I)-DAT(I-1))250,240,240
240  IF( K-I)250,250,200
250  PKHTS(J)=(DAT(I-1)*GMULS)+PKHTS(J)
      M4(J)=ISTAR
      M5(J)=I
      GO TO 290
260  IF(DAT(I)-DAT(I-1))280,270,270
270  IF(K-I)280,280,200
280  PKHTB(J)=(DAT(I-1)*GMULB)+PKHTB(J)
290  ISTAR=ISTAR+ISPAC-2*(I-(K/2))
      IAMU(J)=J1
      IF(JS-J1)296,292,292
292  J1=J1+1
      J=J+1
      I=0
294  GO TO 200
296  N6=N6+1
311  IF(N7-N6) 312,190,190
312  A=FLOAT(N7)
      DO 314 J=1,J4
      PKHTB(J)=PKHTB(J)/A
      PKHTS(J)=PKHTS(J)/A
314  CONTINUE
318  IF(K4) 319,330,319
319  GO TO (325,320), K6
320  PKHBZ=0
      DO 321 J=4,8
      PKHBZ=PKHBZ+PKHTB(J)
321  CONTINUE

```

TABLE A.3-28 (Continued)

```

      PKBZC=PKHBZ/5.
      DO 324 J=1,J4
      PKHTB(J)=PKHTB(J)-PKBZC
324  CONTINUE
      GO TO 330
325  PKHSZ=0
      DO 326 J=4,8
      PKHSZ=PKHSZ+PKHTS(J)
326  CONTINUE
      PKSZC=PKHSZ/5.
      DO 327 J=1,J4
      PKHTS(J)=PKHTS(J)-PKSZC
327  CONTINUE
      WRITE(2,328)PKBZC,PKSZC
328  FORMAT("ZERO OFFSET BKGD.=" ,E12.5,6X,"ZERO OFFSET SYSTEM.=" ,
1E12.5)
330  GO TO (340,130),K6
340  IF(ISSW(0)) 415,350
350  DO 360 J=1,J4
      PKHT(J)=PKHTS(J)-PKHTB(J)
360  CONTINUE
370  DO 390 J=1,J4
      BMAX=ABS(PKHT(1))
      IF(BMAX-(ABS(PKHT(J)))) 380,380,390
380  PKMAX=PKHT(J)
390  CONTINUE
      SUMI=0
      DO 400 J=1,J4
      RELI(J)=((PKHT(J)/PKMAX)*100.)
      SUMI=SUMI+PKHT(J)
400  CONTINUE
      TPRES=SPRES-BPRES
      DO 410 J=1,J4
      PPRES(J)=((PKHT(J)/SUMI)*TPRES)
410  CONTINUE
415  CALL FORM
      WRITE(2,420)
420  FORMAT("          TEST SYSTEM              ON          DATE"//)
425  IF(ISSW(0)) 570,440
440  IF(ISSW(1)) 610,450
450  IF(ISSW(2)) 520,470
470  IF(ISSW(3)) 480,740
480  WRITE(2,430) TPRES
430  FORMAT("TOTAL PRESSURE=" ,E12.2)
      GO TO 650
520  WRITE(2,530)
530  FORMAT("//4X,"AMU",4X,"BACKGROUND ION CURRENT"//)
      DO 550 J=1,J4
      WRITE(2,540) IAMU(J),PKHTB(J)
540  FORMAT(3X,I4,9X,E12.6)
550  CONTINUE

```


TABLE A.3-28 (Continued)

```

CALL FORM
GO TO 470
570 WRITE(2,580)
580 FORMAT(/4X,"AMU",4X,"MULTIPLIER GAIN"/)
DO 600 J=1,J4
GAIN(J)=PKHTS(J)/PKHTB(J)
WRITE(2,590) IAMU(J),GAIN(J)
590 FORMAT( 4X,I4,6X,F12.4)
600 CONTINUE
GO TO 740
610 WRITE(2,620)
620 FORMAT(/4X,"AMU",4X,"SYSTEM ION CURRENT",4X,"PEAK LOCATION",4X,
1 "READINGS"/)
DO 640 J=1,J4
WRITE(2,630) IAMU(J),PKHTS(J),M4(J),M5(J)
630 FORMAT(3X,I4,7X,E12.6,11X,I4,8X,I2)
640 CONTINUE
CALL FORM
GO TO 450
650 WRITE(2,660)
660 FORMAT(/4X,"IAMU",3X,"ION-CURRENT",3X,"PARTIAL PRESSURE",3X,
1 "REL-INTEN"/)
DO 680 J=1,J4
WRITE(2,670) IAMU(J),PKHT(J),PPRES(J),RELI(J)
670 FORMAT(3X,I4,3X,E12.6,4X,E12.2,9X,F6.2)
680 CONTINUE
700 WRITE(2,710) ISTAR,IMASS
710 FORMAT(/"ISTAR=",I6,6X,"IMASS=",I6)
720 WRITE(2,730) SUMI
730 FORMAT(/"TOTAL ION CURRENT=", E12.2," AMPS")
740 PAUSE
CALL FORM
GO TO 10
END
END$

```


- subtract background spectra from acquire spectra
- record and signal-average data from multiple scans
- print results in a report type format

Static Vacuum Test Results

Six metal gasket sealed insulation packages were fabricated to evaluate the effects of the outgassing and leak detection procedures on the long term static vacuum performance, Table A.3-29 lists the six packages, the number of cups in each package, the getter used and the processing history. As of 3/31/78, no static vacuum failure has been observed. Static Vacuum Package No. 2 was not reheated after a power failure because of thermocouple failure. Times up to 1½ years of successful operation in the static vacuum mode have been logged.

A.3.2.2 Low Temperature Protection Insulation

The material recommended for insulating external IVBM blood pump and coolant line surfaces is closed-cell foam polyethylene. Merits of this material are: 1) low thermal conductivity (0.000565 watts/in-°F, low density and low permeability to moisture; 2) high chemical stability; and 3) high ductility. The latter characteristic of polyethylene insulation, which is somewhat unique for a closed-cell foam, permits sheet material to be tightly formed to irregular contours. For example, suitably thin material can be wrapped around the 1/8 inch OD IVBM coolant lines. Thermal conductance measurements were made on polyethylene insulated IVBM coolant tubing operating under simulated in-vivo conditions.

Test samples were prepared by hand wrapping 5/16 inch wide adhesive coated polyethylene insulation tape* on 12 inch lengths of 0.125 inch x 0.008 inch wall thickness IVBM coolant tubing. Three thicknesses of insulation wrap were studied: 1/16 inch, 1/8 inch, and 3/16 inch. A sealing coat of medical grade Silastic adhesive was applied over the insulation.

*Manufactured by Gaskatape, Inc., Elkhart, Indiana.

TABLE A.3-29
STATIC VACUUM PACKAGE PROCESSING HISTORIES TO (3/31/78)

Insulation Package	Bakeout Time (days)	Static Vacuum Time Prior to Pinchoff (days)	Pinchoff Date
No. 1 - 60 Cups No Getter	53	36	10/6/76 550 days
No. 2- 60 Cups Al-Zr Getter	20	43	12/22/76 280 days*
No. 3 - 60 Cups No Getter	24	23	2/2/77 422 days
No. 4 - 30 Cups No Getter	29	37	3/4/77 392 days
No. 5- 45 Cups Al-Zr Getter	20	31	4/26/77 339 days
No. 6 - 60 Cups 27 cups with plasma sprayed ZrO ₂ spacers Al-Zr Getter	20	31	4/27/77 338 days

*Discontinued 9/30/77 because of thermocouple failure.

A typical sample as readied for testing is shown in Figure A.3-170. Electrical leads were attached to the sample ends to provide connectors for self-resistance heating. Temperature measurement was provided by spot welding fine wire (40 gage) chromel-alumel thermocouples along the outside of the stainless steel tube.

A sketch summarizing the thermal conductance testing technique is presented in Figure A.3-171. Testing involved self-resistance heating a sample to establish a steady state higher temperature condition at the tube surface with the sample immersed in a large vessel of 100°F water. The measurement of heat loss to the 100°F water was selected to simulate in vivo coolant tube operation.

Measurements of current and voltage determined the rate of heat loss occurring through the insulation to the surroundings, and data taken at several sample temperatures established the relationship between heat loss and coolant line temperature. Note that the measurement of voltage used in the calculation of heat loss was taken across points falling within the zone of constant sample temperature where the transfer of heat was restricted to the radial direction only. Tube surface temperatures ranging from approximately 110°F to 150°F were investigated. A photograph showing the experimental apparatus arranged for testing in the aqueous surroundings is given in Figure A.3-172.

Heat loss measurements for 1/16, 1/8, and 3/16 inch thick layers of polyethylene insulation are given in Figure A.3-173. The insulation thickness was prepared as one, two, and three wraps of 1/16 inch thick butt joined tape. The maximum IVBM coolant tube temperature is estimated to be 120°F. The heat flux values indicated for coolant lines operating at this temperature and insulated with 1/16, 1/8, and 3/16 inch thicknesses of polyethylene are, respectively, 0.024, 0.012, and 0.006 watts/cm².

Figure A.3-170. Typical Polyethylene Insulation Test Sample
(a) Silastic sealed polyethylene wrap over IVBM coolant tube
(b) Thermocouples
(c) Electrical lead

Figure A.3-171. Thermal Conductance Testing of Polyethylene Foam Insulated IVBM Coolant Tubing under Simulated In Vivo Conditions.

Figure A.3-172. Apparatus Arrangement for Thermal Conductance Measurements of Insulated IVBM Coolant Tubing
 (a) DC power supply; (b) Ammeter; (c) Sample; (d) Water bath;
 (e) Hot Plate; (f) Thermocouple and voltage probe selector switch;
 (g) Digital Multimeter

Figure A.3-173. Measured Heat Losses Through Polyethylene Foam Insulated IVBM Coolant Tubing

A.3.3 INTERCONNECTION SHAFTS

As shown in Volume I of the report (Section 2), the thermal converter of the system is to be implanted in the abdomen and the blood pump element in the chest with a shaft between them. The concept was to use a flexible shaft of alternate "hand" lays of wound wire. This concept has been retained but not without considerable work on wire wound shafts plus investigations of alternate schemes such as universal joints, bellows joints, and thin solid shafts.

A.3.3.1 Wire Wound Shafts

The wire wound flexible shaft type used during the experimental practicability studies of the previous phase failed very quickly when used in vivo.

In Vivo Experience

To obtain in vivo experience on the blood pumping element and interconnecting shafting of the system, experiments were performed by the University of Utah, Division of Artificial Organs, in calves. The experiments were carried out using electric motors as the driver for the blood pump and shaft.

The four initial experiments were done with the motor external to the calf as indicated in Figure A.3-174. The wire wound flexible shaft exited the calf's body in the chest region. The first one was carried out using the suspension system of the dashed lines. For the other three, the motor was mounted on a saddle on the calf's back as indicated by the solid lines. The shafts used were 36 inches long with a core of 1/8 inch OD. The core material was high strength carbon steel (music wire). The shaft outside casing was a teflon tube of 1/4 inch OD for the first two experiments. For the third and fourth experiment a heavier casing was used. It consisted of a wound bronze spiral imbedded in a neoprene casing. The bronze spiral provided the bearing surface for the rotating core and also added an armouring effect against outside damage. The casing was of 3/8 inch OD.

In the first two experiments the wires in the core ruptured in approximately 12 hours. In the third experiment the animal was terminated after 69 hours for medical reasons. A lack of protection for the core in the first two experiments

Figure A.3-174. In Vivo Experiments with External Motor

undoubtedly caused the very quick failures. However, as the fourth experiment showed, when the shaft core failed in 25 hours, stronger casings were not a complete answer to wire wound shaft failures.

After the first four experiments, a decision was made to provide a better simulation of system implantation by implanting the drive motor in the abdomen. The flexible shaft between motor and blood pump penetrates the calf's diaphragm at the juncture of the diaphragm and sternum. All in vivo experiments after the first four used the implanted motor approach.

A motor arrangement for abdominal implantation is shown in Figure A.3-175. The motor is cooled by circulating water through a tube coiled around the motor and rejecting the heat to an external sink. Power, water, and instrumentation leads and a vacuum lead are bundled into a common stalk that penetrates through the calf's hide to the outside. The complete assembly is potted in Silastic. A vacuum is used when a grease lubricated air-filled drive mechanism is used. Under these conditions the ventricles seat better under a mild (5 inch Hg) vacuum than at ambient pressure. When the mechanism and shaft is used, as intended, oil filled, oil leakage into the motor is prevented by a carbon face seal on the motor output shaft. A magnet holds the sealing surfaces in contact. Experimentally this seal was found to give zero leakage for at least 500 hours and have a frictional torque of 0.10 to 0.15 oz-in. In Utah experiments No.5 through 14, the shaft cores were all of 1/8 inch diameter. The casings were basically of the neoprene imbedded bronze spiral type. Additional reinforcement was gradually added to the outside of the casing as the experiments progressed from No.5 through 14 until the casing assembly appeared much as that selected for the nuclear system and illustrated in Volume I, Section 3.4.1.

Experiments No. 5 and No. 14 were terminated in 35 hours and 86 hours, respectively, because of shaft core failures. All of the other experiments in this "batch" were terminated for other than shaft failure reasons. None of the experiments were of more than a few days duration.

Figure A.3.-175. Motor for Abdominal Implant

At Experiment No. 15 and thereafter, the core size was increased to 3/16 inch OD. No shaft failures were experienced until Experiment No. 47. This experiment was terminated 15 days after implantation by a shaft core failure. The maximum duration of any one experiment was 35 days during this set of experiments. Several others ran for three weeks or longer. No definitive reason for the failure of the shaft of Experiment No. 47 was established. Because the shaft was from the first of a newly fabricated batch, a manufacturing flaw is suspected. Several more experiments were run after Experiment 47 using shafts of earlier manufacture without shaft incidents.

Bench Test Procedures

Three different types of shaft endurance tests were carried out. The first two used a blood pump attached to a mock circulation as the loading device. The third type used hysteresis brakes with a cyclical programmable power supply as the load. The driver was an electric motor in all cases.

Blood Pump Load with Laboratory Motor Drive (BP w/LM)

The test setup for BP w/LM is illustrated in Figure A.3-176. In this setup an 80 oz-in rating Electrocraft E650 motor drives the flexible shaft and blood pump through a Lebow Associates 50 oz-in rating rotary torque sensor and bearing block. Connections between motor and torque sensor are by flexible couplings. Data was recorded on a Brush oscillograph with a 100 cy/sec response capability. The basic blood pump operating frequency is 2 Hz. However, since the right and left ventricles of the blood pump are pumped alternately, the basic torque cycle has a 4 Hz frequency.

Normal test conditions on the mock circulation were an aortic pressure of 120/80 mm Hg and a pulmonary artery pressure of 25/15 mm Hg. Water flow rates were in the range of 10 liter/min. Under these conditions the maximum torque on the flexible shaft was about 24 oz-in during left side systole, with a lesser peak of about 16 oz-in during right side systole with average torques in the range of 8 oz-in. The nominal shaft and motor speed were 15 Hz or 54 million shaft cycles per 1000 hours. There was little if any speed change over a blood pump cycle.

Figure A.3.-176. Flexible Shaft Test with Blood Pump Load/Laboratory Motor

Blood Pump Load with Implantable Motor Drive (BP w/IM)

The test setup for BP w/IM is illustrated by Figure A.3-177. In this setup the rotary torque sensor, the flexible couplings and bearing blocks are eliminated and a much smaller Electrocraft 550 motor is attached directly to the flexible shaft. The mock circulation conditions were the same as for the BP w/LM setup. The peak and average torques on the shaft, as deduced from motor current records, were about the same as for the BP w/LM setup within the $\pm 10\%$ accuracy of the measurements. It was observed that the instantaneous speed of the motor varied considerably from the average of 15 Hz over a blood pump cycle. Again the data was recorded on Brush recorder.

Hysteresis Brake Load with Laboratory Drive Motor (HB w/LM)

This was the type setup used, most commonly, for shaft endurance testing. Once a setup was debugged, it could be run continuously with several days between formal data logging and test conditions adjustment. Usually data was logged once a day on week days. An oscillograph trace of load was recorded on weekends.

The setup is illustrated in Figure A.3-178. In the figure the hysteresis brake is to the right. The flexible shaft is supported between bearing blocks. The bearing blocks are in turn connected to the hysteresis brake on the right and a Lebowen rotary torque sensor on the left by flexible couplings. The Electrocraft 650 drive motor, not shown in the figure, is connected to the torque sensor through another flexible coupling in the same way as previously illustrated in Figure A.3-176.

The 4 Hz cyclical loading of the hysteresis brake was provided by programmable power supply. This was normally a Hewlett Packard Model 6824A. Unlike the actual blood pump loading, the torque peaks simulating right and left systole were equal. The "normal" peak torque was set as 24 oz-in. Many of the endurance tests were run at 2.75 "normal" load blood pump calculated loading of the flexible shaft.

As for the other tests, the shaft design speed of 15 Hz was used..

Figure A.3-177. Flexible Shaft Test with Blood Pump Load/Implantable Motor

Figure A.3-178. Flexible Shaft Test with Hysteresis
Brake Load/Laboratory Motor

Three different size hysteresis brakes were used. They were:

<u>Model</u>	<u>Nominal Rating</u>
Magtroll HB 200-2	16 oz-in
Magtroll HB 300-2	32 oz-in
Magtroll HB 500-2	120 oz-in

The two smaller brakes could be used at over twice normal rating without difficulty.

One of the peculiarities of use of this type test setup was the amplification of the torque pulse supplied by the brake by resonance in the setup. This was particularly true using the 120 oz-in rating brake because the natural frequency of the setup with many of the test shafts was just above 2 Hz. As a result, if the torsional stiffness of the shaft changes during testing, the setup natural frequency and the torque amplification changes as illustrated by the following example:

Endurance tests were carried out on two flexible shafts left over from the previous practicability phase of the program. These shafts were the same as used in the first two in vivo experiments except they were 12 inch long instead of 36 inch long. The test setup was as previously described and the 120 oz-in rating hysteresis brakes were used as the loading device. As became standard practice, both shafts were curved to longitudinally to form a 90° angle with an 8-inch radius of curvature.

Shaft No. 1 - The motor speed control was set at 15 Hz (900 rpm) and the programmable power supply was set to produce 26 oz-in peak torque initially as illustrated in Figure A.3-179. In this figure the Cable 1 torque signal is that measured by the Lebowm rotary torque sensor for Shaft No. 1. Cable 2 refers to Shaft No. 2 which will be covered, shortly.

At approximately four hours, the torque recorded by the Lebowm sensor started to increase gradually, without change in the input variables, as shown in Figure A.3-180. At about 5 hours 27 minutes the torque peak increased, suddenly, to

Figure A.3-179. Shafts 1 and 2 Initial Valves

ACCUCART

Gould Inc., Instrument Systems Division

Cleveland, Ohio

Printed in U.S.A.

Figure A.3-180. Shaft 1 Torque Change

50 oz-in or more. About 9 minutes later the torque peak dropped to 32 oz-in peak with a gradual drop for a little over one hour when there was a sudden drop to 14 oz-in peak. The peak torque then remained nearly constant until the 18th hour when the test was terminated. Figure A.3-181 shows the broken outer strands of the flexible shaft.

Shaft No. 2 - This shaft was tested simultaneously with Shaft No. 1 using a duplicate test setup but the peak torque was adjusted to 21 oz-in. The shaft peak torque, with constant brake input, increased gradually over 27 hours 38 minutes to 44 oz-in when it suddenly increased to 52 oz-in. Thirty-four minutes later the torque suddenly decreased to a low level and remained constant to termination of the test at approximately 41 hours. Examination of the shaft after the test showed broken strands similar to that of Shaft No.1.

Because the 120 oz-in rating hysteresis brakes resulted in a test system with a resonant frequency very near to 2 Hz, most of the shaft testing was done using the 32 oz-in rating brake. The resulting system resonant frequency was approximately 5 Hz with most of the shafts tested.

Shaft Stiffness (Compliance Measurement)

Since a wire wound flexible shaft consists of layers of counterwound wires, the torsional stiffness of the shaft is not single valued. This is illustrated in Figure A.3-181. This is a measurement of the angular rotation of one end of a shaft relative to the other end under various applied torques. The shaft core diameter is 1/8 inch and is 8 inch long. As a convention, the winding direction is that rotational direction that will cause the outermost layer of wire to "wind up". This is the normal load carrying direction. As would be expected the shaft is stiffer in the "wind" direction than in the "unwind" direction. There is also a non-resisting torsional zone between the "wind" and "unwind" directions.

During shaft endurance testing, the practice was developed of discontinuing the tests once a week to measure the compliance of a shaft. This is reported as shaft stiffness for the "wind" direction in oz.-in./deg. This is the slope of

Blood Pump
End ○

A

B

Figure A.3-181. Shaft No. 1 After Test

the curve of Figure A.3-182 in the "wind" direction. Many shafts do not exhibit such ideal straight line torsional characteristics. Most also exhibit hysteresis so that a degree of judgment is involved in the selection of the shaft stiffness ratio.

Unlike the shaft used as an example in Figure A.3-181, the normal "winding" direction of the shafts used in this program was counterclockwise.

Bench Test Endurance Results

The results of the bench endurance tests using wire wound flexible shafts are summarized in Table A.3-30. The column headings across the top describe the parameter being summarized. All of the shaft cores tested consisted of layers of spiral wound wires, every other layer being wound opposite hand. In a few instances as noted: (1) a central mandrel core wire was absent (test Nos. 8, 9, and 12) and (2) the shaft core "windup" direction was clockwise instead of counterclockwise and the core was loaded so as to unwind its outermost layer (test Nos. 4 and 5).

The wire wound flexible shaft business is "trade secret" orientated and detail information on most of the cores tested is sketchy. An S.S. White 1/8 inch diameter core, using wires similar to those of cores of test Nos. 3 through 7, was examined in some depth. The results are shown in the following Table A.3-30:

TABLE A.3-30
1/8 INCH DIAMETER S.S. WHITE CORE

Wire Location	Diameter (inch)	Sample	U.T.S kpsi	0.2 Y.S kpsi
Mandrel	0.014	1	291.9	221.9
		2	289.4	186.3
1st and 2nd Winding	0.013	1	330.0	324.6
		2	336.2	306.9
3rd and 4th Winding	0.014	1	296.3	228.8
		2	298.8	243.8
		3	296.9	243.8
5th (outer) Winding	0.015	1	350.0	286.1
		2	344.4	291.7
		3	342.2	290.6

Figure A.3-182. As Received Shaft Compliance (Torsional Stiffness)

TABLE A.3-31

WIRE WOUND FLEXIBLE SHAFTS BENCH TEST RESULTS

Test No.	Shaft Manufacturer	Shaft Length* (inch)	Core Diameter (inch)	Core Wire Material	Casing Bearing Surface Material	Lubricant	Nominal Peak Torque Load (oz-in)	Test Time (hour)
1	S.S. White	12	1/8	Music Wire	Teflon	Grease	26	18
2	S.S. White	12	1/8	Music Wire	Teflon	Grease	21	41
3	S.S. White	8	1/8	Music Wire	Bronze Spiral	Grease	24	56
4	S.S. White	8	1/8	Music Wire	Bronze Spiral	Grease	24	75
5	S.S. White	6	1/8	Same as 3 & 4	Bronze Spiral	Grease	--	250
6	S.S. White	6	1/8	Same as 3 & 4	Bronze Spiral	Grease	--	72
7	S.S. White	6	1/8	Same as 3 & 4	Bronze Spiral	Oil	--	116
8	Stow Mfg.	6	3/16	Music Wire	Teflon	Grease	24 -- --	380 210 410
9	Stow Mfg.	6	3/16	Music Wire	Teflon	Grease	31 35 40 46 55 66	198 144 144 900 535 512
10	Stow Mfg.	6	3/16	Music Wire	Teflon	Grease	24	6,822
11	S.S. White	6	3/16	Music Wire	Bronze Spiral	Grease	24 66	2,088 13,842
12	Stow Mfg.	6	1/4	Music Wire	Teflon	Grease	24 66	1,344 2,022
13	S.S. White	6	1/4	Music Wire	Bronze Spiral	Grease	24 36 48	4,966 481 2,635
14	S.S. White	6	1/4	Music Wire	Bronze Spiral	Grease	24 66	184 6,689
15	S.S. White	6	3/16	Music Wire	Bronze Spiral	Grease	24 66	964 8,639
16	S.S. White	6	3/16	Music Wire	Bronze Spiral	Oil	66	9,207
17	S.S. White	6	3/16	Super Music Wire	Bronze Spiral	Grease	66	3,716
18	S.S. White	6	3/16	Elgiloy Wire	Bronze Spiral	Grease	66	519
19	S.S. White	6	3/16	Music Wire	Bronze Spiral	Grease	24 66	2,088 1,933
20	S.S. White	6	3/16	Elgiloy Wire	Bronze Spiral	Grease	66	2,531
21	S.S. White	6	3/16	Music Wire	Teflon	Grease	66	1,898
22	S.S. White	6	3/16	Elgiloy Wire	Bronze Spiral	Grease	66	1,417

*Casing Couplings Shoulder to Shoulder Distance

TABLE A.3-31. WIRE WOUND FLESIBLE SHAFTS BENCH TEST RESULTS (cont'd)

Test No.	Initial Stiffness (in-oz/deg.)	Final Stiffness (in-oz/deg.)	Test Setup Designation	Reason for Discontinuing Test	Remarks
1	--	--	HB w/LM	Core Failure	Peak torque load increased to 50 oz-in just before failure
2	--	--	HB w/LM	Core Failure	Peak torque load increased to 52 oz-in just before failure
3	0.48	--	HB w/LM	Core Failure	Shaft core windup direction was clockwise
4	0.48	--	HB w/LM	Core Failure	Shaft core windup direction was clockwise
5	1.10	--	BP w/LM	Programmatic	
6	1.10	--	BP w/LM	Programmatic	
7	--	--	BP w/IM	Programmatic	Shaft clamped such as to produce sharp kink
8	--	--	HB w/LM	Programmatic	No center core wire
	--	--	BP w/LM	Programmatic	
	--	--	BP w/IM	Programmatic	1000 hours total testing
9	--	--	HB w/LM	Programmatic	No center core wire
	--	--	HB w/LM	Programmatic	
	--	--	HB w/LM	Programmatic	
	--	--	HB w/LM	Programmatic	
	--	--	HB w/LM	Programmatic	
	--	--	HB w/LM	Core Failure	2433 hours total to failure
10	0.42 @ 60 hr.	0.13	HB w/LM	Programmatic	
11	0.82 @ 694 hrs	--	HB w/LM	Programmatic	
	--	0.18	HB w/LM	Programmatic	15930 hours total testing
12	1.25	--	HB w/LM	Programmatic	No center wire
	--	0.17	HB w/LM	Programmatic	3366 hours total testing
13	4.0	0.43	HB w/LM	Programmatic	
	0.43	0.43	HB w/LM	Programmatic	
	0.43	0.30	HB w/LM	Programmatic	8082 hours total testing
14	2.1	--	HB w/LM	Programmatic	
	--	0.20	HB w/LM	Core Failure	6,873 hours to failure
15	1.58	0.30	HB w/LM	Programmatic	Nuclear System Model
	0.30	0.20	HB w/LM	Programmatic	9,603 hours total testing
16	1.26	0.20	HB w/LM	Programmatic	Nuclear System Model
17	1.50	0.45	HB w/LM	Core Failure	Nuclear System Configuration
18	--	--	HB w/LM	Core Failure	Nuclear System Configuration
19	2.35	0.25	HB w/LM	Programmatic	Same as shaft of No. 11 test
	0.25	0.21	HB w/LM	Programmatic	4,021 hours total testing
20	2.9	1.10	HB w/LM	Core Failure	Nuclear System Configuration
21	4.3	0.17	HB w/LM	Programmatic	Nuclear System Configuration
22	2.1	0.41	HB w/LM	Programmatic	Nuclear System Configuration. Wire specially heat treated after core winding.

The carbon steel wire was coated with a brass colored material. Photomicrographs of sections of the wires showed a rather large number of inclusions of large size relative to the wire diameter. A metalurgical examination of the wire from a 3/16 inch diameter Stow core revealed similar inclusions but fewer of them.

The casing material against which the shaft cores rub is thought to be the only pertinent parameter as far as these results are concerned. The casings were fixed in one position throughout the test and were not subject to actual environmental flexing and blows.

As will be noted in the continuation of Table A.3-31, tests were discontinued for programmatic reasons as well as shaft core failure. The programmatic reason for discontinuing Test Nos. 5 through 8, 10, 12, and 13 was to use the equipment for other tests. Test Nos. 11, 15, 16, 19, 21, and 22 were discontinued in the course of program phase-out.

In all tests where the parameter was recorded, the torsional stiffness of the shafts decreased substantially. In general the stiffness reduced by $\frac{1}{2}$ to one order of magnitude in the first 500 to 1000 hours of test. Stiffness then remained essentially constant or decreased slowly. It is hypothesized that the initial relaxation is the result of yielding in the various wire layers to redistribute the stresses and later changes may represent wear.

The 1/8 diameter core shafts that failed rapidly and frequently in vivo did likewise in the bench tests. The 3/16 inch diameter cores made of special materials, such as Elgiloy and Super Music wire, had a high failure rate and do not look attractive for use.

Few 1/4 inch diameter cores were tested but since one of these failed in somewhat more than 6,000 hours (Test No.14), they are not a complete answer to shaft life for a nuclear heart system.

Of the 3/16 inch diameter cores tested, which were identical to or similar to those selected for use with the implantable nuclear system, one failed on the bench (Test No.9) and one, as has been mentioned previously, failed in vivo. On the bench there was one failure in eight tests (a Stow shaft). In vivo

there was one failure in approximately 33 tests (an S.S. White shaft). A 12% rate of 2,500 hours on the bench is not inconsistent with a 3% failure rate at 360 hours in vivo.

While both failures are thought to be the result of manufacturing deficiencies, perhaps these deficiencies are characteristic, at the present, for a run of shafts.

A.3.3.2 Other Torque Transmission Shaft Investigations

Other torque transmission shaft arrangements were investigated. One type uses several universal joints connected with short shafts rotating in ball bearings supported in a flexible casing. Another scheme consists of a solid or hollow core made of materials with a high ratio of endurance strength to modulus of elasticity rotating in a flexible casing with a limited minimum bend radius. A third has a core consisting of short rigid shafts connected with flexures or bellows rotating in ball bearings supported in a flexible casing.

Universal Joint Shaft

A universal joint flexible shaft assembly was designed and manufactured. Figure A.3-183 shows the rotating assembly and the casing parts prior to their final assembly. The rotating assembly consists of a series of three double universal joints (Hookes couplings) supported in ball bearings from short cylindrical housing members which are, in turn, supported in a flexible casing. The universal joints employ a cruciform arrangement of six (6) contiguous balls which are joined together by spot welding to obtain a zero backlash assembly. The universal joints were manufactured by General Thermodynamics Corporation of Wilmington, Mass. These joints were selected for the initial assembly because of their zero backlash, their relatively low price and the willingness of the manufacturer to promise early delivery of the special shaft configurations required for the assembly. Some of the special features, including the square section end shafts, machined integrally with the end universal joints, and the ball bearing retainer ring grooves can be seen in Figure A.3-183. The dark material coating the universal joints is a molybdenum disulphide based grease formulated by the Westinghouse Research Laboratories.

Figure A.3-183. Universal Joint Flexible Shaft Assembly

During fabrication of the universal joint flexible shaft, three vendor samples of universal joints of the type to be used in this assembly were run on the bench under simulated blood pump torque loading and at a joint angle of 10° . The 10° joint angle is produced in the assembly when the flexible outer casing approximates a 5 inch radius bend. Figure A.3-184 shows a photograph of the test setup.

The tests were run using a hysteresis brake to provide a 4 Hz cyclical load with a peak torque of 24 oz-in. The first unit was tested at 900 rpm. It developed excessive backlash and fretting corrosion in the joint at the brake end of the unit after a total running time of 441 hours.

The most notable feature of this test was the apparent complete lack of wear in the joint at the other end of the unit which suggested that the deterioration experienced at the brake end resulted from a lubrication breakdown rather than from a basic lack of load capacity inherent in the design of the joint. However, a second universal joint, tested at a joint angle of $8\frac{1}{2}^\circ$ and apparently well lubricated initially, failed after a running time of only 119 hours 35 minutes. A third universal joint, also tested at a joint angle of $8\frac{1}{2}^\circ$ and lubricated with a special molybdenum disulphide based grease prepared by the Westinghouse Research Laboratories developed excessive backlash after a run time of 300 hours. This latter unit was relubricated at intervals of 25-50 hours during the test, the relubrication tending to reduce the hash which had developed on the torque-meter strip chart record during the previous run period. Figure A.3-185 illustrates the test setup used with the second and third joints. The PIC couplings used in the first test were replaced by plain sleeve couplings to reduce random motion at the joints.

Figure A.3-186 shows the completely assembled universal joint flexible shaft mounted on the test bench. Initially, the shaft assembly was run with a bend radius of 6 in and a speed of 900 rpm for periods of 2 hours at zero brake load, 1 hour at $1/4$ load and $1\frac{1}{2}$ hours at each of $1/2$, $3/4$, and full load, full load being a 24 in-oz maximum torque simulated blood pump cyclic loading applied by a hysteresis brake. The shaft assembly was somewhat noisy in operation, probably due to large radial clearance between the bearing housing members and the flexible outer casing. The large clearance results from the use of bearing

Figure A.3-184. Universal Joint Test Setup

Figure A.3-185. Test Setup for Universal Joints No. 2 and 3

Figure A.3-186. Universal Joint Flexible Shaft Test Setup

housings of outer diameters capable of insertion into a flexible casing having a bore at the low end of the tolerance range. The particular flexible casing used in the assembly had a bore on the high end of the tolerance range which is rather broad (0.020 diametral) in the commercial Anaconda type E.F. 3/8 in. metal hose material used in the initial assembly. This clearance could be substantially reduced in future assemblies by machining the bearing housing outer diameters to suit the measured bore of the particular casing material used in the assembly. The shaft assembly was disassembled after a total run time of 7- $\frac{1}{2}$ hours and examined for evidence of rubbing, fretting or other operational problems. No evidence of any such problems was discovered.

After this inspection the universal joint shaft was reassembled and run with its casing filled with sperm oil. Test conditions of 6 in bend radius, 900 rpm, and simulated normal blood pump loading applied by a hysteresis brake were maintained. The test was shut down after 247 hours. Torsional backlash increased substantially during the test.

A measurement of the tare torque of the universal joint flexible shaft assembly was made. A value of approximately 0.1 in-oz was obtained by measuring the drag of the complete test system, consisting of flexible shaft supported between two sealed bearing blocks, and then subtracting the drag torques of the two bearing blocks measured individually. Since the drag of the complete system was approximately 1.25 in-oz and the drags of the two bearing blocks were 0.65 and 0.55 in-oz, respectively, the net tare torque of the flexible shaft was obtained by subtracting a value for bearing block drag almost as large as the value for the complete system. Thus, the measurement must be considered to be approximate only, especially in view of the relatively large amount of hash ($\pm$ 0.1 in-oz approx.) present on the strip chart recording of the torque.

As a result of the limited life experienced with the General Thermodynamics units, universal joints of a different design were procured from the Curtis Universal Joint Company, Inc., Springfield, Mass. These joints are of more conventional design using block and pin type cruciform bearings rather than contiguous balls and are available in both 18-8 stainless steel and alloy steel. A stainless steel model was tested in order to evaluate the capability of this design on a fairly comparative basis with the General Thermodynamic joints at an angle of

8-1/2° under the 24 in-oz maximum, cyclically applied, simulated blood pump torque loading using a hysteresis brake. In this test, to supply the working parts of the joint with lubricant, wicks housed in the hollow end shafts of the joint were used. A small portion of each wick is visible in Figure A.3-187, which shows the test setup. A 500 SSU oil doped with 7% Vanderbilt 622 E.P. additive suitable for application by wick was recommended by the Westinghouse Research Laboratories for this purpose.

The Curtis stainless steel universal joint test was terminated after a total running time of 1030 hours with normal simulated blood pump loading. Very little degradation, in the form of increased backlash and torque meter hash, was evident.

Metal Bellows Couplings Shaft

A flexible shaft arrangement which employs an assembly of metal bellows couplings and short solid shaft segments supported in bearings inside a flexible outer casing was designed.

Two Servometer SC-7 bellows couplings of the type to be used in the flexible shaft assembly were subjected to endurance testing under simulated blood pump cyclic torque loading applied by a hysteresis brake. Both couplings were tested at an angle of 10° and at 900 rpm. The first coupling failed after 5 hours and 47 minutes and is shown in Figure A.3-188. This failure may have been precipitated by damage to the coupling during previous handling, by a neglect to provide adequate hold-down for the brake assembly or by a combination of both. The second coupling was carefully handled and more attention was paid to the alignment and security of the brake. After running for a period of 917 hours at 24 in-oz peak torque, the loading was increased to a peak value of 30 in-oz. At a total run time of 1,255 hours, the load was again increased to a peak value of 35 in-oz. The second bellows coupling failed after a total run time of 1,366 hours.

A third bellows coupling was put on test. Its test history was as follows:

<u>Loading</u>	<u>Test Hours</u>	<u>Peak Torque</u>
Normal	186	24 oz-in
25 Percent Overload	175	30 oz-in
50 Percent Overload	2747	36 oz-in

Figure A.3-187. Curtis Universal Joint Test Setup with Wick Lubrication

Figure A.3-188. Servometer SC-7 Bellows Coupling No. 1 After Failure

The test was discontinued after the total of 3,108 hours without failure was accumulated.

A complete bellows coupling shaft was fabricated. It consisted of a flexible outer casing and a core built up from bellows couplings and short connecting shafts supported in ball bearings. It was put on life test with simulated blood pump loading supplied by a hysteresis brake in the same manner as for wire wound shafts. Operation was at 15 Hz shaft speed (900 rpm) with a 4 Hz cyclical loading.

The test history of this shaft prior to failure is as follows:

<u>Peak Pulsatile Torque Loading</u>	<u>Test Hours</u>
12 oz-in	720
18 oz-in	870
24 oz-in	39

A photograph of the shaft core and casing taken after the bellows failure is shown in Figure A.3-189.

Tubular Nylon Shaft

A section of Nylon 12 tubing was tested as a shaft in one of the hysteresis brake/laboratory motor test setups used for wire wound shaft testing. The tubing had a 1/4 inch OD x 3/16 inch ID. The test conditions were 24 oz-in pulsatile 4 Hz load, 15 Hz (900 rpm) shaft speed and a 4 inch 90° bend radius [rather than the normal 8 inch 90° bend radius]

The test was stopped after 611 hours. No visible deterioration of the tubing was observed. However, the shaft felt quite warm to touch while running, indicating a significant power loss.

Solid Titanium Shaft

At solid 0.069 inch diameter titanium flexible shaft core was machined from Ti-6Al-4V. The shaft was shot peened after machining to avoid surface conditions that could cause premature failure. The shaft was operated 185 hours (10^7 stress cycles)

Figure A.3-189. Bellows Coupling Shaft after Bellows Failure

with a 10 in bend radius and cyclic peak torque loading of 22 oz-in provided by a hysteresis brake, as shown in Figure A.3-190. A casing was not used in this test so that the dynamic behavior of the core could be observed. Although stress calculations predicted fatigue failure of the core during the test, no evidence of damage was observed when the test was terminated. The core was then operated 185 additional hours with an overstress condition obtained by reducing the bend radius to 8 in. No evidence of damage was observed after this additional testing.

The following table summarizes the additional testing of the solid titanium flexible shaft without a casing performed with loading provided by a hysteresis brake.

Bending Overload (percent)	Torque Overload (percent)	Test Duration (hours)
11	25	185
0	50	185
11	50	185
25	50	185

Since no change in the shaft appearance was observed after this testing, the shaft was installed in a special casing with a teflon liner and operated with a 25 percent bending overload and a 50 percent torsion overload to determine any effects due to operation in the casing. The shaft failed after 83 hours of operation. A microscopic examination of the failure site (Figure A.3-191) indicated that the failure was caused by a groove cut into the shaft by a metal chip or a particle of abrasive foreign matter.

A second 0.069 inch diameter solid flexible shaft made of annealed titanium alloy Ti-6Al-4V was tested without a casing under 25 percent bending and 50 percent torsion overload conditions. The shaft was operated through 10^7 bending stress cycles, the approximate endurance limit for the shaft material. Since no evidence of damage was observed after this test, the shaft was then installed in a casing and operated with no bending or torsion loading to determine whether

Figure A.3-190. Titanium Shaft Test Setup

(a)

(b)

Figure A.3-191. Failure Site of Titanium Flexible Shaft

- (a) A reassembly of the fracture showing circumferential scratches at and adjacent to the failure site.
- (b) A scratch rim and fatigue and torsion failed regions of the fracture surface.

operation in a casing would cause shaft surface damage. A photograph of the casing and shaft is shown in Figure A.3-192. After 185 hours of operation, the shaft was again examined and found to be free of surface damage.

Then the shaft was operated 185 hours (10^7 bending cycles) with an 8 in. bend radius but without torque loading to determine whether operation of the shaft in a casing would cause shaft surface damage. Since no damage of the shaft surface was observed after this test, the shaft was operated 185 additional hours with an 8 inch bend radius and a torque loading 50 percent above normal. An examination of the shaft and casing after this test again revealed no shaft surface damage.

Although the examination of the titanium flexible shaft 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.

No further testing of the titanium flexible shafts was carried out.

Figure A.3-192., Titanium Flexible Shaft and Casing

A.3.4 BLOOD PUMP MECHANISM

In support of the design, fabrication, and testing of the nuclear system, research and development efforts were carried out on the blood pump drive mechanism. These efforts were directed primarily at demonstrating or improving life and reliability, reducing weight through better material selection and material protection and improving mechanism biological compatibility and responsiveness.

A.3.4.1 Oils and Ventricle Material Compatibility

Immersion tests were performed to evaluate the compatibility of both the normally used ventricle Silastic material, MDX 4-4514 and a more oil resistant fluoro-silicone compound, MDX 4-4026, with candidate oils for the drive mechanism cavity. Tests of 10, 30, 38 and 90-day durations were made in accordance with the ASTM D-471-72 test procedure for determining the change in properties of elastomeric vulcanizates resulting from immersion in fluids. The oil temperature in the tests was 120°F, and the specimen size was 2 in x 1 in x 0.08 in thick. The results of volumetric swell measurements made after each test are shown in Table A.3-32. These results indicate that the present ventricle material did not undergo a significant dimensional change when immersed in sperm oil, but did experience an objectionable change when immersed in the other oils. The fluoro-silicone compound did not undergo significant dimensional change in any of the oils tested.

An investigation of the diffusion of sperm oil through ventricle Silastic and Vitrea 21 oil through Avcothane and Lycra was carried out to determine whether an objectionable amount of blood pump lubricant will diffuse through the blood pump ventricle diaphragms into the circulatory system. The test setup for this investigation, shown in Figure A.3-193, consists of glass permeability jars, a stand and holders. The jars have a capacity of 70 ml (4.27 cu-in.) with an opening of 31.8 mm (1.25-in.) diameter and an inside diameter of 33 mm (1.30-in.). The jar cap has screw threads and forms a leak-proof seal without subjecting the ventricle diaphragm specimen to excessive torsional stress. Each jar was filled with about 50 ml of oil. The diffusion rate of the oils through the diaphragm specimens was measured, periodically, by wiping the outside surface of the

TABLE A.3-32

SWELL OF VENTRICLE ELASTOMERS IN MECHANISM OILS

Elastomer	Oil	% Swell for Various Days				Comments
		10 days	30 days	38 days	90 days	
MDX 4-4514	Sperm	3.8	5.4	3.3	3.0	No dimensional change
MDX 4-4514	Vitrea 21	15.0	11.6	14.5	12.8	Material widened 1/32 in. Color changed to light yellow
MDX 4-4514	Santotrac 30	37.6	37.9	45.0	41.2	---
MDX 4-4514	Transformer 55 plus 20% additive	----	33.8	34.4	31.4	Material lengthened 1/16 in. and yellowed
MDX 4-4026	Sperm	2.0	2.0	0.5	0.7	No change
MDX 4-4026	Vitrea 21	1.6	1.6	1.0	1.1	Material yellowed
MDX 4-4026	Transformer 55 plus 20% additive	2.0	3.4	2.9	3.0	No change
MDX 4-4026	Santotrac 30	----	----	5.8	5.5	---
MDX 4-4026	Mobil TK	----	----	0.1	0.2	No change

Figure A.3-193. Ventricle Specimen Oil Diffusion Test Setup

specimen with a soft tissue and determining the difference in the weight of the tissue before and after wiping. Any substance picked up by the tissue was examined to determine whether it was in fact oil that diffused through the diaphragm specimen. These were ambient temperature measurements. The total oil diffusion rates through both right and left blood pump ventricle diaphragms projected from the test results are as follows:

<u>Ventricle Material</u>	<u>Oil Diffusion Rate (grams/month)</u>	
	<u>Sperm Oil</u>	<u>Vitrea 21 Oil</u>
Silastic - 0.032 in. thick	0.39	0.09
Avcothane - 0.18 in. thick	0.09	0.09
Lycra - 0.007 in. thick	0.12	0.18

The diffusion rates of both oils through candidate ventricle materials appear to be quite small.

A.3.4.2 Wear Tests

Wear tests were performed with a four-ball wear tester to evaluate blood pump mechanism oil additives that could extend the operating life of the mechanism bearings. The tester consists of four balls arranged in the form of an equilateral tetrahedron. The three lower balls are held stationary in a clamp and form a cradle in which the fourth or upper ball is rotated about a vertical axis for an arbitrary period of time under prescribed axial load and speed conditions. The points of contact are lubricated by immersion of the four-ball assembly in an oil pot. During the test, circular scars are worn on the surface of the stationary balls. The average diameter of the scars is used as a measure of wear.

Initially, sperm oil additive combinations with 1, 2, and 20 percent polyisobutylene and 1 percent antimony-dialkylphosphorodithioate were evaluated in 6 hour tests with a rotational speed of 600 rpm and a 5 kg (250,000 psi Hertz stress) axial load. The results obtained, shown in Table A.3-33, indicate that the ball wear with any of the polyisobutylene combinations was the same as that obtained with pure sperm oil, but the wear with the antimony compound was significantly less.

TABLE A.3-33

WEAR TEST DATA FOR SPERM OIL-ADDITIVE COMBINATIONS

<u>Lubricant</u>	<u>Additive</u>	<u>Average Scar Diameter, mm</u>
Sperm Oil	No Additive	0.42
Sperm Oil + 1% Additive	Polyisobutylene	0.42
Sperm Oil + 2% Additive	Polyisobutylene	0.42
Sperm Oil + 20% Additive	Polyisobutylene	0.42
Sperm Oil + 1% Additive	Antimony Compound*	0.30

*Antimony - dialkylphosphorodithioate

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	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 showed 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.

A.3.4.3 Bearing Friction and Life

The pump drive mechanism bearings originally selected are given in the upper portion of Table A.3-34. Three developmental bearings subjected to life testing are given in the lower portion of the same table. As can be seen, two of the developmental bearings were possible replacements for the heavily loaded S4C platform stabilizer bearings. The third is a possible substitute for the less heavily loaded SR4C bearings of the mechanism.

The test setup for life tests of bearings is illustrated schematically in Figure A.3-194. In this schematic a driven runner surface applies load and rotation to the test bearing. The load is applied in a constant manner from a weight by a flexible wire and pulley arrangement. In Figure A.3-194 the test bearing is bathed in oil stirred by a propeller to simulate the mechanism cavity environment. A forerunner to the illustrated test rig did not have this circulating oil feature. Bearings tested in the forerunner rig failed prematurely from the buildup of wear debris within the bearings.

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

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

Transformer 55 Oil		Vitrea 21		Sperm Oil	
Time	Torque (mg-mm)	Time	Torque (mg-mm)	Time	Torque (mg-mm)
24 hr	22,900	24 hr	21,000	24 hr	22,950
456 hr	23,800	192 hr	21,000	168 hr	20,100
1008 hr	24,650	648 hr	21,250	672 hr	19,600
1292 hr	25,650	1272 hr	21,700	1220 hr	21,000

The bearings showed little signs of wear upon inspection.

TABLE A.3-34

PUMP MECHANISM BEARINGS

MPB Corp. Designation	Size	Where Used	Max. Load (lb)	nd ²	Load Rating (lb)	Type Bearing
S4C___	0.078" ID x 0.250" OD	Platform Stabilizing Rollers	6.55	0.0195	47	Ball
S4C___	0.078" ID x 0.250" OD	Platform Torque Resisting Rollers	~0	0.0195	47	Ball
SR3C___	0.188" ID x 0.500" OD	Crankshaft End	11.75	0.0615	140	Ball
SR4C___	0.250" ID x 0.625" OD	Crankshaft End	11.75	0.0702	156	Ball
SR4C___	0.250" ID x 0.625" OD	Main Scotch Yoke	11.75	0.0702	156	Ball
SR4C___	0.250" ID x 0.625" OD	Platform Scotch Yoke	11.75	0.0702	156	Ball

DEVELOPMENT BEARINGS

F155___	0.078" ID x 0.350" OD	Platform Stabilizing Rollers	6.55	0.0381	---	Ball
F160___	0.078" ID x 0.350" OD	Platform Stabilizing Rollers	6.55	0.0381	---	Roller
F338___	0.250" ID x 0.720" OD	Main Scotch Yoke	11.75	0.0932	----	Ball

Figure A.3-194. Bearing Test Apparatus

A similar test was performed using SR4 ball bearings and Mobile +K lubricating oil. The test was performed with the bearings rotating at design speed, 120 rpm, and with a radial load of 50.8 pounds. The torque required to rotate the bearings, tabulated hereafter, did not change significantly throughout the test, and an examination of the bearings after the test did not reveal any signs of wear or damage.

<u>Time (hr)</u>	<u>Torque (mg-mm)</u>
96	21,500
432	22,500
1032	22,800
1536	23,800

Life tests on a number of bearings in a number of different candidate oils were carried out. These tests were run to bearing failure or to program termination whichever occurred first. All tests were carried out at 120 rpm and a constant radial load on the bearing corresponding to the design maximum for the bearing. The results are summarized in Table A.3-35 following.

TABLE A.3-35
BEARING LIFE TESTS

<u>Bearing</u>	<u>Lubricant</u>	<u>Test Time (hr)</u>	<u>Remarks</u>
S4C	Sperm Oil	4,850	Failed - Ball Spalling
SR4C	Vitrea 21 Oil	18,624	Failed - Retainer Wear
S4C	Vitrea 21 Oil	7,272	Failed - Retainer Wear
F155	Sperm/Jojoba Oil	16,200	Changed to Jojoba Oil at 6048-hr
F160	Sperm/Jojoba Oil	15,168	Changed to Jojoba Oil at 5000-hr
F338	Jojoba Oil	10,338	---
F155	Vitrea 21 Oil	9,336	---

Testing of the last four bearings of Table A.3-35 was discontinued without bearing failure because of program termination. These tests clearly show that F155 development bearing should replace the S4C design selection bearings in the pump mechanism.

A.3.4.4 Mechanism Endurance Test

An endurance test of blood pump mechanism No. 6 was carried out. For the purpose of the test, the mechanism was fitted with University of Utah supplied Silastic ventricles and driven by electric motor through a flexible shaft. The loading device was a simplified mock circulation filled with a 1 percent saline solution. The test setup is illustrated in Figure A.3-195. The mechanism was filled with jojoba oil. It was operated continuously at the design speed of 120 rpm and pumped against nominal aorta arterial pressures of 120/80 mm-Hg.

At program termination the mechanism had accumulated 6909 hours of operation without any failures of the mechanism. Testing of the mechanism was interrupted several times during the period it was on test. Before the life test started, the mechanism had been operated for a total of 664 hours for other testing purposes without failure. A test history follows:

Hours to Interruption	Cumulative Hours	Reason for Interruption	Action Taken
384	384	Draw Oil Sample	Withdraw Oil Sample
1951	2335	Draw Oil Sample	Withdraw Oil Sample
94	2429	Compliance Check Flex Shaft	None
623	3052	Noise from Torque Meter	Replaced Torque Transducer
432	3484	Hole in Ventricle Diaphragm	Repaired Hole Replaced Jojoba Oil and 1% Saline
75	3559	None	None
1058	4617	Use Ventricles on Other Test	Partially Replaced Jojoba Oil
1040	5657	Replace Leaking Ventricles	Torn Down and Inspected Replaced Jojoba Oil
588	6245	Program Termination	Place in Storage

As may be noted from the test history, the mechanism was torn down and inspected at 5657 hours of the life test (6321 total hours). At that time substantial wear of both the pinion and gear of the Evoloid[®] gear set was observed as shown in Figures A.3-196 and A.3-197. No other severe wear was observed.

Figure A.3-195. Life Test Setup

Figure A.3-196. Evoloid Gear After 6321 Hours

Figure A.3-197. Evoloid Pinion After 6321 Hours

A.3.4.5 Drive Mechanism Power Losses

In the design of the implantable nuclear system blood pump drive mechanism, a decision was made to change the implantable model design from that of a preceding bench model by flooding the mechanism cavity with oil and using an Evoloid[®] speed reducer rather than a planetary gear speed reducer. The advantages in flooding the mechanism cavity with oil were better mechanism lubrication, desensitizing the unit from changes in barometric pressure, and provision for transfer of thermal converter waste heat to the blood without an increase in blood pump envelope or special heat transfer surfaces. The Evoloid[®] gear set was smaller than a planetary set for the same duty. It offered the opportunity to reduce mechanism envelope and volume. As an aid to making this decision, tests were run using an existing Scotch yoke mechanism with and without oil and with Evoloid[®] gear set and with planetary gear set. In addition, the two gear sets were evaluated independently.

The test setup for evaluating the difference in power losses in the presence or absence of oil in the mechanism cavity consisted of a 1/8 hp Bodine dc motor coupled with a Minarik speed indicating and control system. The motor was connected by flexible coupling to a calibrated Lebow 0 to 50 in-oz strain gage torque transducer. The transducer in turn was coupled to a bearing block used to stabilize the 12 in. long flex shaft connected to the blood pump. Figures A.3-198 and A.3-199 show the complete test stand and pump mount, respectively. The blood pump as viewed in Figure A.3-198, rotates CCW. A gaseous N₂ supply and pressure regulating system was connected to the ventricle system. This supply was used to pressurize this system to a nominal value of 2 psi to insure that the ventricle sacks were in proper contact with the pump mechanism pusher cups. Sperm oil was the oil used. During these tests no net work was being done by the ventricles, so in effect the oil tests were no-load or tare-torque evaluations. However, it should be added that the oil churning losses should not be much changed by the presence or absence of mechanism net power output.

Torque sensor output signals were conditioned and recorded by a Minneapolis-Honeywell Visicorder as a dynamic cycle with peaks and valley representative of the pulsating torque requirements of the reciprocating blood pump mechanism.

Figure A.3-198. Oil Filled Mechanism Test Stand

Figure A.3-199. Blood Pump Mounted on Test Stand

An end-to-end calibration of the data acquisition system was performed. The main function of this calibration was to check the ability of the system to reproduce dead weight torque values. Dead weight torque values were produced by locking the output shaft of the torque sensor and hanging calibrated weights on a lever arm clamped to the input shaft. A ± 4 in-oz scale was set up on the recorder as the optimum range for the data expected from the experiment. Results of the calibration indicate that a positive fixed error of 0.002 in-oz existed over the 4 in-oz scale. This bias error represented an insignificant error in terms of the full scale reading.

Six separate dynamic cycles were recorded at six different speeds on two different pump configurations. This amount of testing represents 36 sets of data per pump configuration.

Average torque was determined by measuring the area of an oscillograph trace over a dynamic cycle and dividing that value by the duration of the cycle. Areas were measured with a planimeter. The average torque associated with a particular rpm represented an average of six different area determinations. Rpm listed is an average shaft rpm calculated as follows:

$$\frac{1}{\text{sec/cycle}} \times 7.5 \text{ gear ratio} \times 60 \frac{\text{sec}}{\text{min}} = \text{shaft rpm}$$

Sec/cycle in this case is the time for one pump revolution; i.e., the pump period.

The pump mechanism was filled through a tube fitted to the shaft connection. This tube was filled with oil and air trapped in the mechanism was withdrawn through the oil using a syringe. When the syringe was removed, atmospheric pressure forced oil into the pump. This operation was repeated until no more air could be extracted. Pressurization of ventricles with 2 psi N_2 would then force some oil out of the flex shaft, indicating that no air was trapped in the system.

An evaluation of experimental accuracy was made considering the existence of fixed and accidental or random errors in the data acquisition system and data

reduction techniques. The overall error was determined by combining the square root of the sum of the squares of estimated uncertainty for each component.

The test setup for evaluating the two speed reducer sets also consisted of a 1/8 Hp Bodine dc motor coupled with a Minarik speed indicating and control system. The motor was connected by flexible coupling to a calibrated Lebow 0 to 50 in-oz strain gage torque transducer. The transducer in turn was coupled to the planetary gear train mounted on a test block. The output shaft of the gear train was connected by flex coupling to a calibrated Lebow 0 to 200 in-oz strain gage torque transducer. The output shaft of the transducer was connected to two Magtrol hysteresis brakes connected mechanically in series. The range of each brake was 0 to 120 in-oz. With the coils of the units connected in parallel the loading capability of the test stand was in excess of 200 in-oz. Rotation of the gear train was in the same direction as it would be during normal pump operation.

Torque sensor output signals were conditioned and readout on a Vidar 520 Integrating Digital Voltmeter system. Rpm was measured with a strobotac. Figures A.3-200 and A.3-201 show the total test stand and gear train, respectively.

Experimental techniques consisted of input and output torque readings as load was increased over a range at constant rpm. The total load applied to the gear box output shaft was the reading taken from the output shaft torque transducer. The load applied by the hysteresis brakes was the transducer reading minus the output tare torque measured at zero load. This tare torque was included in the data and calculations of loss because it is a real number characteristic of the gear train configuration. Some positive error is associated with this number due to the energy required to turn the free-wheeling brakes. For this experiment, the load was varied from 0 to 200 in-oz while speed was held constant at 900 rpm.

An end-to-end calibration of the data acquisition equipment was performed. The main function of this calibration was to check the ability of the system to reproduce dead weight torque values. Dead weight torque values were produced

Figure A.3-200. Gear Set Test Stand

Figure A.3-201. Planetary Gear Set Mounted on Test Block

by locking the transducer output shaft and hanging calibrated weights on a lever arm clamped to the input shaft. Results of the calibration indicated that the equipment reproduced a dead weight value to within +0.35 percent of sensor full-scale range.

Torque losses were calculated by the difference between input and output torque following conversion of output torque to terms of input rpm. This operation consisted of dividing output torque by the gear train ratio of 7.5.

Power in terms of watts was calculated from the expression:

$$P_{\text{watts}} = \frac{T \times \text{rpm}}{1352} \text{ where } T \text{ is in in-oz.}$$

The initial cavity oil-filled test was carried out using the planetary speed reducer. The power required to rotate the mechanism was measured with no oil in the drive, oil in the mechanism (including the speed reducer), and oil in the mechanisms and flexible shaft. The values recorded at a blood pump beat rate of 120 (flex shaft speed of 900 rpm) are as follows:

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

Filling the pump drive with sperm oil increased the required drive power only 0.20 watt. This seems a small sacrifice for the other advantages gained from use of an oil-filled mechanism system. It will be noted that half the oil loss is associated with its use in the flexible shaft.

A summary of the data for no oil and complete filling with oil is given in Figure A.3- 202 for mechanism with planetary gears and mechanisms with Evoloid[®] gears.

Figure A.3-202. Effect of Oil in Pump Drive on No-Load Input Power

The power loss evaluations of the gear sets separately, grease lubricated, is given in Figure A.3-203. The speed values given are those of the flex shaft or gear set input. The speed reduction across the gear sets is 7-1/2 to 1.

Taking these two sets of data from Figures A.3-202 and A.3-203, the Evoloid[®] gears showed a lower no-load loss but a higher loss under load. The lower no-load loss is particularly noticeable in the oil-filled condition. These effects tend to balance each other in making a choice of gear sets. An estimate of the net effect in terms of total power loss for each gear type driving an oil-filled pump at design speed and load is shown in the following table:

TABLE A.3-36
AVERAGE POWER LOSS IN WATTS FOR VARIOUS PUMP
DRIVE MECHANISM CONFIGURATIONS AND CONDITIONS

	Speed Reducer [®]	
	Planetary Gear	Evoloid [®] Gear
Pump Drive Mechanism (including Speed Reducer) Oil-Filled, No-Load, 900 rpm	0.45	0.26
Gear Box Only, 900 rpm, 6 W Input	0.45	0.67
Gear Box Only, 900 rpm, No-Load	0.17	0.08
Loss in Gear Box due to Design Pump Load	0.28	0.59
Total Loss Oil-Filled Mechanism, 6 W Input*	0.73	0.85

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

A.3.4.6 Magnesium and Beryllium Corrosion Investigation

Both magnesium and beryllium because of their low densities, would be attractive materials for use in the blood pump mechanism, if they can be protected against corrosion by the body. Laboratory tests were conducted on magnesium and beryllium coupons, protected in various ways, by immersing the coupons in a 1 percent saline solution for various periods of time and observing the results.

Figure A.3-203. Gear Set Power Loss Versus Input Power

Magnesium Tests

Three series of tests were carried out. In the first series coatings some times used for protection of magnesium in salt water environments were tried rather unsuccessfully. In the second series of tests the bare metal coupons were encapsulated in Silastic by two different methods; first by brushing on the adhesive and second by forming a jacket from Silastic sheet. This latter method proved the most promising and the third series was carried out using sheet thickness as the parametric variable.

For the first series of tests, candidate thin corrosion protection coatings were selected as given in Table A.3-37. Magnesium test coupons with these coatings were made and subjected to cyclic flexure testing to evaluate the ductility and adherence of the coatings at the expected strain levels in the pump housing. Each specimen was cycled 2000 times with a maximum strain of 0.15 percent. No evidence of damage to any of the coatings was observed after the flexure tests.

TABLE A.3-37

CANDIDATE THIN COATINGS FOR CORROSION PROTECTION OF MAGNESIUM

<u>Treatment</u>	<u>Seal</u>	<u>Typical Dimen. Change (in.)</u>	<u>Application Specification</u>
DOW 17 Light	None	+0.0003	MIL-M-45202B, Type I, Class C
DOW 17 Light	Sodium Silicate	+0.0003	
DOW 17 Heavy	None	+0.0010	MIL-M-45202B, Type II, Class D
DOW 17 Heavy	Sodium Silicate	+0.0010	
DOW 9	None	None	MIL-M-3171D, Type IV
Magnadize	*	+0.0013	Type III, Class C**
Magnadize	Teflon	+0.0013	Type III, Class F**
Nickel Plate***	None	+0.0015	MIL-C-26074, Class 3, Grade C

*Proprietary; probably sodium silicate

**Coating meets requirements of MIL-M-45202

***Electroless nickel plate applied over a zinc immersion coating and copper strike

The specimens were immersed in a 1 percent saline solution at 100°F to evaluate the relative corrosion protection effectiveness of the electrochemical and plated coatings. The conditions of the specimens after one through nine days exposure is shown in Figure A.3-204. Coating failure and corrosion occurred to some degree in all specimens. Corrosive attack was most severe in the electroless nickel coated specimen, undoubtedly the result of galvanic coupling. Specimens coated with the various Dow treatments displayed overall surface corrosion, with attack being somewhat more severe in cases where lighter coatings were used. Sodium silicate sealing did not improve the corrosion resistance provided by the Dow 17 light and heavy coatings. The Magnadize coated specimens failed at several small edge sites. A few of these edge failures appeared to occur at locations where the specimen was supported for coating.

The corrosion condition of the Magnadize coated samples resulting from 21 days of 100°F salt water exposure, the longest time period examined, is displayed in Figure A.3-205. Corrosion progressed at the same sites detected in earlier examinations and a few new failure sites developed.

A general coating survival rating for exposure in 100°F salt water, in order of best-to-poorest, is 1) Magnadize Type III, Class F and Magnadize Type III, Class C; 2) Dow 17 heavy coat; 3) Dow 17 light coat and Dow 9; and 4) electroless nickel.

In the second series of tests, an examination of the corrosion protection provided magnesium by a coating of silicone rubber was carried out. The silicone rubber coatings evaluated were Medical Grade Silastic Adhesive Type A, and unreinforced 0.020-in. thick vulcanized Medical Grade Silastic Sheeting. Testing involved immersing magnesium coupons coated with the Silastic materials in a 1 percent salt (NaCl) water bath held at 100°F, with periodic visual examination used to evaluate corrosion behavior. Photographs revealing the severe corrosion resulting when unprotected magnesium is placed in this saline environment are presented in Figure A.3-206. Test coupons of magnesium were made from ZK60A alloy.

Dow 17 (c)
Light Coat

Dow 9 (b)

Dow 17 (d)
Light Coat and Silicate Seal

Electroless Nickel (a)

Dow 17 (d)
Heavy Coat

Magnadize (e)
Type III Class F

Dow 17 (d)
Heavy Coat and Silicate Seal

Magnadize (e)
Type III Class C

Figure A.3-204. Corrosion of Coated Magnesium Specimens Caused by 1 to 9 Days of Exposure in 1 Percent Salt Water Held at 100°F. (a) 1 Day; (b) 4 Days; (c) 5 Days; (d) 6 Days; (e) 9 Days. (Small and not readily apparent sites of coating failure are circled.)

Magnadize
Type III, Class F

Magnadize
Type III, Class C

Figure A.3-205. Corrosion of Magnadize Coated Specimens Caused by 21 Days Exposure in 1 Percent Salt Water Held at 100°F. (Small and not readily apparent sites of coating failure are circled.)

The Silastic coating conditions evaluated to date are defined in Table A.3-38. Coating of the Type A adhesive and vulcanized sheeting, both alone and in combination, were examined. It should be noted that a small amount of Type A adhesive was used to seal the edge of the sheeting package applied around magnesium specimen No. 4, but that this material did not intimately contact the specimen.

Photographs of the Type A adhesive coated specimen No. 1 are shown in Figure A.3-206. Note that defects were present in the coating as a result of air entrapment during brushing. A minimum of two brush coatings totaling $\sim 1/32$ in. in thickness, the second applied after the first has cured, are required to obtain a Type A adhesive coating that does not exhibit evidence of the presence

of small holes. Small holes in a Type A adhesive coating, which are usually unapparent to the unaided eye, can be detected by use of a high voltage arc probe. The arc will scatter around when passed over solid coating, but will concentrate to any site where coating is missing. This inspection technique was used to insure that holes were not present in any of the Silastic coatings applied over the corrosion test specimen.

(a)

(b)

(c)

Figure A.3-206. Corrosion of Magnesium Alloy ZK60A in 1 Percent Salt (NaCl) Water Held at 100°F - (a) Time Zero; (b) 1 Day; (c) 4 Days.

TABLE A.3- 38

SILASTIC COATING CONDITIONS EVALUATED FOR THE PROTECTION OF MAGNESIUM

Spec. No.	Silastic Coating	Coating Application	Adhesive Curing Condition
1	Type A Adhesive	Brush. Two coats 0.032 in. total thickness	Ambient. First coat 2 days; Second coat 6 days.
4	Vulcanized Sheeting 0.020 in. thickness	Specimen sandwiched between two sheets bonded at edge with Type A adhesive	Ambient. 8 Days
26	Type A Adhesive and Vulcanized Sheeting	Specimen sandwiched between vulcanized sheets coated with uncured Type A adhe- sive. 0.014 in. adhesive thickness and 0.020 in. sheeting thickness	Ambient. 8 Days

The corrosion behavior of Silastic Type A adhesive coated specimen No. 1 is shown in Figure A.3-207. The Silastic envelope bulged slightly during the first day of immersion in 100°F salt water, and the magnesium specimen was observed to darken slightly. The bulged coating condition persisted without change as time was accumulated in the corrosive environment, and the magnesium specimen developed a rough and grey appearance. The test was terminated after 23 days, and the coating stripped to clearly reveal the magnesium specimen appearance. The specimen was found to be corroded uniformly over its entire surface. The corrosion developed, however, was superficial and far less than that which would have occurred had bare magnesium been exposed in the saline solution.

Photographs revealing the influence of salt water exposure on the appearance of specimen No. 26, which was brush coated with Type A adhesive and then covered with vulcanized sheeting, are shown in Figure A.3-208. Chemical attack similar to that of Type A adhesive coated specimen was observed in this specimen.

Figure A.3-207. Magnesium Specimen No. 1 Brush Coated with Silastic Adhesive Type A. Note the presence of coating defects caused by air entrapment.

Figure A.3-208. Silastic Type A Adhesive and Vulcanized Sheetting Coated Magnesium (Specimen No. 26) Exposed in Saline Solution. (a) Time Zero; (b) 1 Day Exposure in 100°F 1% Salt Water; (c) 23 Days Exposure

Photographs revealing the appearance of specimen No. 4 after 17 and 30 days of salt water exposure are shown in Figure A.3-209. Exposure in the saline solution caused no corrosion of the magnesium, although water collected in the specimen compartment (entrapped droplets and water vapor are apparent in the photographs). The obvious conclusion that can be drawn is that salt ions did not penetrate the cured Silastic sheeting barrier. Furthermore, since the magnesium specimens protected by Silastic coatings containing Type A adhesive did display a measure of chemical attack during salt water exposure, it would appear that this must have occurred by water leaching some corrosive chemical component from the adhesive. Acetic acid would be the logical corrosive chemical to suspect since it is a major component of the Type A adhesive.

In the third series of tests, results were obtained in corrosion tests of magnesium covered solely with Silastic sheeting in 0.020, 0.040 and 0.060 in. thicknesses.

Photographs revealing the appearance of a magnesium specimen covered with a 0.020 in. thickness of Silastic sheeting and then exposed to 100°F salt water for 17 to 55 days are shown in Figure A.3-210. Exposure to the saline solution caused no apparent corrosion of the magnesium specimens for up to 30 days, even though water penetrated the sheeting barrier (entrapped droplets of water can be seen in the 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 in. thicknesses of Silastic sheeting and exposed for 36 days to salt water are shown in Figure A.3-211. These specimens tarnished in a manner similar to that noted for magnesium protected with 0.020 in. thick sheeting. Very small droplets of water, which are not apparent in the photographs, were also present inside the Silastic covering. The specimens protected with 0.040 and 0.060 in.

thick covering of Silastic sheeting were left on test for a total of 3 months. The metal was not severely corroded but did tarnish progressively during this period.

(a)

(b)

Figure A.3-209. Vulcanized Silastic Sheeting Coated Magnesium Exposed in Saline Solution. (a) 17 Days Exposure in 100°F 1% Salt Water; (b) 30 Days Exposure

(a)

(b)

(c)

(d)

Figure A.3-210. Magnesium Coated with 0.020 in. Thick Silastic Sheeting and Exposed to Saline Solution. (a) 17 Days Exposure in 100°F 1% Salt Water; (b) 30 Days Exposure; (c) 48 Days Exposure; (d) 55 Days Exposure (covering opened).

As-coated

37 Days
Exposure

Figure A.3-211. Magnesium Specimens Coated with 0.040 and 0.060 in. Thicknesses of Silastic Sheeting and Exposed to Saline Solution. 0.040 in. Sheeting Thickness on Spec. No. 18; 0.060 in. Thickness on Spec. 17.

Beryllium

An effort to evaluate the salt water corrosion protection provided beryllium by Nedox coatings was 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. They report them to be resistant to attack in sea water, and their applications have included desalination plant tubing.

Two Nedox coatings, whose characteristics are described in the following table, 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 A.3-212. The Teflon and electroless nickel layers measured nominally 0.0010 and 0.0005 in. in thickness, respectively.

TABLE A.3- 39

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**	MIL-C-26074

*Similar to hard black Teflon commonly found on cookware
**Easily removed with thumb nail

The coatings were applied to 2 in. diameter by 1/8 in. thick discs of beryllium. Photographs showing typical coating specimens are presented in Figures A.3-213 and A.3-214. Visual examination revealed the presence, in all cases, of coating defects, typical examples of which are also shown in Figures A.3-213 and A.3-214. 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 Figure A.3-215 and A.3-216. 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.

Figure A.3-212. A Photomicrograph Showing the Cross Section of Nedox U-4 Coated Beryllium

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. Corrosion developed at hole locations that 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.

1X

10X

Figure A.3-213. 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 A.3-214. Nedox SF-2 Coated Beryllium Corrosion Test Specimen.
(Note coating defects revealed in the higher magnification photomicrographs.)

Figure A.3-215. The Progressive Corrosion of Nedox SF-2 Coated Beryllium Caused by Exposure to 1 Percent Salt Water Held at 100°F

Figure A.3-216. The Progressive Corrosion of Nedox U-4 Coated Beryllium Caused by Exposure to 1 Percent Salt Water Held at 100°F

A.3.4.7 Mechanism - Ventricles Interface Investigations

Two laboratory investigations are covered under this mechanism-ventricle interface handling. The first of these was a test to determine the effects of pulling a vacuum on the mechanism cavity. Pulling a vacuum on the ventricles gives better seating of the ventricles and hence a larger stroke volume. As will be seen a mild vacuum causes no problems. A somewhat larger vacuum can be very damaging to the mechanism.

The second investigation was to see if unrestrained ventricle diaphragms could be used. It worked well in the laboratory. When implanted, two successive pinching type failures of the ventricles were experienced.

Mechanism Cavity Vacuum

A test was performed in which a blood pump was operated at normal pressure and flow conditions and the shaft input torque was measured as mechanism cavity vacuum was increased. Although the pump began to squeak at about 8 in-Hg, the maximum value of the basic torque-time curve did not change significantly from the normal 24 in-oz when the vacuum was increased to the maximum possible value of 16 in-Hg reported by the University of Utah for an implant experiment. However, spikes in the torque-time curve that increased the maximum torque value began to appear at about 12 in-Hg and gradually increased in number and magnitude to a value of about 44 in-oz at 16 in-Hg. A second vacuum test was performed, and the vacuum was increased to 19 in-Hg, where a maximum spike torque of about 84 in-oz was reached. The results of the second test indicated that very high spike torques and failures could be caused by a high vacuum.

After the vacuum test, the pump was disassembled and the yoke and crankshaft assembly were photographed. These photographs (see Figures A.3-217 and A.3-218) show that the roller and yoke bearing surfaces were severely scored at the high vacuum conditions when the yoke was not able to resist the atmospheric pressure forces that tend to collapse the yoke around the roller. It was recommended that cavity vacuum not be allowed to exceed 4 to 5 in-Hg.

Figure A.3-217. Yoke After Vacuum Test

Figure A.3-218. Crankshaft After Vacuum Test

Unrestrained Blood Pump Ventricle

A ventricle diaphragm secured only to the blood pump housing rather than to the housing and pusher cup was evaluated. This unrestrained diaphragm arrangement could permit the blood pump ventricles to be implanted separately and avoid the need for quick-connects between the ventricles and the blood vessel grafts. Unrestrained ventricle diaphragms provided by the University of Utah were assembled on an oil-filled pump mechanism and the assembly, shown in Figure A.3-219, was operated without ventricle domes so that the behavior of the diaphragms could be observed. The test setup is shown in Figure A.3-220. The diaphragms automatically positioned themselves properly and adhered tightly to the pusher cup domes throughout the pumping cycle, as they would if they had been attached to them. Photographs of one of the ventricle diaphragms in the full-in and full-out positions are shown in Figure A.3-221. This test was repeated with the mechanism chamber dry and at a vacuum pressure of 4 in-Hg. Again, the ventricles positioned themselves properly and adhered to the pusher cup domes. However, in this case, it was necessary to apply a lubricant on the pusher cup domes to allow the ventricles to properly position themselves rapidly. Based on these bench observations, in vivo use of the unrestrained ventricles, was attempted. Two successive attempts resulted in pinching type failures of the ventricles.

A.3.4.8 Pump Size - RPM Tradeoff Analysis

An analytical model of blood pump flow-pressure characteristics was developed using parameters adjusted to match the observed pump bench test performance at 110 and 130 beats per minute. Calculations were made with this model to predict flow-pressure performance at higher and lower speeds. In the model, the valve leakage into the pump from the aorta during diastole was assumed to be at a constant rate, as was the leakage out of the pump into the veins during systole. In addition, the backflow that occurs during the closing of each valve was assumed to be a constant volume of blood per beat. These assumptions concerning valve action should cause the model to predict somewhat higher flow that could actually be achieved at a given filling pressure. At high speeds, there is a tendency for the valves to bounce, decreasing the net flow rate below that predicted. At low speeds, the hydrodynamic forces closing the valves are reduced, causing sluggish action, and again reducing the net flow.

Figure A.3-222 shows the predicted performance for the IVBM model pump. As can be seen, increasing the speed from 130 to 170 beats per minute reduces the venous pressure required to achieve 12 liter/minute flow by only about 2.5 mm-Hg. Further increase in speed will yield no further reduction in pressure. Operation at lower than design speeds will generally increase the filling pressure required to achieve a given cardiac output. By making speed a function of venous pressure, characteristic curves, as shown by the dashed lines on Figure A.3-222 could be achieved. While these steepen the cardiac output curve, the pressure required for a given flow is generally higher. Figure A.3-223 shows the effect of reducing the displacement of the pump from 110 to 100 cc while maintaining the same flow parameters and valves as the IVBM pump. For every pump speed, the flow obtained at a given pressure has been reduced. Since cardiac output curve for the 111 cc IVBM pump may be marginal, clearly a simple reduction in pump size is unacceptable.

Figure A.3-219. Blood Pump Mechanism with Unrestrained Ventricle Diaphragms

Figure A.3-220. Unrestrained Ventricle Diaphragm Test Rig

Figures A.3-224 and A.3-225 show the effects of changing various flow parameters at speeds of 130 and 110 beats per minute, respectively. In these figures the curves labeled (1) are representative of the IVBM pump. The pumps displace 111 cc, and the valves are assumed to be the same as for the IVBM pump. The maximum flow achieved for each pump is set by the valve performance. Improvements in the valves would have the effect of shifting the entire set of curves upward. The parameter K represents the spring constant of the compliant portions of the pump, and has the units of mm-Hg/cc. The parameter I is a measure of the inertia of the fluid entering the pump and has the units mm-Hg/(cc/sec²), while F is a frictional loss term with the units mm-Hg/(cc/sec)². By examining Figure A.3-224, it can be seen that reducing the friction term tends to shift the curve slightly to the left, increasing the flow somewhat, but not a great amount. Decreasing the inertia term moves the flow curve significantly to the left, producing significantly higher flows for a given pressure, and increasing the slope somewhat. Curves (3) and (4) achieve maximum flow at quite reasonable pressures, but have higher flow than desired at low pressures. Decreasing the stiffness of the compliant portion of the pump produces curve (5) which appears to match the physiological requirements quite well.

Figure A.3-221. Unrestrained Ventricle Diaphragm in Full-In and Full-Out Position

Figure A.3-222. Cardiac Output as a Function of Venous Pressure and Speed for IVBM Pump

Figure A.3-223. Cardiac Output as a Function of Venous Pressure and Speed for 100 cc Pump

Figure A.3-224. Effect of Variation of Flow Parameters on Performance for 111 cc Pump at 130 B/M

Comparison between Figures A.3-224 and A.3-225, indicate that there is little change in flow with speed in the low pressure region for any given pump while low speeds limit the maximum flow attainable. It appears, therefore, that the use of variable speeds will not materially help in improving the cardiac output characteristics of the pump. The maximum flow which can be achieved by a given pump is set by the pump size, speed, and the performance of the valves used. The venous pressure at which the maximum flow is achieved is primarily a function of the inertia of the fluid entering the pump, with flow friction in the inlet providing a relatively minor contribution. The flow at lower inlet pressures is primarily a function of inertia and the stiffness of the compliant portions of the pump. The study has indicated then that variable speed operation would offer little advantage in terms of pump response to physiological demands.

The stiffness parameter, K , is a function of the thickness, the size, the shape, and the material forming the pump domes. In general, K can be reduced by making the dome thinner in relation to its diameter. More flexible materials will reduce the value of K . It is difficult to relate the shape of the dome directly to the value of K . This requires a detailed analysis of the particular shape involved. As the collapse of the dome during filling is essentially a buckling process, analysis of the undeformed shape will yield little information.

The physical interpretation of the inertial and friction parameters (I and F) can best be explained by briefly outlining the derivation of the analysis used in this study.

For one dimensional, frictionless flow of an incompressible fluid, the axial pressure gradient can be related to the flow rate by:

$$\frac{\ddot{Q}}{A} + \frac{1}{2} \dot{Q}^2 \frac{\partial}{\partial z} \left[\frac{1}{A^2} \right] = - \frac{g}{\rho} \frac{\partial P}{\partial z} \quad (1)$$

where:

$\dot{Q}$ = instantaneous flow rate

$\ddot{Q}$ = rate of change of flow rate

A = cross-sectional area of flow channel

Figure A.3-225. Effect of Variation of Flow Parameters on Performance for 111 cc Pump at 110 B/M

z = axial distance along channel

P = pressure

ρ = density of fluid

g = acceleration due to gravity

$\dot{Q}$ and $\ddot{Q}$ are independent of the axial distance along the tube, as the fluid is incompressible. Therefore, Equation (1) can immediately be integrated along the length yielding:

$$\dot{Q} \int_0^L \frac{dz}{A(z)} + \frac{1}{2} \ddot{Q}^2 \left[\frac{1}{A(L)^2} - \frac{1}{A(0)^2} \right] = g/\rho [P(0) - P(L)] \quad (2)$$

Multiplying through by ρ/g , the coefficient of $\ddot{Q}$ is a function of geometry and the properties of the fluid only,

$$I = \rho/g \int_0^L \frac{dz}{A(z)} \quad (3)$$

In this expression, the cross-sectional area, A , can be only a function of the distance, z .

In general, the losses in an expansion are high; therefore, the second term can be represented by:

$$F = \rho/g \frac{C}{A_{\min}^2} \quad (4)$$

Where $A_{\min}$ is the minimum cross-sectional area of the tube, and C is a constant. In general, C will have a value ranging from 0.2 for very carefully designed diffusers to something greater than 1.

Equation (1) has now been reduced to an ordinary differential equation in time:

$$I \ddot{Q} + \frac{1}{2} F \dot{Q}^2 = P_v - P_p \quad (5)$$

Where P_p is the pressure in the ventricle during filling, and P_v is the venous return pressure (assumed constant through diastole). The pressure in the ventricle is given by:

$$P_p = K \left[Q + Q_I - \frac{V_0}{2} (1 - \cos \omega t) \right] \quad (6)$$

In Equation (6), K represents the stiffness of the pump, Q_I is the volume of blood leaking through the valves, and V_0 is the pump stroke volume. Equations (5) and (6) are combined to compute the blood flow into the pump during diastole. Q_I is assumed linear with time, and Q is assumed zero until time, t_0 , when P_p , as defined in Equation (6), goes negative. With the boundary conditions that $\dot{Q}$ and $\ddot{Q}$ are zero at time, t_0 , Equation (5) is integrated numerically until P_p again goes positive. The value of Q , minus a constant term representing the loss as the intake valve closes and a second constant rate leak during systole, gives the net quantity of fluid pumped per stroke. The parameters, F , I , and K , as well as the valve leakage, were adjusted to obtain the best possible fit with the data presented by Bachman, et al.* for pump speeds of 110 and 130 beats per minute, to simulate the IVBM pump. Values of F , I , and K for this case are shown as Case I in Figures A.3-224 and A.3-225. The leakage terms that gave a best fit were:

$$\begin{aligned} \dot{Q}_I &= 50 \text{ cc/sec during diastole} \\ Q_I &= 10 \text{ cc/ sec during systole} \\ Q_I &= 0.0 \text{ for the valve closing term} \end{aligned}$$

The values of F and I obtained for a best fit are considerably higher than expected from measurements of the pump and connecting tubing. The reason for this discrepancy is not clear. In order to reduce the value of these parameters, the inlet flow path must be examined in detail for flow restrictions that are not immediately apparent. The equations listed are based on one-dimensional flow and for abrupt changes in area, they are not strictly true. The friction terms (as expressed in Equation (4)) are equivalent to the loss through an abrupt change in area when C is 1 to 1.5. In evaluating the inertia term from Equation (3), the presence of a jet of finite length downstream of an orifice must be considered.

*D.K. Backman, F.M. Donovan, G. Sandquist, and W.J. Kolff, "The Design and Evaluation of Ventricles for the AEC Artificial Heart Nuclear Power Source," American Society for Artificial Organs 19th Annual Meeting, Statler Hilton Hotel, Boston, Massachusetts (April 7-9, 1973).

A.3.4.9 Analysis of Linear Bearing Failure

During the course of the program there were two linear bearing failures during blood pump operation. These failures were totally unexpected because these bearings are lightly loaded. They serve to guide the pusher rod between the scotch yoke and pusher cups of the pump mechanism. This is an account of the investigation of the second of these failures from which it was concluded that the bearing failed because it had not been manufactured properly.

A description of the linear bearing is presented by Figure A.3-226. A photograph of the failed bearing is shown in Figure A.3-227. The bearing failed by outer sleeve cracking across the end ring support lip and circumferentially around the base plane of the internal chamber. The outer sleeve circumference was cracked along approximately 90 percent of its length, and the ring of material so separated had sprung to a diameter greater than original, indicating a release of tensile hoop strain had occurred. (Note the widening of the axial crack branch in Figure A.3-227.) The level of residual tensile stress within the cracked ring of material was estimated from crack dimensions as being in the order of 250,000 psi. The ultimate tensile strength of the outer sleeve hardened to RC60 is ~325,000 psi.

A microstructural examination revealed that crack propagation had occurred along grain boundaries without any apparent plastic deformation. This information combined with the evidence of a very high residual tensile stress within the cracked region were concluded to be signs that a stress-corrosion failure mechanism had operated. Chloride stress-corrosion cracking was suspected because some saline contamination could have occurred during a coolant circulating system test.

Tests were run to evaluate the susceptibility of linear bearings to chloride stress-corrosion cracking. This involved exposing bearings in salt water to determine whether outer sleeve cracking would occur. All the linear bearings purchased for the artificial heart project had been supplied by one manufacturer in two orders, one filled in 1972, the other in 1975.

Salt water testing cracked the outer sleeves of bearings supplied in 1975 within an exposure period of two days or less (three bearings were tested), while the

Figure A.3-226. Construction of the Linear Blood Pump Bearing (4.5X) . Ball Bearings in a Plastic Retainer are Restrained Inside a Sleeve by End Rings Captured Within Grooves Formed by Internal Chambers.

Figure A.3- 227. The Failed Linear Bearing. Failure Occured by Outer Sleeve Cracking Across an End Ring Support Lip and Around the Base Plane of the Internal Chamber. The Outer Sleeve Circumference was Cracked over approximately 90 percent of its Length.

sleeves of bearings manufactured in 1972 did not crack when tested for seven days (two 1972 bearings were tested). The bearings that failed during blood pump testing had been manufactured in 1975. Photographs showing the outer sleeve failure of a 1975 bearing caused simply by exposure to salt water are presented in Figure A.3-228.

The 1972 and 1975 bearing sleeves exhibited similar hardnesses, RC60, and overall microstructural conditions. They differed, however, in microstructural detail in the end ring support lip regions. Close inspection revealed a worked metal layer around the chamfer forming the end ring support lips on the 1975 outer sleeves, indicating that machining had been performed after the hardening heat treatment. This condition was not observed at these regions of the 1972 outer sleeves, indicating that the end ring support lips were machined prior to hardening.

The results of salt water testing and microstructural investigations were presented to the bearing manufacturer who confirmed that the 1972 and 1975 outer sleeves were fabricated in a different manner. Already hardened 0.750 in.-long outer sleeves prepared for a high production bearing were shortened to our special 0.718 in. length requirement in the 1975 bearings by remachining the end ring support lip regions. The sleeves of the 1972 bearings were fabricated to our special size requirement using the standard production bearing procedure of completely machining the end ring support lips prior to hardening. It was concluded that a redistribution of residual stresses resulting from machining the hardened outer sleeves left the end ring support lips under a high tensile stress and, consequently, very susceptible to chloride stress-corrosion cracking.

Before Exposure
to Salt Water

After Two Days
in Salt Water

Figure A.3-228. Outer Sleeve Cracking of a 1975 Vintage Linear Bearing. The Bearing was Photographed Before and After Salt Water Exposure.

A.3.5 WASTE HEAT REJECTION

A.3.5.1 Bellows Pump Water System

As stated earlier in Volume I, bellows arrangement shown in Figure A.3-229 was incorporated into the nuclear system blood pump to circulate water to transport the thermal converter waste heat to the pump. The heated water from the converter rejects heat across the walls of the bellows to the oil in the pump mechanism chamber. Then the heat is transferred from the oil to the blood in the ventricles by conduction through the pump components. Because the travel of the pump displacement cup yoke during the pumping cycle is twice as large as that of the yoke housing, a relative motion of one-half the yoke travel occurs between the ends of the yoke drive rods and the top and bottom walls of the yoke housing. This motion is used to actuate the bellows, which are installed around the yoke housing bosses for the linear bushings that guide the yoke.

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

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

Because the same dimensions are specified for manufacturing both bellows, only a very small compliance volume is required in the water loop to accommodate any difference in the stroke volumes of the two bellows. A small diaphragm was

Figure A.3-229. Converter Coolant System Schematic Diagram

incorporated in one of the end closures of one of the bellows to provide the compliance volume required. This diaphragm will bulge either inwardly or outwardly as required to provide a coolant loop volume that will remain constant during the pumping cycle.

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

Flow and Power Required Performance

The overall performance of the coolant loop depends upon the heat transfer across the coolant pump bellows. A test rig was fabricated to determine the characteristics of the coolant loop. The tests were performed to determine the following parameters:

- Coolant flow rate
- Hydraulic power dissipated in the coolant loop
- Mechanical power required to drive the coolant pump
- Heat transfer between coolant and sperm oil filling the blood pump mechanism

To achieve these goals, the test rig consisted of two major components which were connected by flexible tubing. The thermal converter was simulated by a mockup of the cold end heat exchanger which was electrically heated. The blood pump was represented by a mechanism designed to simulate the action of the pump components under actual use. It was decided not to use an actual blood pump mechanism for this purpose to allow ready access to the pump components for instrumentation. The test coolant pump consisted of components designed for use in the actual blood pump slightly modified to accept pressure taps and to allow attachment to the test rig.

Prior to activation of the bellows test rig, two bellows were mounted in a blood pump with the flexures and other required components. Flow measurements taken with the bellows pump installed in the blood pump indicated a coolant flow of $0.25 \text{ in}^3/\text{sec}$ (near the high end of the design tolerance), while operating at 120 beats per minute. No estimates of pumping power for the coolant loop were made because of difficulties in reproducing the pressure and flow conditions in the mock circulation loop. Bellows failures prevented further experimentation.

Two bellows mounted in the bellows test rig were used to estimate the power required to circulate coolant through the coolant loop. With the test rig filled with sperm oil, tests were run at 100, 120 and 130 beats per minute with the coolant loop empty, then repeated with the loop filled with water. At a given beat rate, the power consumption with the coolant loop filled, compared to the consumption with the loop empty, represents the power required to circulate the coolant. To minimize the effect of uncertainties in the rig drive train performance, the displacement of the bellows assemblies and the force required to move the assemblies were measured and recorded. By plotting force as a function of displacement, and integrating the area of this closed curve, an accurate measurement of the work required to drive the bellows assemblies was obtained, and thus the power consumption. At 120 beats/minute, 0.70 watts were required to drive the loop while dry and 0.76 watts with water in the loop. The power dissipated in the coolant loop was therefore 0.06 watts. This was slightly below the nominal predicted power consumption of 0.08 watts, and well within the design range of 0.02 to 0.19 watts.

The total power required to drive the bellows (0.70 watts) was considerably higher than expected. Measurements taken on the dry test rig without bellows indicate that 0.05 watts are required to drive the rig alone. Estimates of the power dissipated in the sperm oil indicate that this should be of the order of 0.1 watts or less. The discrepancy of approximately 0.5 watts was not explained. Some of this energy was clearly being dissipated in the bellows since their short life indicated they were being operated beyond the elastic limit.

Structural Problems

When functional testing of the IVBM blood pump was started, the pump mechanism and the coolant system functioned well. After the pump had operated about an hour, leakage of mechanism oil into the coolant system was observed. An examination of the coolant system revealed that the mechanism oil leaked into the coolant loop through a tear in an end convolution of the outer bellows of one of the coolant pumps. Photographs of the bellows coolant pump and the failure site are shown in Figures A.3-230 and A.3-231.

The damaged bellows assembly was then replaced and testing was resumed. After an additional hour of operation, the test was stopped again because leakage of oil into the coolant loop was again observed. A crack in one of the coolant tubes inside the blood pump was found to be responsible for this leakage. A photograph of the damaged tube is shown in Figure A.3-232. The tube crack was attributed to early fatigue due to the wrinkled condition of the bent portions of the tube. Although all of the internal coolant tubes that were initially procured were found to have relatively large wrinkles after they were bent, it was decided to use these tubes for preliminary pump testing because new tubes from another supplier would not be available until several weeks later. Since the limited functional testing that had been performed indicated that the wrinkled tubes would not permit more than a few hours of pump operation, two of these tubes were modified so that the planned checkout testing could be performed before new tubes were received. A section of the internal coolant tube was removed and replaced with a plastic tube to avoid high stresses in the tube when it is flexed during the pumping cycle. A photograph of the modified tube is shown in Figure A.3-233.

After modified coolant tubes were installed and the blood pump had been operated about four additional hours, leakage of oil into the coolant loop was again observed. This leakage was found to be caused by a failure of the replacement coolant pump bellows similar to that of the original pump. The outer bellows of the eight coolant pumps procured for the IVBM blood pumps were then carefully examined for uniformity of bellows convolution deflection. It was found that the end convolutions of the outer bellows of all the pumps deflected two to three times as much as the other convolutions. The coolant pump supplier, Sealol, indicated that this large variation of the convolution deflections caused a severe overstress condition that probably was responsible for the bellows failures.

Figure A.3-230. Bellows Coolant Pump

(400X)

Figure A.3-231. Bellows Coolant Pump Failure Site

Figure A.3-232. Damaged Blood Pump Internal Coolant Line

Figure A.3-233. Modified Blood Pump Internal Coolant Line

Sealol manufactured a new set of bellows for a pump using improved welding and heat treatment fixturing. They then cycled the individual bellows to determine whether the new fixturing had improved the poor fatigue life reported above. All three of the bellows in the pump assembly failed before 10^5 cycles were completed. Examinations of the failed bellows indicated that the bellows plates did not nest properly when the bellows were fully compressed, and longer compressed lengths would be required to avoid plate clash.

Since the outer pumping bellows appeared to exhibit the severe plate interference, a new bellows of this type with a compressed length 0.125 in. longer than that of the original bellows was built and tested to establish whether adequate life could be achieved by increasing the compressed length. After this bellows survived 4×10^6 cycles, it was concluded that this configuration should have adequate life for system use.

Two new bellows pumps, manufactured by Metal Bellows Corporation, improved quality flexible feed tubes and new superalloy MP-35 N flexures were installed in a blood pump for a short-term test of the bellows pump coolant circulating system. Previous testing showed that the bellows pump survived 5×10^6 cycles without any evidence of leakage or distortion. The system functioned properly, and the measured flow slightly exceeded the design flow requirement of 240 cc/min at a beat rate of 120 beats/minute. However, after about three hours of operation, leakage of mechanism oil into the circulating loop was observed. When the system was disassembled for inspection, it was established that the leakage had occurred through a tear in one of the small connecting bellows that connects the flexible feed tubes to the blood pump coolant ports. The damaged bellows was replaced and operation of the blood pump was resumed. After about three additional hours of operation, leakage of oil into the coolant loop was again observed. When the system was disassembled, a tear was found in the replacement connecting bellows. Although dimensional measurements indicated that the bellows that failed were loaded more severely than the one that survived, the loading conditions were not as severe as those designed for and should not have caused the failures.

Metal Bellows was consulted regarding the difficulties experienced with the connecting bellows that had been supplied by Servometer. Because Metal Bellows indicated that properly designed and manufactured bellows would be lightly stressed in this application and should have a very long operating life, an order for three replacement connecting bellows was placed with them.

Sealol, the original bellows coolant pump supplier, completed the study to establish an installed length for their bellows pump that should allow the pump to meet the design life requirements. Because the length set could not be accommodated in the blood pump and the Metal Bellows pump appears to be capable of long life cycling, it was decided to terminate the Sealol development effort.

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 A.3-234. However, after three hours of operating, the axial crack shown in Figure A.3-235 developed in one of the flexible coolant tubes. After the cracked tube had been replaced and another six 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 (as shown previously in Figure A.3-233). 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. This failure has been discussed previously in Section A.3.4.9. 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. Based on a successful 30 day life test of a similar bellows, it seems almost certain that the bushing failure caused the bellows failure.

Figure A.3-234. Bellows Pump Coolant Circulating System Test Setup

Figure A.3-235. Flexible Coolant Tube with Axial Crack

Thermal Performance Testing

Thermal performance testing of the bellows coolant pump system in a blood pump with an electric motor drive was set up using the "barrel" simplified mock loop as a heat sink and load for the blood pump. Heat was supplied by an electric heater in the mockup IVBM heat exchanger. Coolant connections were made using 1/8 in. ID PVC tubing. Expanded polyethylene tape was used to insulate the coolant lines, and the heat exchanger was insulated with polystyrene foam. Fiber insulation was used on the blood pump and on the upper end of the dummy heat exchanger. In order to facilitate filling of the cooling system and to permit flow measurements, two tees were inserted into the cooling lines, and a pair of auxiliary lines was provided. For flow measurements, the flow was diverted into a beaker through one of the auxiliary lines with suction through the other from another beaker. During this period, a hemostat was used to close the line between the two tees, blocking flow through the loop. For thermal testing, the auxiliary lines were blocked using hemostats and the hemostat between the auxiliary lines and the coolant system was insulated with fiber insulation.

Figure A.3-236 shows the test setup. The dummy heat exchanger is in the foreground with the blood pump directly behind it. The flexible shaft connected to a sealed pillow block goes off toward the right of the figure. The electric motor driving the pump does not appear in the figure. In the right foreground is the beaker providing water for filling the system and a source of water for flow measurements. The front end of the simplified mock loop appears in the left background.

Temperature measurements of the inlet and outlet to the auxiliary heat exchanger, at the surface of the blood pump, in the water filling the mock loop and the ambient temperature were recorded. The ambient air temperature was taken with a thermometer; all other measurements were taken using copper-constantan thermocouples, and an auxiliary thermometer was provided in addition to the thermocouple in the mock loop.

Tests were run with 20, 25 and 30 watts of electric heat input. During testing, the blood pump was operated at 120 beats per minute. After starting the experiment, the thermal losses from the surface of the heat exchanger were found to

Figure A.3-236. IVBM Bellows Coolant Loop Test Setup

be much higher than expected. Heat input to the coolant system was, therefore, calculated from the coolant temperature rise across the heater and its flow rate. The temperature drop across the coolant-blood pump interface was computed using the average of the heat exchanger inlet and outlet temperatures and the blood pump surface temperature. This data is plotted in Figure A.3-237. For each value of electrical heat input, a large spread of heat input to the system can be observed. This is due to the fact that the temperature rise across the heat exchanger is small (2 to 3°F), and small errors in measurement of this difference lead to fairly large errors in calculated heat input. A least square curve fit was performed, assuming that the temperature drop across the coolant-blood pump interface is proportional to the heat input. The resulting value for the thermal impedance of the interface is 0.51°F/watt, and the coolant temperature drop as a function of heat input using this value is plotted in Figure A.3-237. This is slightly higher than the predicted value of 0.48°F/watt.

Figure A.3-238 shows a similar plot for the blood pump diaphragm temperature drop. The computed thermal impedance of this interface is 0.61°F/watt compared with values of 0.5°F/watt measured in earlier tests.

Using the computed values of thermal impedance, the temperature drop across the diaphragm is expected to be 12°F at the design heat input of 20 watts, and the drop between the coolant system and blood pump 10°F. Adding 1°F (one-half of the temperature rise across the heat exchanger) gives a peak coolant temperature 23°F higher than the blood temperature.

In operation inside the body, the losses from the surface of the blood pump and tubing can be expected to be somewhat higher than were encountered in the laboratory. In the laboratory situation, the combined radiation and convection heat transfer coefficient on the tube surface can be estimated between 1 and 2 Btu/hr-ft²-°F, depending on the emissivity of the insulation. This represents a thermal impedance approximately equal to that provided by the insulation. In an animal, the surface heat transfer coefficients at the interface between the insulation and tissue must be about 8 Btu/hr-ft²-°F to support a heat flux of 0.01 W/cm² with a 4°F temperature drop.

Figure A.3-237. Coolant-Blood Pump Temperature Drop

Figure A.3-238. Temperature Drop Across Blood Pump Diaphragm

Since the surface losses are expected to be higher when the cooling system is implanted than in the bench test, the heat fluxes across both interfaces will be reduced for an implant experiment, and the temperature drops reduced accordingly. This could reduce the peak temperature by as much as 6°F.

Concluding Life Test

The system was returned to life test and after 210 additional hours of operation had been accumulated, the test was terminated because leakage of the blood pump mechanism oil into the coolant loop was observed. When the system components were examined, it was established that the mechanism oil had leaked into the coolant loop through a crack in one of the pumping bellows. The damaged bellows was then removed from the system and testing was continued with a single pumping bellows and external check valves. However, after 376 additional hours of testing, the remaining bellows failed. Because all the bellows procured exhibited poor fatigue life, bellows coolant circulation system investigations were discontinued.

3.5.2 Thermal Oscillation Pump System

A primary candidate for the artificial heart waste heat transport/rejection system is the thermal oscillation pump (TOP). The operational characteristics of the TOP system can be described by viewing the system schematic in Figure A.3-239. Based on observation of this simple experimental loop, the system functions as follows:

- Heat rejected by the thermal converter begins to vaporize liquid working fluid in the TOP system of the converter/TOP interface (the vaporizer).
- The resulting vapors are transported to a heat exchanger (the condenser) integrated with the blood pump housing and condensed back to a liquid state. The rejected heat is absorbed by the blood and is eventually dissipated throughout the body.
- The condenser working fluid is then forced through check valve B into the pressurizer where it is accumulated and stored.
- This continues until the rate of condensation is greater than the rate of vaporization. When this condition is reached, the pressure in the vaporizer is less than the pressure in the pressurizer. Liquid is then forced through check valve A from the pressurizer into the vaporizer to complete one cycle.

Figure A.3-239. Thermal Oscillation Pump Concept Schematic

For subcooled boiling, the time for one cycle will be defined as the slugging time constant. The slugging time constant can be estimated from the following equation (neglecting sensible heat).

$$\tau = \frac{V_v \rho_l H_{lv}}{Q} \quad (A-1)$$

where:

V_v = Volume of vaporizer

ρ_l = Density liquid

H_{lv} = Heat of vaporization of the fluid

Q = Rate of heat input

This equation assumes the vaporizer boils dry. This isn't necessarily true for the nucleate boiling mode of operation. The slugging time constant can be used as a measurement of the check valves opening and closing for the particular working fluid under investigation.

As the working fluid vaporizes in the TOP/thermal converter heat exchanger and forces liquid into the pressurizer, the pressure level in the TOP loop increases. This in turn increases the saturation temperature. This increase can be approximated by:

$$\Delta T_s \cong \frac{dT_s}{dP} (P - P_o) \quad (A-2)$$

where:

T_s = Saturation temperature

P = Pressurizer pressure

P_o = Initial pressurizer pressure

For an ideal pressurizer gas and assuming reversible adiabatic expansion:

$$P = P_o \left(\frac{V_o}{V} \right)^k \quad (A-3)$$

where:

$V = V_o + \Delta V$

where:

V = Gas volume in pressurizer

V_o = Initial gas volume in pressurizer

ΔV = Volume of vaporizer plus line from vaporizer to condenser plus the volume of the vapor in the condenser

k = Ratio of specific heats for the gas in the pressurizer

Substituting in (A-2) above, gives:

$$\Delta T_s \cong \frac{dT_s}{dP} P_o \left[\left(\frac{V_o}{V_o - \Delta V} \right)^k - 1 \right] \quad (A-4)$$

The product of $P_0 dT_s/dP$ is a fluid property. In order to minimize the change in saturation temperature during evaporation, the initial volume (V_0) of gas in the pressurizer should be large and the total vapor volume (ΔV) in the vaporizer, vaporizer line and condenser should be small.

The pressure drop in the vapor line (from vaporizer to condenser) produces a drop in the saturation temperature. By assuming an acceptable temperature drop, which was estimated to be 0.6°K for the TOP vapor line, the inside tube diameter for the TOP system was calculated. Here, the pressure drop is assumed to be:

$$\Delta P = \frac{\frac{\Delta T}{dT_s}}{\frac{dP}{dP}} \quad (\text{A-5})$$

Knowing the pressure drop, the inside tube diameter can then be estimated for either laminar or turbulent flow by using the Fanning friction factor equation for circular pipes. Here:

$$\Delta P = 4f \frac{L}{D} \frac{v^2 \rho}{2} \quad \text{where} \quad \begin{array}{ll} \rho - \text{fluid density} & D - \text{tube diameter} \\ L - \text{tube length} & V - \text{fluid velocity} \end{array} \quad (\text{A-6})$$

For laminar flow in long tubes

$$f = \frac{16}{\text{Re}} \quad (\text{A-7})$$

$$\text{Re} = DV\rho/\mu. \quad \text{also } W = \rho \frac{\pi}{4} D^2 V$$

Substituting into Equation (A-5) and solving for tube diameter gives:

$$D = \sqrt[4]{\frac{64 \mu L W}{\Delta P \frac{\pi}{2} \rho}} \quad \text{laminar flow} \quad (\text{A-8})$$

For turbulent flow in tubes:

$$f = \frac{0.079}{\text{Re}^{1/4}} \quad (\text{A-9})$$

This equation, known as the Blasius formula, is good up to about $\text{Re} = 10^5$.

Substituting into Equation (A-6) and solving for tube diameter gives:

$$D = 4.75 \sqrt{\frac{0.316 L \mu^{0.25} W^{1.75}}{\left(\frac{\pi}{4}\right)^{1.75} 2 \Delta P \rho}} \quad \text{Turbulent flow} \quad (\text{A-10})$$

NOTE: Mass flow in Equations (A-9) and (A-10) can be calculated by:

$$W = \frac{Q}{H_{lv}} \quad (\text{A-11})$$

In most cases, the Reynold's number will be in the transition range. The larger of the two calculated diameters was conservatively used.

Once the tube diameter for TOP is determined, the vaporizer, lines and condenser volume are estimated. With the volumes determined, the increase in saturation temperature caused by the finite pressurizer volume is calculated from Equation (A-4). The respective temperature difference is then added to the base blood temperature to arrive at maximum fluid temperature in the TOP system at the thermal converter interface.

Table A.3-40 gives a comparison of some potential fluids for the thermal pump heat transport system. The choice of these fluids for investigation was based on: first, water was selected for base comparison; trichlorotrifluorethane, methylene chloride, methyl formate, methanol, and benzene were selected because of their vapor pressure levels (34 to 210 kPa) at 322°K; and propane and ammonia were selected because in fluid cycles and powerplants with bottoming cycles, these fluids required the least flow areas. The vaporizer and condenser fluid volumes in Table A.3-40 were calculated using an appropriate calculated tubing diameter and assuming four wraps of tubing around the thermal converter cooler and four wraps around the pump housing. This length of tubing for the vaporizer and condenser (43.2 cm for four wraps) allows for sufficient heat transfer area contact. Also, the pressurizer volume was assumed to be 32.8 cubic centimeters of initial gas volume for these comparisons. Keep in mind that a large pressurizer volume or small vapor line volume minimizes the effective increase in vaporization temperature,

TABLE A.3-40

COMPARISON OF FLUIDS FOR THERMAL PUMP HEAT TRANSPORT SYSTEM

		WATER H ₂ O	TRICHLOROTRI- FLUOROETHANE C ₂ Cl ₃ F ₃	METHYLENE CHLORIDE CH ₂ Cl ₂	METHYL FORMATE	PROPANE	BENZENE	AMMONIA	METHANOL
τ/V_V	$\frac{\text{sec}}{\text{cm}^3}$	94.3	9.1	16.6	16.3	5.2	14.1	23.9	33.4
$P_o \frac{dT_{\text{sat}}}{dp}$	$^{\circ}\text{K}$	274.7	288.7	299.8	286.4	301.1	281.3	293.3	294.0
D	cm	0.2733	0.6553	0.5055	0.3912	0.2108	0.3835	0.1351	0.1438
V_V	cm ³	2.802	16.141	9.603	5.752	1.655	5.522	0.688	0.770
V_C	cm ³	5.604	32.282	19.173	11.504	3.343	11.045	1.377	1.557
ΔV	cm ³	10.209	58.666	34.904	20.975	6.063	20.156	2.491	2.819
τ	sec	264.0	81.1	159.0	93.5	8.60	78.0	16.4	25.7
Blood Temperature	$^{\circ}\text{K}$	310.1	--	--	--	310.1	--	310.1	310.1
Blood to Sperm Oil	$\Delta T^{\circ}\text{K}$	6.9	--	--	--	6.9	--	6.9	6.9
Sperm Oil to Blood Pump Body	$\Delta T^{\circ}\text{K}$	4.3	--	--	--	4.3	--	4.3	4.3
Condensing ΔT	$\Delta T^{\circ}\text{K}$	0.6	--	--	--	5.3	--	1.7	3.2
ΔT Due to ΔP	$^{\circ}\text{K}$	0.6	--	--	--	0.6	--	0.6	0.6
Pressurizer Penalty ΔT	$^{\circ}\text{K}$	11.8	--	--	--	13.8	--	4.1	4.7
Maximum Fluid Temperature	$^{\circ}\text{K}$	334.3	--	--	--	341.0	--	327.7	329.8

 V_V - Vaporizer Volume V_C - Condenser Volume ΔV - See Equation A-3.

as shown in Equation (A-4). Note: In Table A.3-40 the choice of a working fluid for TOPS can change the slugging time constant by a factor of 20 for the subcooled boiling mode.

Thermal performance characteristic tests of the TOP system were made to evaluate the effect of system orientation, pressurizer volume, type of fluid, power level, pressure level, and sink temperature on the maximum working fluid operating temperature. The TOP components were assembled as shown in Figure A.3-240. A thermocouple was added at location H on the surface of the tube approximately midway between the heater and the heat sink. The output of this thermocouple was recorded on a strip chart.

Copper tubing having 0.17 cm ID was used. In the operation of the test loop, three modes of operation were identified. The first mode (the subcooled boiling mode) was characterized primarily by the fact that the temperature of the tube between the heater and the heat sink (Location H in Figure A.3-240) periodically dropped below the fluid saturation temperature. This indicated complete collapse of the vapor phase with subcooled liquid being drawn through the heater into the condenser. A typical strip chart trace of thermocouple H when the test loop was operating in this mode is shown in the lower portion of Figure A.3-241. The working fluid in the loop is methanol. The fluid temperature oscillates approximately 2.8°K in this mode.

In the second mode of operation (called the nucleate boiling mode), the working fluid temperature remains essentially constant. A trace of thermocouple H in this mode for methanol is shown in the upper portion of Figure A.3-241. The temperature oscillates approximately 0.6°K in this mode.

The third mode of operation (called the film boiling mode) occurred only at very high powers. In this mode the temperature at location H was always greater than the fluid saturation temperature. This mode of operation is undesirable since the working fluid temperature would be difficult to predict and control. Transition from the subcooled boiling mode into nucleate boiling could be accomplished three different ways. With fixed coolant water temperature, increasing the power level would produce the transition as shown in Figure A.3-242. Here the initial heater power level is 15 watts and the subcooled spike appears approximately every 180 seconds. By increasing the power level to 17.5 watts, the subcooled spike period increases to about 300 seconds and upon further increasing power to 20 watts, the operational mode changes to the nucleate boiling mode.

Figure A.3-240. Thermal Oscillating Pump (TOP) Heat Transport System Test Configuration

Figure A.3-241. Methanol Subcooled Boiling Mode with Transition into Nucleate Boiling Mode (Power Level Increasing)

Figure A.3-242. Subcooled Boiling Mode with Transition into Nucleate Boiling Mode (Coolant Water Temperature Increasing)

With a fixed power level, increasing the coolant water temperature to within 3.9 to 5.6°K of the vaporizer temperature gives an operational mode transition shown in Figure A.3-242. Also, by reducing the pressurizer gas volume down to about 4.9 cubic centimeters forces a change in the system mode of operation from subcooled boiling to nucleate boiling. However, in the nucleate boiling mode the volume of the pressurizer gas was varied from infinity (open to atmosphere) down to 2.5 cubic centimeters with no detectable difference in thermal performance.

Increasing the pressure level in the Top test loop increases the working fluid saturation temperature. For methanol, a pressure level increase of 37.1 kPa gives a vapor temperature increase of 12.2°K.

During most of the testing of TOP test loop, methanol was used for the working fluid (normal boiling point 337.5°K. Acetone was also tried due to its more desirable lower normal boiling point of 328.7°K. A comparison of the operating vapor temperature (location H) at atmospheric pressure was made for the two fluids. The tube temperature is approximately 6.7°K cooler when operating with acetone rather than methanol. However, the flow rate of acetone is more than double that of methanol because the heat of vaporization of acetone is less than half of that of methanol. The higher flow rate may be more advantageous in the TOP system where flow actuated check valves are used. Also, a reduction in working fluid temperature would result in reduction of insulation requirements.

No significant difference in the thermal performance of the test loop could be detected due to orientation of the components. The heater was placed 15.24 cm above the level of the condenser and rotated from the reference position to its side, and then inverted while the operating parameters remained essentially constant.

The test condenser was changed from the flat plate configuration to a plastic tube (1/8 in. ID) coiled inside a beaker, so that the vapor length could be observed. The beaker was filled with water and the temperature of the water was controlled. The condenser surface area required as a function of power level could then be determined. The range of condenser surface area versus power level is shown in Figure A.3-243. A range of surface area is given due to the oscillating nature of the TOP system.

Figure A.3-243. Range of Condenser Area Versus Power Level

The TOP system was also retrofitted onto an artificial heart pump. Tests were run using a simulated thermal converter as an evaporator. This existed as a coiled tube maintained at a fixed power output and is schematically shown in Figure A.3-244. The TOP condenser/pump interface consisted of a 3-loop cooling coil over the metal pump body utilizing high thermal conductivity grease to reduce interface thermal resistances. The vapor lines were well insulated to prevent a significant heat loss.

The temperature of the fluid entering the condenser (exiting the evaporator) is at saturation temperature and is, therefore, independent of the power level and condenser water temperature. However, the fluid exiting the condenser is a subcooled liquid and is affected by the power level and condenser temperature.

It is assumed that the temperature of the fluid exiting the condenser is an indication of the condenser tube temperature. Plots of the temperature difference between the fluid exiting the condenser and the mock loop cooling water versus input power are shown in Figure A.3-245 for acetone and in Figure A.3-246 for methanol. These tests were run using the coil-type evaporator described previously. The slope of the least squares fit to the acetone data gives a value of $0.28^{\circ}\text{K}/\text{watt}$ and for the methanol data gives a value of $0.31^{\circ}\text{K}/\text{watt}$. These values compare favorably with design values allowing for a small thermal impedance difference from the coil to the internal lubricating oil of the blood pump. Assuming most heat loss occurs in the vapor line of the TOP system and is a function of the saturation temperature of the fluid, it is, therefore, independent of the power level. Extrapolating the least squares fit to acetone (saturation temperature 329.2°K), to zero temperature difference (Figure A.3-246) indicates a loss of about 6.5 watts. For methanol (Figure A.3-245, saturation temperature 329.2°K), the heat loss appears to be about 11 watts. These heat losses are significant when compared to the input power. However, these lines can be insulated allowing a heat flux of approximately $\leq 0.03 \text{ W}/\text{cm}^2$. The heat loss would be ≤ 1.1 watts.

A possible design of the TOP system at the thermal converter interface is schematically shown in Figure A.3-247. Here, liquid working fluid enters the upper portion of the TOP clamp on the system where waste heat is picked up (24 watts). Resulting outlet vapors then transport the waste heat to the heart pump. The working fluid is condensed and returned for the next cycle.

Figure A.3-245. Temperature Difference from the Condenser Exit Fluid (Methanol) to Water in the Mock Loop Versus Heater Power

Figure A.3-244. TOP Component Board

Figure A.3-246. Temperature Difference from the Condenser Exit Fluid (Acetone) to Water in Mock Loop Versus Heater Power

Figure A.3-247. Schematic of TOP System/Implantable Thermal Converter Interface

Proof-of-principle operation of the TOP system above used methanol and acetone as the working fluids. Although these fluids performed satisfactorily in the TOP system, more desirable ones may be available. In order to explore a wide range of other potential working fluids for application in future TOP testing, an extensive review of the chemical literature was made. The choice of additional potential working fluids was made based on the following desirable fluid characteristics:

- The saturation temperature of the fluid should be $>115^{\circ}\text{F}$ but $\leq 120^{\circ}\text{F}$ at approximately one atmosphere.
- The fluid's toxicity and flammability characteristics should permit ease of handling in the laboratory and in manufacturing facilities.
- The working fluid should be noncorrosive in the TOP environment.

The requirements that the working fluid operating temperature be above 115°F was based on the thermal impedance of the condenser system (TOP interface with the pump). The condenser wall temperature could be $110 \rightarrow 115^{\circ}\text{F}$ (the oil is $\approx 110^{\circ}\text{F}$). This lower temperature limit can also be determined from an approximate value of $0.5^{\circ}\text{F}/\text{watt}$ obtained in previous tests for the thermal impedance from the oil to the water in the mock loop. There is only a small thermal impedance from the condenser to the oil ($\sim 0.1^{\circ}\text{F}/\text{watt}$).

The upper TOP operating limit of 120°F is a performance specification established by Philips Laboratories. This temperature limit of the working fluid is set by roll sock materials life considerations. However, it should be pointed out that this upper limit is only possible under normal body temperatures. If an implant patient had a high fever, the TOP operating temperature could increase as much as 7°F . This higher temperature would only last for a short period of time (until the patient's fever is reduced) and should not affect the roll socks long term materials life. Design of the TOP system, however, should address a body temperature increase by allowing working fluid vapor pressure to increase. This would raise the boiling point of the working fluid and prevent film boiling and possible vapor lock. Increasing system pressure can be made by adding working fluid to the pressurizer gas volume. Any body temperature increase would be experienced by the fluid in the pressurizer resulting in a higher vapor pressure in the gas volume. This coupled with the pressurizer diaphragm tension, would permit TOP working fluid pressure to increase, thus raising the boiling point.

The ideal operating pressure of the waste heat removal system would be around one atmosphere. Here, the differential pressure or pressure drop between TOP and the body would be small. This would minimize the possibility of leakage. A slight TOP system vacuum would even be more desirable. If a small leak should occur over the operating life the system, body fluids would enter the system rather than loss of the working fluid to the body. This case would still permit (depending on the fluid) sufficient working fluid for waste heat removal. However, with a pressurized system, any outleakage may result in unacceptably low boiling temperatures and mixing of the working fluid with body fluids. This must be considered in choosing a fluid.

Fluid toxicity and flammability characteristics were addressed mainly from a safety point of view in the laboratory. Some of the fluids had boiling point temperatures around 120°F but were so toxic and reactive that handling would require unusually expensive laboratory equipment and present a real danger to personnel involved in testing and manufacturing the TOP system. Also, highly corrosive fluids were obviously ruled out based upon the criteria they would not meet the TOP operating life expectancy and might present a real leakage concern.

Tables A.3-41 and A.3-42 list all the various fluids considered for future TOP testing. These candidate fluids were evaluated based upon their boiling point temperature. Table A.3-42 lists those fluids considered acceptable for future testing. Table A.3-41 lists those fluids considered but for various reasons, as indicated, were rejected. Based upon the extensive literature search of inorganic property data, these tables list only the potential working fluids within the operating specifications outlined above.

As indicated in Table A-42, nine fluids and four azeotropes have acceptable fluid characteristics for possible application in the TOP heat transport system. As discussed above, these fluids have the desired boiling point temperature, are reasonably safe in handling and are stable. The azeotropes have been considered as a means to decrease the boiling point temperatures of otherwise desirable fluids such as methanol and acetone. These mixtures are listed in Table A.3-42.

TABLE A.3-41

OTHER POSSIBLE WORKING FLUIDS CONSIDERED FOR THE
ARTIFICIAL HEART THERMAL OSCILLATING PUMP (TOP) HEAT TRANSPORT SYSTEM
BUT REJECTED

FLUID NAME	MOLECULAR WEIGHT	SPECIFIC GRAVITY	BOILING POINT °C/°F	HEAT OF VAPORIZATION Btu/lb	GENERAL DESCRIPTION
Carbon Disulfide	76.1	1.261	46.2/115.2	152	Boiling point temperature too low; poisonous, harmful vapors, highly volatile.
Acrolein	56.1	0.843	52.5/126.5	≈150	Poisonous; extremely dangerous reaction with eyes; very reactive polymerizing readily.
1 - Hexene	84.2	0.686	64.1/147.4	≈150	Boiling point temperature too high; could possibly use under vacuum.
2 - Pentyne	68.1	0.687	56.0/132.8	≈150	Boiling point temperature too high; could possibly use under vacuum.
n - Propylchloride	78.5	0.890	46.6/115.9	≈150	Boiling point temperature too low; could possibly use under pressure.
Isopropylbromide	123.0	1.310	59.6/139.3	≈150	Boiling point temperature too high; could possibly use under vacuum.
Ethyl Formate	74.0	0.924	54.3/129.7	175	Extremely flammable; unstable liquid; gradual decomposition in water.
Methyl Acetate	74.0	0.924	57.1/134.8	206	Boiling point temperature too high; could possibly use under vacuum.
Diethylamine	73.1	0.711	56.0/132.8	164	Extremely flammable; could cause skin burns; boiling point temperature too high.
Tert-butylamine	73.1	0.696	43.8/110.8	≈150	Boiling point temperature too low; reacts with skin.
Silicon Tetrachloride	169.9	1.480	57.6/135.7	65	Decomposes in water; quite corrosive to metal; suffocating odor; boiling point temperature too high.
Sulfur Dichloride	103.0	1.621	59.0/138.0	≈100	Boiling point temperature too high; decomposes upon rapid heating above 40°C; very corrosive; causes burns.
Sulfur Trioxide	80.1	1.923	44.6/112.3	213	Boiling point temperature too low; highly corrosive.
Ethanoyl Chloride	78.5	1.105	51.0/123.8	≈150	Flammable fuming liquid; irritating to eyes and violent reaction with water.
Acrylic Aldehyde	56.1	0.841	52.5/126.5	≈160	Very reactive and dangerous; violent action on eyes; poisonous.
3 - Chloro-Propane - 1	76.5	0.938	44.6/112.3	≈150	Boiling point temperature too low.
2 Cl ₂ - 2 Me - Propane	92.6	0.847	51.0/123.8	≈150	Unstable.
Chloro-buta - 1, 3 diene	88.5	0.958	59.4/138.9	≈150	Boiling point temperature too high.
Zinc Dimethyl	95.5	1.386	46.0/114.8	----	Boiling point temperature too low; dangerous to handle; may burn when exposed to air

TABLE A.3-42

COMPARISON OF FLUIDS FOR POSSIBLE APPLICATION IN
THE ARTIFICIAL HEART THERMAL OSCILLATING PUMP (TOP) HEAT TRANSPORT SYSTEM

FLUID NAME	MOLECULAR WEIGHT	SPECIFIC GRAVITY	BOILING POINT °C/°F	HEAT OF VAPORIZATION Btu/lb	GENERAL DESCRIPTION
Trichlorotrifluoroethane	187.4	1.420	47.7/117.9	64	Very stable; refrigerant in process water and brine cooling; used in aerosol products; Freon 113; $C_2Cl_3F_3$; solvent.
Propionaldehyde	58.1	0.807	48.8/119.8	~170	Soluble in water; used as a disinfectant; flammable; Propionic aldehyde; C_2H_5CHO ; suffocating odor.
Acetone	58.1	0.791	56.1/133.0	243	Solvent; extremely flammable; miscible with water; vacuum to get boiling point to 120°F; dimethylketone; CH_3COCH_3 ; TOP tested DRM DA-0252.
Dichlorethylene (trans)	97.0	1.257	48.4/119.1	154	Slightly toxic; decomposes slowly on exposure to air, light and moisture; flammable, Acetylene dichloride; $CHCl = CHCl$.
Methanol	32.0	0.792	64.5/148.1	474	Poisonous; soluble in water; flammable; Methyl Alcohol; CH_3OH ; TOP tested DRM DA-0252; very high heat of vaporization; vacuum to get boiling point to 120°F; vapors harmful.
n - Propylamine	59.1	0.718	48.7/119.7	~180	Soluble in water; flammable; $CH_3(CH_2)_2NH_2$.
Cyclopentane	70.1	0.745	49.3/120.7	~160	Insoluble in water; flammable; Pentamethylene; C_5H_{10} .
Neohexane	86.2	0.649	49.7/121.5	~150	Used high octane rated fuels; flammable; Dimethylbutane; C_6H_{14} .
Methanol - 1, 1 Dichloroethane	----	~1.047	49.1/120.4	~240	Minimum boiling azeotrope mixture of 28.5 wt-% Methyl Alcohol; Methyl Alcohol properties described above; 1, 1 dichloroethane otherwise ethylidene chloride is nonflammable and slightly soluble in water; CH_3CHCl_2 B.P. is 57.3°C.
Methanol - Hexane (N)	----	~0.725	50.6/123.1	~300	Minimum boiling azeotrope mixture of 51.0 wt-% Methyl Alcohol properties described above; Hexane (N) C_6H_{14} is highly flammable and insoluble in water; B.P. C_6H_{14} is 68.7°C.
Methanol - Benzene	----	~0.821	53.8/128.8	~370	Minimum boiling azeotrope mixture of 61.4 wt-% Methyl Alcohol; Methyl Alcohol properties described above; Benzene C_6H_6 is flammable and slightly soluble in water; vapors are harmful and poisonous. B.P. C_6H_6 is 80.1°C.
Acetone - Diethylamine	----	~0.740	51.5/125.0	~190	Minimum boiling azeotrope mixture of 43.5 wt-% acetone; acetone properties described above; diethylamine $(C_2H_5)_2NH$ is extremely flammable and miscible with water; B.P. $(C_2H_5)_2NH$ is 55.5°C.
Water	18.0	1.000	100/212.0	1027	Very high heat of vaporization; must have vacuum to 1 or 2 psig to have 120°F B.P.; high vapor velocities.

TOP testing of various working fluids used an arrangement having a pressurizer as schematically shown in Figure A.3-239. This pressurizer can be quite large ($0.1 \rightarrow 0.4\text{-in}^3$) depending on the working fluid used. A second TOP design concept was developed that eliminates the need for a pressurizer. This concept incorporates a second loop that can easily be accommodated at both the thermal converter and heat pump interfaces. It also affords some redundancy in design. Here each loop acts as the pressurizer for the other. A schematic of this dual loop system is shown in Figure A.3-248.

A.3.5.3 Heat Pipes

Replacing the pumped coolant loop by a heat pipe can eliminate the need to supply mechanical power to drive the coolant loop, and increase the overall efficiency of the system. As a preliminary step in evaluating the potential of heat pipes for this application, three $1/4$ in. OD by 10 in. long methanol filled flexible heat pipes were initially tested. These heat pipes were commercially available and not designed for heart use or loads. Of the three tested, only one performed approximately as rated. One of the remaining two had a larger amount of noncondensable gas, and had essentially zero heat transport capability. The other reached its transport limit at approximately one-half of its rating point. The reason for its limited capability is not clear. After testing, one of the pipes was filled with plastic to immobilize its internal components, then sectioned to gain a better understanding of its internal construction. Figures A.3-249, A.3-250 and A.3-251 are longitudinal, transverse and oblique section of this pipe. The main portion of the wick can be clearly seen at the bottom portion of Figures A.3-249 and A.3-250 and the right side of Figure A.3-251. This part of the wick provides a return path for the working fluid from the condenser to the evaporator. The single layer of screen covering the inner circumference of the bellows forming the outer wall of the heat pipe serves to transport the working fluid circumferentially around the pipe wall. A spring is used to hold the wick components in place against the inner wall of the heat pipe.

Figure A.3-248. Dual Loop Thermal Oscillating Pump (TOP) System Schematic (No Pressurizer)

Figure A.3-249. Longitudinal Section of Hughes Flexible Heat Pipe

Figure A.3-250. Transverse Section of Hughes Flexible Heat Pipe

The pipes tested were not designed for the conditions which would be required for use with an artificial heart. Their heat transport capability is not high enough for this application, particularly when oriented against gravity. To obtain meaningful data to be used to evaluate the feasibility of a heat pipe rejection system, a vendor of heat pipes was employed to construct a breadboard heat pipe to specifications representative of the requirements for an artificial heart.

The Dynatherm Corporation of Cockeysville, Maryland initiated investigations leading to the construction of a "breadboard" heat pipe heat rejection system. This system was used to evaluate the suitability of heat pipes for transporting waste heat between the thermal converter and blood pump.

Figure A.3-251. Oblique Section of Hughes Flexible Heat Pipe

Design analyses and wicking experiments on flexible heat pipes included material selection, detailed wick and heat pipe envelope analysis, selection of the corrugated tubing for the flexible portion of the heat pipe, and the detailed design of the breadboard pipe.

The heat transport requirement of 20 watts in any orientation resulted in the selection of a copper-water heat pipe system. Water has the highest heat transport capability and static wicking height in the specified temperature range. In addition, experience has demonstrated that a copper-water system is very stable over long operating periods.

A detailed design analysis of the wick was performed to determine the smallest heat pipe envelope area capable of meeting specifications. An axially graded wick was chosen as the reference design. A development effort was undertaken to fabricate a graded wick using a commercially available copper foam. This effort consisted of compressing the foam to change its wicking characteristics. This process, however, succeeded in decreasing permeability more than pumping radius, and because of this adverse effect and the questionable flexibility of the compressed configuration, the use of sintered foams was discarded. A segmented wick design, consisting of 150 and 400-mesh screen, was shown by analysis to be capable of meeting specifications. To maintain wick flexibility, the wick consists of several individual strands which formed by sintering individual screen layers. Initial attempts to sinter phosphor-bronze screen were successful, and the breadboard heat pipe was fabricated.

The assembled unit included evaporator and condenser saddles and a 6 in. flexible section. A photograph of the assembled unit is shown in Figure A.3-252.

During fabrication of this heat pipe, several steps in the fabrication procedure were breadboarded. The attachment of the flexible tubing to the heat pipe was accomplished by brazing. The wick was ebonized to increase the water-wetting capability of the phosphor-bronze wick. Retainment of the wick, using a retainer spring within the evaporator and condenser sections of the heat pipe, was also breadboarded.

After fabrication was completed, the heat pipe was charged and performance tested in a vertical position with the evaporator above the condenser (negative vertical), with the condenser above the evaporator (positive vertical or reflux mode), and in the horizontal position. Although the heat pipe was always able to carry 10 watts in the negative vertical orientation, the corresponding temperature gradient from the evaporator to the condenser was initially 19.2°F.

Figure A.3-252. Breadboard Heat Pipe

Comparing this to the measured temperature gradient of 9.7°F that existed during the reflux mode, an overcharge condition was indicated. To correct this condition, the pipe was bled to remove some of the excess fluid and then it was retested. The temperature gradient in the negative vertical orientation at a power level of 10 watts was reduced to 15.0°F , and the corresponding temperature gradient in the reflux mode at 10 watts was reduced to 7.7°F . Although the performance capabilities of the heat pipe with this fluid charge were acceptable, the pipe was bled again and then retested. The test results with this final charge are shown in Figure A.3-253. The temperature gradient at 10 watts is 1.3°F higher than at the previous charge level. Also, the bulk of the gradient was found to exist between the evaporator and the adiabatic section, possibly indicating a slight undercharge condition. At this final charge, the heat pipe was also tested with the bellows flexed to a 5 in. radius of curvature. With the evaporator located directly above the condenser, the temperature gradient at 10 watts was 11.5°F , as shown in Figure A.3-254. The heat losses throughout the performance tests were determined through calibration tests to be 2 watts.

With the demonstrated ability of the heat pipe to perform thermally in the straight condition and in flexure, a final design was completed and fabrication of a prototype heat rejection system was carried out.

The heat pipe coolant system, shown in Figure A.3-255, was fabricated by Dynatherm where performance testing of both flexible heat pipes was conducted. The results of these tests are shown in Figure A.3-256 through A.3-259. For each pipe, the tests were performed in the following orientations with the pipe straight:

- The condenser directly below the evaporator (vertical).
- The condenser and evaporator in the same horizontal plane (horizontal).
- The condenser directly above the evaporator (reflux).

In addition, the pipes were tested with the bellows bent to a 5 in. radius. One pipe was tested with the condenser directly below the evaporator and the bend oriented to produce the greatest possible orientation difference. Both pipes were tested with the evaporator and condenser in the same horizontal plane and the bend in the bellows oriented upward and downward.

Figure A.3-253. Thermal Performance of Breadboard Heat Pipe (Straight)

Figure A.3-254. Thermal Performance of Breadboard Heat Pipe
(5 in Radius Bend in Bellows)

Figure A.3-255. Prototype Heat Pipe Heat Rejection System

Figure A.3-256. Thermal Performance of Left Hand Heat Pipe with Pipe Straight

Figure A.3-257. Thermal Performance at Left Hand Pipe with Pipe Bent

Figure A.3-258. Thermal Performance of Right Hand Heat Pipe with Pipe Straight

Figure A.3-259. Thermal Performance of Right Hand Heat Pipe with Pipe Bent

The results of these tests indicated that the copper-water heat pipe coolant system has the capability of handling the desired power in any orientation with acceptable, if somewhat large, temperature drops between the evaporator and condenser. In all three tests, the heat loss from the surface through the insulation was measured to be approximately 2.5 watts for each pipe. This is approximately twice the design goal for an implantable device, but is not large enough to change the conclusion concerning the adequacy of the system to carry the required load of 20 watts.

Several unexpected phenomena were observed during the course of testing. The first is that both pipes have a lower overall temperature drop when operating against gravity than when operating in the reflux mode. In going from reflux to negative gravity orientation, the evaporator temperature drop increased slightly, but the condenser drop decreased significantly causing the overall temperature drop to decrease. This is just opposite to what would be expected if the effect were due to pooling of the working fluid on the heat transfer surfaces.

The second phenomena observed consists of a slow transient in the evaporator-condenser temperature drop, which was observed on startup of the pipe. This transient was observed even though the mean temperature of the working fluid stabilized in a matter of minutes after startup. In Figures A.3-258 and A.3-259, the solid data points were taken after several hours of operation at high power levels. The open data points represent data obtained during startup.

Although the test data have too much scatter to draw firm conclusions, the general trend indicates the presence of some noncondensable gas in each of the pipes. The thermal impedance mismatch between pipes indicates that the two pipes would not carry equal amounts of heat when connected in parallel to form a complete system. This match probably could be improved and the total temperature drop reduced by eliminating gas trapped in the pipes during filling or evolved after filling.

Structurally, the pipes, especially in the bellows portion, are larger in diameter than desired. In addition, they are stiffer than desired, and most of the bending takes place in the two end convolutions of the flexible portion of the pipes, unless the flexible portion is supported while bending. As a result, the system as designed, cannot meet the lifetime flexure requirements for an implantable device. Careful redesign of the bellows forming the flexible portions of the system should eliminate the mechanical life problem, and might reduce the diameter of this portion of the system slightly.

A.3.5.4 ΔV Pusher Cup Oil System

As an alternative to the bellows coolant pump, a scheme was investigated which utilized the motion of the pusher cups to circulate sperm or jojoba oil as a coolant. In this device, shown in Figure A.3-260, the right pusher cup was made slightly smaller than the left. As the blood pump operates, the internal volume of the blood pump mechanism varies periodically. This volume was connected to a limp bladder through a check valve which allows flow into the bladder only. The bladder was, in turn, connected through the coolant loop and back to the pump mechanism volume through a second check valve which only permits flow into the pump mechanism volume. Therefore, during the half cycle when the pump mechanism volume was contracting, sperm oil was forced into the bladder. During the remaining half cycle, sperm oil was sucked out of the bladder, through the coolant loop and back to the pump mechanism. That portion of the circuit connecting the bladder directly to the pump mechanism was extremely short and designed to offer a very low flow impedance. Therefore, the pressure in the blood pump housing never rose significantly higher than ambient. During the second half cycle, the pressure in the blood pump housing fell significantly below ambient to force the coolant to flow through the relatively high impedance offered by the coolant loop. Therefore, the mean pressure in this region was maintained below ambient, allowing the oil to serve its function of maintaining the ventricle diaphragms in close contact with the pusher cups.

Due to the high viscosity and low specific heat of sperm or jojoba oil when compared with water, larger tubes and a higher temperature rise through the converter can be expected using this system than with the bellows pump system design. The pusher cup delta pump offers several advantages, however, not the

Figure A.3-260. ΔV Pusher Cup Oil System

least of which is its simplicity. The thermal interface between the coolant and sperm oil is eliminated, offsetting the higher temperature rise through the converter. Finally, making the right pusher cup smaller than the left will tend to offset the observed mismatch in the right and left ventricles output curves. This mismatch appears to be related to increased leakage through the valves in the left ventricles that results from the higher pressure drop across these valves.

A pair of modified pusher cups were installed on a blood pump, along with the check valves and compliance chamber necessary to implement tests of the pusher cup delta concept. The blood pump was filled with sperm oil, connected to the mock circulation loop and driven through a ferro-magnetic seal to prevent leakage of the sperm oil.

Over a speed range from 12 to 120 beats per minute, the output flow remained fairly constant but considerably below the design flow rate. At extremely low speeds, the displacement was approximately 90 percent of the design displacement of 2.5 cc. These results led to an hypothesis that the ventricle diaphragms were lifting off the pusher cups, thus absorbing the volume changes generated by the differing sizes of pusher cups. The test was repeated with the mock loop drained and pressurized with air to 100 mm-Hg. In this test, the coolant flow increased to approximately 1/3 of the design flow. A more detailed investigation of this concept should be made for future designs to pinpoint the problem and to design a system capable of meeting the flow requirements.

A.3.5.5 Gear Pump Oil System

At the time the bellows coolant pump was selected for the nuclear system, experience indicated that it would be extremely difficult to construct a gear pump of the size required for circulating coolant with acceptable efficiency. Developments on the oil pump for the thermal converter indicate that considerable progress has been made in this area. This progress, coupled with the difficulties encountered in designing and fabricating the bellows coolant pump, led to the need for re-evaluating a gear pump for use as a coolant pump. As a preliminary step in the re-evaluation, a set of curves relating hydraulic power, temperature rise of the coolant, pressure drop and flow rate for water and sperm oil were derived.

From this analysis, a sperm oil loop with a nominal 6°F temperature rise appears to be a feasible alternative to the water filled bellows pump. This temperature rise is equivalent to a flow of 0.34 in³/sec, and requires 0.05 watts of hydraulic power when the flow is through 3/16 in. ID tubing. For a gear pump with overall efficiency of 25 percent, approximately 0.2 watts of mechanical power would be required from the converter. For comparison, the nuclear system with its bellows pump and water coolant requires an equivalent of 0.06 mechanical watts with a 3.9°F temperature rise, for 25 watts thermal input. Figures A.3-261, A.3-262 and A.3-263 summarize the results of these calculations. Figure A.3-261 indicates the effect of the lower specific heat and density of sperm oil on the temperature rise through the converter for a given flow rate. Clearly, for a given thermal performance, a much higher volumetric flow rate of sperm oil is required than for water. Figure A.3-262 illustrates the relationship between temperature rise and pressure drop for the two fluids. The differing slopes for water and oil and the curvature near the ends is due to the large difference in viscosity between the two fluids. Water was found to have turbulent flow throughout most of the region investigated, approaching laminar for pressure drop of 0.15 psi or less. Throughout most of the range, the "minor" (contractions, expansions, etc.) losses represented a significant portion of the total pressure drop for water. The sperm oil flows were laminar throughout the entire range of investigation. For high flows with oil (low temperature drops) the "minor" losses contributed about 20 percent to the total pressure drop, and were negligible throughout most of the range of flows investigated. Figure A.3-263 shows the relationship between hydraulic power required to circulate the coolant and the temperature rise of the coolant through the converter. Temperature rise was selected as the common parameter relating these curves because of its influence in the rest of the system. In general, increasing temperature rise will increase the insulation required to thermally separate the two coolant tubes. In addition, increasing temperature rise will drive the system peak temperature up, increasing the insulation thickness required to protect the surrounding tissue as well as degrading the performance of the converter somewhat. With respect to this last observation, a direct comparison cannot be made between water and sperm oil on the basis of temperature rise alone. Unless a lubricant is found which would enable the blood pump mechanism to operate in water, the use of water as a coolant will entail an additional thermal interface between the coolant and the sperm oil. Water also would require that the coolant loop be

Figure A.3-261. Temperature Rise through Thermal Converter as a Function of Hydraulic Pumping Power

Figure A.3-262. Temperature Rise through Thermal Converter as a Function of Pressure Drop

Figure A.3-263. Temperature Rise through Thermal Converter Versus Flow Rate

sealed, a difficult problem for rotating components. The scope of this analysis was limited to water and sperm oil.

The flow of both fluids was assumed to be steady with time, and the mean fluid temperature was assumed to be 120°F. The cold end heat exchanger of the thermal converter was incorporated as part of the loop. For purposes of evaluation, the coolant heat load was set at 25 watts. Using a gear pump, the flexible tubes in the blood pump, the crossover pipe and the check valves were eliminated as flow resistances. Water was evaluated to provide a reference using the present 0.11 in. ID coolant lines between the converter and blood pump. Sperm oil performance as a coolant was evaluated for 0.11 in. ID, 3/16 in. ID connecting tubing, as well as for tubing large enough to make losses in the tubing negligible in comparison with the losses through the heat exchanger. In addition to the friction losses in the piping, the "minor" losses due to bends, expansions and exit losses were estimated at two velocity heads in the heat exchanger and associated tubing, and between eight and nine velocity heads in the remainder of the loop. This latter value includes losses in the gear pump, the crankcase cooler, and expansion between the cooler and connecting tubing, as well as exit losses at the blood pump.

Design specifications for the oil coolant pump were prepared. Highlights of these specifications include:

- The pump should be capable of delivering 0.45 in³/sec of oil against a pressure drop of 1.0 psi while the blood pump is operating at 120 bpm
- The pump shall consume less than 0.20 watts of mechanical power while operating under the conditions described above
- The pump shall be incorporated into the right angle gear box housing, and shall not increase the envelope dimensions of the converter-gear box assembly
- The weight of the pump shall be less than 20 grams

The specification includes 0.4 psi allowance for pressure drop through the converter cooler. Using this value, a parametric study of the oil side cold end heat exchanger for the converter was carried out. The heat exchanger consists of two semi-circular sets of cooling fins wrapped around the outside of the converter cooler. Fourteen and sixteen fins, with 0.030 in. wide flow pass-

ages and varying heights were examined, with the results shown in Figures A.3-264. and A.3-265. The design point selected (fin height = 0.145 in., 14 fins) results in a satisfactorily compact heat exchanger, with acceptable hydraulic and thermal performance. The fin thickness for this design is 0.020 in., and the design can be fabricated readily.

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

Two preliminary gear pump designs were completed. The gears are supported by ball bearings in one design in Figure A.3-266 and journal bearings in the other in Figure A.3-267.

A.3.5.6 Other Methods

Methods for dissipating waste heat locally from the outer surface of the thermal converter power source were investigated. The outer surface area of the thermal converter power source appears adequate for waste heat rejection directly to body tissue. Design of the heat rejection system using this approach must ensure uniform distribution of heat to the tissue interface to prevent the formation of hot spots. Four potential concepts for waste heat rejection to tissue were investigated:

- Conductive heat transfer
- Forced convective single phase pumped loop
- Forced convective pumped boiling and condensing loop
- Capillary pumped two-phase loop

The advantages and disadvantages of these concepts are summarized in Table A.3-43. Although a conductive system has some inherent advantages, it is the heaviest of the four concepts. A capillary-pumped, vaporization-condensing loop has several advantages, including relatively low weight and uniform heat flux. However, such a system is relatively complex and its reliability is unknown.

Figure A.3-264. Film Temperature Drop as a Function of Fin Height

Figure A.3-265. Pressure Drop as a Function of Fin Height

Figure A.3-266. Gear Pump Envelope where Gears are Supported by Ball Bearings

GEAR DATA	PITCH DIAMETER	.3333
	NO. OF TEETH	8
	PITCH (DIAMETER)	24
	OUTSIDE DIAMETER	.417
	ROOT DIAMETER	.233
	FACE WIDTH	.370

Figure A.3-267. Gear Pump Envelope where Gears are Supported by Journal Bearings

TABLE A.3-43

COMPARISON OF WASTE HEAT REJECTION CONCEPTS

CONCEPT	DESCRIPTION	ADVANTAGES	DISADVANTAGES
1	Conductive Heat Transfer	Static, simplest design High reliability	Significant temperature gradients Relatively heavy approach
2	Forced Convective Pumped Loop - Single-Phase	Relatively simple, straightforward design	Some temperature gradients Requires power for pumping
3	Forced Convection Pumped Boiling and Condensing Loop	Uniform heat distribution Uniform temperature	Requires power for pumping Relatively complex system Reliability unknown
4	Capillary Pumped Two-Phase Loop	Uniform heat distribution Uniform temperature Requires no power Self-regulated and static system Minimum weight system	Relatively complex design Reliability unknown

A mechanically pumped, boiling-condensing loop has fewer advantages than the capillary pumped loop, but it does have all the disadvantages of the latter. In contrast to these concepts, a mechanically pumped single-phase loop offers the simplest and most straightforward design. For these reasons, efforts were concentrated on the design of a forced convective single-phase loop for waste heat rejection to body tissue. Emphasis was placed on the attainment of uniform heat flux and avoidance of hot spots.

Prior to the design of a forced convective, single-phase, mechanically pumped waste heat rejection system, the following set of design constraints was established:

- Maximum surface heat flux 0.01 watt/cm²
- Maximum tissue interfacial temperature 37.8°C (100°F)
- Maximum engine/heat exchanger interface temperature 57°C (135°F)
- Total waste heat 28 watts
- Maximum total hydraulic pumping power 0.1 watts

The maximum surface heat flux constraint is conservative relative to estimates of the maximum heat flux that can be dissipated by vascularized tissue.* The tissue interface temperature constraint was taken as 37.8°C (100°F).

The engine/heat exchanger interfacial temperature constraint is fixed by engine design. The maximum total waste heat to be rejected through the waste heat exchanger is conservatively set based on a 33.1 watt(t) fuel capsule. The maximum allowed total hydraulic pumping power required was arbitrarily set at 0.1 watt. It was judged to be realistically attainable for a system rejecting waste heat to blood in the blood pump and should be conservative for the present approach because of the relatively large flow cross-sectional areas required to attain relatively uniform heat flux.

*Kallfelz, Francis A., Richard A. Wentworth and K. Bingham Cudy, "Biological Effects of Implanted Nuclear Energy Sources for Artificial Heart Devices," Program Report No. C00-2506-2, Cornell University, U.S. Erda Contract No. E(11-1)-2506.

A proposed waste heat rejection system for dissipating heat locally is shown conceptually in Figure A.3-268. The heat pickup region consists of a single coiled tube wrapped around the Stirling engine outer wall. The cooling fluid (water) is pumped to a plenum at the bottom of the thermal converter where it is distributed through a number of parallel tubes that converge into an upper plenum from which the water returns to the coiled, heat pickup section. Uniform heat flux is approached by: minimizing the coolant temperature change; controlling the thicknesses of the Silastic cover around the entire thermal converter; and using relatively large numbers of parallel tubes in the heat rejection section.

Another proposed waste heat rejection system is to have a large flat plate directly under the skin; the plate surface area should be large enough so that waste heat conducted to the plate and rejected to surrounding body tissue would be ≤ 0.01 watts/cm². Work performed in this area is reported in the Cornell Report referenced on the previous page.

Figure A.3-268 Schematic of Pumped Loop Waste Heat Rejection

A.4.0 TOP ASSEMBLY DRAWINGS

NUCLEAR SYSTEM

ASSEMBLY

(W) DWG. 712J107

STIRLING ENGINE ASSEMBLY

PHILIPS LABS DWG 005311

HOT FINGER ASSEMBLIES

IN VITRO ELECTRICAL

PHILIPS LABS DWG 002844 (Rev.B)

NUCLEAR

PHILIPS LABS DWG 005947 (Rev.B)

IN VIVO ELECTRICAL

PHILIPS LABS DWG SK 605

STIRLING ENGINE - THERMAL INSULATION INTERFACE

(W) DWG 389D316

THERMAL INSULATION

UNSKIRTED CUPS

(W) DWG. 101E643

SKIRTED CUPS

(W) DWG. 101E731

BLOOD PUMP MECHANISM

EQUAL PUSHER CUPS

(W) DWG. 712J315

ΔV PUSHER CUPS

(W) DWG. 712J355

COUPLING MECHANISM

FLEXIBLE SHAFT

(W) DWG. 389D376

RIGHT ANGLE GEAR BOX

(W) DWG. 389D379G01

INTEGRAL MOTOR DRIVEN BLOOD PUMP

ASSEMBLY

(W) DWG. 712J443G01

MECHANISM

(W) DWG. 101E867G01

TORQUE MOTOR

SIERRACIN/MAGNEDYNE-S/M 531-01A

VENTRICLE PUMPING DIAPHRAGM

FIGURE 3-87, C00-3043-30

ABDOMINAL IMPLANTABLE ELECTRIC MOTOR

(W) DWG. 101E743