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STUDY ON AN IMPLANTABLE POWER
SOURCE FOR CIRCULATORY SUPPORT DEVICES

November 1, 1968

Thermo Electron Corporation
Waltham, Massachusetts

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STUDY ON AN IMPLANTABLE POWER SOURCE FOR
CIRCULATORY SUPPORT DEVICES

Prepared by

Thermo Electron Corporation

for

Division of Isotopes Development
U.S. Atomic Energy Commission

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The design of an implantable, radioisotope power source for a circulatory support system requires intensive considerations of heat dissipation, blood pumping, materials compatibility and thermal storage in as general a manner as possible. Thermo Electron, in cooperation with Harvard Medical School, has actively investigated these problems for several years under the sponsorship of the National Heart Institute (NHI)*. This background, as is noted in the text where data from NHI-sponsored programs have been incorporated, has been factored into this report in order to place the system design in better perspective. The use of these data in presenting a comprehensive and self-contained report is gratefully acknowledged.

* Study of the Effects of Additional Endogenous Heat, NHI Contract No. PH43-66-982.

Physiological Effects of Blood Pumps in Animals, NHI Contract No. PH43-67-1123.

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Thermal Energy Storage System, NHI Contract No. PH43-67-1473.

CHAPTER I

INTRODUCTION

"Perhaps most dramatic of all, research efforts are now being directed toward the development of an artificial heart. This challenge--as exciting as any across the entire range of science--is enormously complex. It requires the combined excellence of physicians, engineers, experts in the development of synthetic materials, and many others. But physicians and engineers alike agree that this is not a dream. The goal is feasible; the problems are not insuperable."

-- President's Commission on Heart Disease, Cancer and Stroke, 1964.

This report describes the studies and conceptual design of a radioisotope-powered Rankine-cycle engine which will serve as the power source for an implantable circulatory support system. The studies include an evaluation of candidate radioisotopes, parametric analysis of Rankine cycle performance, blood heat exchanger design, operation of control system, selection of thermal energy storage material, and packaging in vacuum-foil thermal insulation.

The fundamental requirements of the implantable power source are high efficiency, reliability and small size; in other words, its requirements differ little from those of most other advanced energy-conversion systems. It does differ greatly, however, in the severity of these requirements and in the nature of the environment in which it must function. Interaction of a complex mechanical system with a living organism has never been attempted to this extent and for so long

a period of automated operation. As usual, these requirements are mutually conflicting, and the designers must reach certain compromises which, in a system of this complexity, cannot always be derived on the basis of quantitative analysis, but must sometimes depend on engineering judgment.

The special nature of this power source and the complexity of its function permit little margin for error. It is therefore imperative to base decisions on experimental data whenever possible and to accept high-risk approaches only when no acceptable alternative exists. This design philosophy has not excluded imaginative solutions and novel approaches to the technical problems encountered. It has capitalized, however, on Thermo Electron's experience of the last five years with Rankine-cycle engines and with almost all circulatory support system components. This study included not only the design of the power supply, but also its coupling to the blood pump. Considerable effort has also been devoted to the evaluation of control techniques which allow efficient utilization of the available hydraulic power by the blood pump.

The basic concept of the circulatory system is illustrated in Figure I-1. The heat comes from radioactive decay. The radioisotope inventory is minimized by using thermal energy storage. Heat is "stored" in the form of thermal energy by melting a eutectic mixture of lithium halide salts. Stored thermal energy is recovered by solidifying the molten salt, producing steam in the process. Mechanical energy is extracted from the steam by expansion in the engine and is used to pump hydraulic fluid (water) into the high-pressure reservoir. Water from the high-pressure reservoir is used to drive the blood pump.

Figure I-1. Simplified Block Diagram of Radioisotope-Fueled, Rankine-Cycle Circulatory Support System.

Control of pumping rate and volume, and efficient coupling are accomplished by the hydraulic coupling and control element interposed between the blood pump and the high-pressure reservoir.

The circulatory support system components are incorporated into three separate packages. The power source, which includes the thermal energy storage material, the engine, and the high-pressure reservoir, will be located in the abdominal cavity and attached to the ilium or another large skeletal bone. The coupling and control element will be located in the abdominal cavity below the diaphragm. The third package is, of course, the blood pump located in the chest. Hydraulic lines connect these packages and transmit power and heat.

Specifications for the system are given in Table I-1. The power source weighs 2.2 kg, has a volume of 900 cc, and is fueled by Plutonium-238. It provides an average of 4 watts of hydraulic power output and a peak output of 12 watts. These characteristics result in a power source of acceptable size and weight which can perform the required function within acceptable physiologic limitations on reject heat, as shown under the National Heart Institute endogenous heat program.

The engine efficiency (defined as the ratio of thermal-to-shaft efficiency to ideal Rankine efficiency) for the reference design is expected to be 62.5%. This performance has been demonstrated in laboratory tests in engines of only twice the bore of the proposed engine. (The key feature responsible for the achievement of high efficiencies in such small engines is the valving developed and patented by Thermo Electron.)

TABLE I-1
SPECIFICATIONS FOR REFERENCE
IMPLANTABLE POWER SOURCE

Size	4" in diameter x 3.5" long
Weight	4.5 lb
Fuel	Plutonium-238
Thermodynamic Converter	Rankine Cycle
Working Fluid	Water
Boiler Temperature	900° F
Efficiencies	
Ideal Rankine	32%
Practical Rankine	24%
Shaft (thermal to engine shaft)	20%
Thermal Energy Storage	LiCl-LiF Eutectic
Temperatures	
Fuel Capsule Surface	935° F
Boiler	850° F
Condenser	140° F
Pressure Levels	
Inlet	400 psia
High-Pressure Reservoir	80 psia
Low-Pressure Reservoir	15 psia
Condenser	2.9 psia
Power Levels	
Radioisotope	20 watts
Pump, resting	4 watts
Pump, peak	12 watts

Maintaining this performance level in the smaller engines is a major problem. Thermal losses are minimized by reducing the cylinder wall thickness and strength is obtained by using a glass filament winding. Mechanical friction losses are reduced by avoiding the use of bearings in the transmission of the main power output of the engine; the power piston is connected directly to the hydraulic pump piston. Thus, by utilizing the Rankine cycle advantages of high cycle efficiency at relatively low temperature and lack of recompression work, the miniature steam engine can achieve more than adequate performance with credible advances beyond the state of the art.

The engine piston is coupled directly to the piston of the hydraulic pump, which uses the output of the engine to pump water into the high-pressure reservoir. If the pressure of the high-pressure reservoir is lowered by withdrawing water from it, the load on the hydraulic pump, and therefore on the engine, will be reduced. Since the inlet steam pressure is fixed, the engine will speed up, thereby increasing its power output. Conversely, if the rate of withdrawal of water is reduced, the increased pressure in the reservoir will slow the engine down, reducing its power output. The possibility of stalling is prevented by bleeding the accumulator with a pressure relief valve. The maximum output is determined by the maximum rate of production of steam, which corresponds to 12 watts of hydraulic power output.

This mode of operation largely decouples the power supply from the system, using the hydraulic power stored in the high-pressure reservoir. An advantage of this decoupling is that the power supply can be used with any assist pump or total heart replacement capable of being powered hydraulically with no modification, as long as this pump

consumes no more than 12 watts of power. Even more important is the stability of operation that results from this concept. The power supply can respond to wide variations in power demand as well as step-function changes in demand without danger of stalling or overspeeding. A three-dimensional view of the power source is shown in Figure I-2, and Chapter II contains a detailed description of the unit.

The reject heat from the steam engine is exchanged with the blood, since this appears to be the only feasible method of rejecting waste heat from the power source and conversion mechanism. The blood in turn rejects heat to the outside of the body by means of the usual physiological mechanisms of perspiration, radiation, and convection-conduction. Since blood must pass through the heat exchanger, either the heat exchanger should be located in an area of high blood flow (an artery, vein, or the blood pump), or prosthetic tubes must be relied upon to bring blood to the condenser.

The most natural way to reject the heat in or adjacent to the blood pump or pumps of a circulatory support device is to use the artificial ventricle surface area as the heat exchanger surface. This has the advantage of requiring no additional prosthesis, or additional prosthetic surface in contact with the blood. It also simplifies the surgical implantation (anastomosis). This can be accomplished by transferring the heat through the bladder wall, across the pseudo-endothelial lining on the bladder, and into the blood being pumped. The temperature gradient across the polyurethane wall of the bladder is 7°C and that across the endothelial layer is 1°C. These values are acceptable, as shown in studies by Pierce Foundation and Thermo Electron.

The characteristics tabulated in Table I-1 are suitable for an implantable power source for a circulatory support device. These characteristics are attained through use of many technology items currently under development at Thermo Electron. The heat source design is quite similar to that demonstrated with the thermal energy storage prototype. A large number of small engines have been operated at 900°F steam inlet temperature, including a test run of over 3000 hours. The calculated thermal-to-shaft efficiency of 20% represents a modest improvement over measured efficiencies (17.6%) already achieved in engines of only twice the bore of the reference design. The additional endogenous heat load of 18 watts (average) should be well within the body's ability to dissipate. Since the reference design power source is based on relatively straightforward extrapolation of existing technology, it offers considerable potential for further improvement in performance, size, and weight. Therefore, the implantable Rankine-cycle radio-isotope power source design discussed represents an attractive solution for driving a circulatory support device.

Figure I-2. Implantable, Rankine-Cycle, Radioisotope Power Source for Circulatory Support Devices.

CHAPTER II

CONCEPTUAL DESIGN

This chapter describes the circulatory support system in general and the radioisotope fueled, Rankine Cycle power source design in particular. Supporting analyses and technical discussion are given in Chapter III. Although the primary effort has been expended on the conceptual design of the power source, its interaction with the blood pump and blood pump controls has also been considered.

The circulatory support system requirements are discussed in detail in Appendix A and the blood pumping requirements are analyzed in Appendix B. The problem of additional endogenous heat is discussed in Chapter IV.

The circulatory support system will first be discussed using block diagrams. Next, the component layout and subsystem function of the reference power source is described. Finally, the system performance will be summarized.

A. BLOCK DIAGRAMS

Because of the complexity of interaction between the system components, block diagrams are given as graphic aids in Figures II-1 through II-7. The shafts of the arrows denote fluid, thermal and mechanical flow. To reduce weight and volume and improve reliability, components sometimes serve multiple functions. This should be kept in mind while studying the block diagrams. Examples of multiple functions of components are:

Blood Pump

- (1) Pumps blood
- (2) Exchanges reject heat from steam engine to blood.

Figure II-1. Block Diagram of Radioisotope Fueled, Rankine-Cycle Circulatory Support System

Figure II-2. Block Diagram of Heat Source Assembly

Figure II-3. Block Diagram of Steam Engine

Figure II-4. Block Diagram of Hydraulic Subsystem

Figure II-5. Block Diagram of Blood Heat Exchanger

Figure II-6. Block Diagram of Blood Pump Controls

Figure II-7. Block Diagram of Blood Pump

Water

- (1) Serves as working fluid for Rankine-cycle engine.
- (2) Provides mechanical coupling between power source and blood pump.
- (3) Acts as heat-transfer medium between power source and blood-pump heat exchanger.
- (4) Lubricates all system bearing surfaces.

High Pressure Reservoir

- (1) Decouples power source from blood pumps.
- (2) Buffers engine controls from steep transients.
- (3) Serves as part of thermal transfer loop for reject heat.
- (4) Enables engine to operate efficiently.
- (5) Provides feedback signal (pressure) for engine control.

A radioisotope heat source provides the energy to run the Rankine-cycle engine which pumps up a hydraulic reservoir (or accumulator). The blood pump(s) and associated controls are powered from the hydraulic reservoir and are external to the power source.

The heat source consists of the radioisotope fuel with void volume, encapsulation and shielding imbedded in the thermal energy storage material. The thermal energy storage material reduces the fuel inventory (with its associated cost, heat and radiation) while still allowing the power source to respond to peak loads.

The decay heat vaporizes the working fluid in the boiler. The vapor enters the expansion cylinder through the inlet valve and delivers work on expansion against the steam piston which is directly coupled to the piston of the hydraulic pump.

The hydraulic pump and reservoir are placed in the power source so that it can be used with both total replacement and assist devices. Based on experience with external assist pump control systems, it did not seem realistic to expect the engine to synchronize directly and otherwise respond physiologically to circulatory demands (variable heart rate, stroke volume, and systemic pressure) and still run efficiently. In addition, the hydraulic reservoir decouples the power source as far as possible, buffers the engine from responding to fast transients, provides the feedback for non-dissipative engine control and serves as part of the heat transfer loop to carry the reject heat from the power source to the blood pump.

The reject heat is dumped through the condenser into the hydraulic system where it is transferred to the blood pump. In order to reduce weight and volume of the circulatory system, the blood pump

1. Pumps blood.
2. Exchanges reject heat from the Rankine-cycle engine to the blood.

The blood in turn rejects heat to the outside of the body by means of the usual physiological mechanisms of perspiration, radiation, and convection-conduction. Since blood must pass through the heat exchanger, either the heat exchanger should be located in an area of high blood flow or prosthetic tubes must be relied upon to bring blood to the condenser. The most natural way to reject the heat in or adjacent to the blood pump or pumps of a circulatory assist device is to use the artificial ventricle surface area as the heat exchanger surface. This has the advantage of requiring no additional prosthesis, or additional prosthetic surface in contact with the blood. It also

simplifies the surgical implantation and reduces the maximum temperature rise of the blood. The temperature drop across the 30 mil polyurethane bladder of an assist pump is approximately 0.5 degrees F per thermal watt.

The feed pump operates between the high pressure reservoir pressure of about 80 psia and the boiler pressure of 400 psia. The sump pump operates between the condenser pressure of 2.9 psia and the low pressure reservoir. Fluid for bearing lubrication is bled from either high or low pressure reservoir.

B. HEAT SOURCE SUBSYSTEM

1. Fuel Capsule

The layout of the power source in Figure II-8 shows the heat source configuration sized for 20 watts of Plutonium-238 (Pu-238). The fuel form can be either the nitride (which is pyrophoric) or depleted oxide (which is expensive). The void volume for helium buildup is roughly equal to the fuel volume. Six millimeters of tantalum are used to shield the Pu-238. For an initial impurity of a PPM Pu-236, the six millimeters of tantalum shielding is sufficient to keep the gamma dose rate equal to or less than the neutron dose rate for a period of five years. The fuel is encapsulated in 80 mils of Haynes 25.

2. Thermal Energy Storage

The radioisotope fuel capsule is embedded in a cylinder of thermal energy storage material chosen as the eutectic composition of a 30 mol percent LiCl, 70 mol percent LiF. This material has a

high latent heat of fusion (362 cal/cc) and a convenient melting point (930°F). This material is stable, compatible with 304 stainless steel and has no problem of hydrogen release.

Lithium hydride is an outstanding thermal energy storage material from the standpoint of weight. However at its melting point (1250°F) it releases hydrogen, which escapes from the container at some small but finite rate. To prevent hydrogen from degrading the thermal insulation, either the lithium hydride must be sealed in heavy containers or the vacuum foil insulation must be provided with a hydrogen "getter." Neither solution has yet been demonstrated, so that the hydrogen diffusion associated with lithium hydride is a serious drawback.

3. Boiler

The boiler is a single tube wound around the thermal energy storage capsule. This kind of boiler has been used successfully on all of Thermo Electron's steam engines. The boiler is actually a part of the engine subsystem, but it is cocooned within the thermal storage subsystem.

4. Vacuum Foil Insulation

Vacuum foil thermal insulation is used around the heat source. These foils are located perpendicular to the direction of heat flow and form radiation shields to reflect the radiant heat back to the heat source. The separator or thermal conduction barrier between the metal foils is formed by a thin layer of oxide powder applied to each layer of the foil. Under AEC sponsorship (Contract No. AT(30-1)-3634), Thermo Electron has investigated tungsten, nickel, tantalum, and moly foils in combination with thorium and yttria powders. Experience

Figure II-8. Layout Drawing of Radioisotope-Fueled Rankine-Cycle Power Source.

has shown that the average spacing required per foil layer in the insulation is approximately two mils. Thermo Electron data (given in Figure II-9) with these insulation systems show a planar heat flux of less than $1/100$ of a watt/cm² at the boiler temperature of 900°F. However, to realize this performance, the insulation must be conducted in a vacuum environment to prevent gaseous conduction across the interfaces.

The reference thermal storage assembly is similar to the thermal energy storage prototype successfully demonstrated under NHI sponsorship. This unit has

- (1) Demonstrated thermal energy storage of 140 watt-hours using LiCl/LiF eutectic.
- (2) Shown the effectiveness of thermal insulation using vacuum foils (see Figure II-9).
- (3) Confirmed that transient boiler temperatures are stable for good engine control
- (4) Demonstrated the fabrication and containment of LiCl/LiF eutectic in 304 stainless steel.

The insulation loss of 2.7 watts at the melting point of the LiF/LiCl eutectic (see Figure II-9) is encouraging. The surface area at high temperature in this test (68.2 in^2) is almost double that of the heat source assembly shown in Figure II-8. Adjustment of the measured heat loss by the surface area ratios gives an insulation loss of 1.5 watts for the reference design. This loss can be reduced further by

- (1) Use of a larger number of foils
- (2) Refinement of "edge" configuration in vacuum foil package
- (3) More reflective foils.

Figure II-9. Heat Loss Through Vacuum Foil Insulation on Prototype TES Unit.

C. RANKINE-CYCLE ENGINE SUBSYSTEM

The power source concept is to convert the heat from the heat source assembly into the mechanical energy of a hydraulic accumulator by means of a miniature steam engine directly coupled to the hydraulic pump.

The primary components of the Rankine-cycle engine are:

- working fluid
- boiler
- inlet valve
- exhaust valve
- cylinder
- piston
- flywheel
- condenser
- feed pump

The relationship of these components within a radioisotope fueled, Rankine-cycle circulatory assist system is shown in Figure II-10.

1. Working Fluid

The choice of the working fluid for a heat engine is of fundamental importance. Factors which influence its choice include the cycle efficiency, thermal stability of the fluid, heat source temperature, toxicity and corrosiveness.

Potential working fluids for low-power Rankine engines are analyzed parametrically in Chapter III. The most promising fluids were water and the organic fluids -- Freon E-1, Monsanto CP-34, and Monsanto CP-9. All the organic working fluids require regeneration.

Figure II-10. Block Diagram of Radioisotope-Fueled Circulatory Assist System.

The organic fluids are especially attractive because, up to their maximum working temperature, they give a cycle efficiency exceeding that with the use of water. The working fluids were investigated as a function of inlet temperature, inlet pressure, release pressure, condenser temperature and mechanical efficiency. A typical data sample is shown in Figure II-11. Here the ideal efficiency of the Rankine cycle using a water working fluid is given as a function of inlet pressure, parametric in boiler temperature. The sensitivity of the steam engine efficiency to release pressure is shown in Figure II-12.

The conclusion of this investigation was that although the potential performance of organic working fluids is most promising, water was chosen as the working fluid for the reference design. Water gives good cycle efficiency, its thermodynamic properties are well known, and there is a wealth of practical experience involving its use in Rankine engines (i.e., steam engines). Unfortunately, thermal stability data for organic fluids for periods of time comparable to the present life requirement are not available. A more detailed evaluation of working fluids is given in Chapter III, Section C.

2. Engine Configuration

A number of different engine configurations were evaluated. Review of older steam engine designs showed that the popular double-acting design with slide valve, in addition to high valve leakage, had a feature which inevitably was inefficient. Steam was introduced and exhausted through the same port. Thus heat was transferred from the steam in its hot state to the wall and from the wall to the steam in its cold state. It follows that the enthalpy of the steam leaving the engine was greater than it would have been had the process been reversible,

Figure II-11. A Typical Family of Curves for Ideal Rankine Cycle Efficiency, Parametric in Superheater Temperature

Figure II-12. Steam Engine Efficiency as a Function of Release Pressure

and the work delivered by the engine was less by the amount that the enthalpy was greater.

This drawback is avoided in a "uniflow" engine by admitting the steam at one end of the cylinder and exhausting it through ports uncovered by the piston near the end of its stroke. However, more detailed analysis of the "uniflow" engine showed it was liable to high leakage losses and was sensitive to mechanical losses arising from the same problem found in Stirling engines, namely, that the compression work to return the piston is a large fraction of the total work. Since the net work obtained is the difference between two expansion and compression work terms of comparable magnitude, the output and efficiency become quite strongly dependent on frictional and leakage losses.

In order to avoid this problem, the steam engine is designed so that the poppet exhaust valve on the piston stays open during the entire return stroke, and negligible compression work is done in the steam cylinder. Of course, feed pump work to introduce the working fluid into the boiler must still be supplied, but this compression work is a few percent of the expansion work, even with a relatively inefficient pump. Thus, this kind of steam engine is much less sensitive to irreversible losses than the "uniflow," Rankine-cycle or Stirling-cycle engines.

A detail drawing of the steam cylinder assembly is shown in Figure II-13. The material for most engine components is 6-4 titanium alloy (6 per cent aluminum, 4 per cent vanadium). This material has a thermal conductivity less than half that of stainless steel. It is strong,

weldable and of relatively low density. The 6-4 titanium alloy is used in Thermo Electron's present steam engines.

The inside of the cylinder is chrome plated to reduce radiation losses and provide a hard bearing surface for the graphite piston rings. This bearing surface is steam lubricated. Thermo Electron's present steam engines have demonstrated excellent results with steam lubrication of graphite-hard material interfaces. Thermal losses are minimized by reducing the cylinder wall thickness while obtaining strength by using a glass filament winding.

The valve sequence which has been successfully demonstrated in all of Thermo Electron's operational steam engines is illustrated in Figure II-14. This figure also shows an indicator diagram for the cycle. During the power stroke (see Figure II-14), both the inlet and exhaust valves are closed and the steam in the cylinder expands near adiabatically against the piston. At the end of the power stroke (see Figure II-13) the exhaust valve stem engages the striker arm on the cylinder wall and "pops" the spring loaded exhaust valve. The cylinder pressure immediately drops from cutoff pressure to the condenser pressure. The lightly loaded spring keeps the exhaust valve open through the return stroke until it contacts the inlet valve. This is an important feature of the valving mechanism because it minimizes the energy to return the piston to top dead center. Thus the net work per cycle is maximized. The exhaust valve is closed as it contacts the inlet valve and opens the cylinder to superheated steam from the steam chest housing the inlet valve. The timing depends on the nipple height of the inlet valve. As the piston moves downward the

Figure II-13. Steam Cylinder Assembly.

VALVE SEQUENCE FOR THERMO ELECTRON STEAM ENGINE

INDICATOR DIAGRAM FOR THERMO ELECTRON STEAM ENGINE

Figure II-14. Valve Sequence and Indicator Diagram for Thermo Electron Steam Engine.

inlet valve is seated by the pressure differential between the steam chest and cylinder. Note that no spring is required on the high temperature inlet valve. This valving technique has significantly improved the efficiency of small steam engines and is used in the reference design power source.

Basically, the high ratio of expansion to compression work achieved with the valving just described enables the Rankine engine to operate at higher practical efficiencies at lower temperatures than gas engines (Stirling, Brayton or Ericsson). Also lower operating temperatures minimize both conduction and radiation heat losses.

The use of a spring rather than a flywheel was considered to return the piston. However, dynamic analysis indicated that it was difficult to make the components small enough, and to accommodate response to changes in power level.

Multiple-cylinder arrangements were considered, but their advantages over the single-cylinder arrangement did not appear to be significant enough to justify the additional complexity.

A single-cylinder configuration with flywheel is used in the proposed design. The cylinder bore is $1/2$ inch and the piston stroke $1/2$ inch. The engine speed is 600 rpm while the host is at rest.

The boiler is a single tube wound around the TES material, a design which has been found satisfactory on small engines. Starting at the boiler, the steam enters the cylinder through the inlet valve, expands against the piston, exits through the exhaust valve in the piston, and is conducted through the exhaust line to the condenser, located in the high-pressure system. Bearing loads are reduced by rigidly connecting the steam piston to the piston of the hydraulic pump.

The flywheel is driven from the hydraulic piston. The bearings are water-lubricated graphite; they need only to be supplied with enough water to cool and wet the loaded surfaces, and this is provided from the low-pressure reservoir.

3. Thermal Losses

One of the more difficult problems in scaling down a dynamic power supply is reducing the thermal losses through the structure and insulation. Direct losses from the heat source can be reduced to an acceptably low level by the use of vacuum foil insulation. In addition, the hot engine must be connected to the main structure close to body temperature, in such a way as to maintain its alignment and accommodate shock and vibration loads while minimizing thermal losses. In this design titanium alloy is used for these functions and also for the cylinder, piston and piping. It was chosen because of its high strength-to-weight ratio, low thermal conductivity, compatibility with water, machinability, fabricability and demonstrated reliability in engines.

Thermal loss studies indicate that the overall efficiency of the engine at 900° F is approximately the same as that of a comparable engine operating at 1250° F. This apparent anomaly arises because, although the higher inlet temperature increases the cycle efficiency by approximately 10 percent, it also increases thermal losses. Furthermore, the structural materials needed at these temperatures have higher thermal conductivity than titanium and vitiate the increase in temperature. The higher temperature also raises more difficult problems with materials, insulation, thermal energy storage and fuel containment. Comparable performance and efficiency can be realized

more easily with a moderate temperature system than with a high-temperature system.

The details of the steam cylinder, piston, intake valve, and exhaust valve are shown in Figure II-13. Graphite piston rings seal against the chrome-plated titanium cylinder wall. Side loads are minimized by extracting the bulk of the expansion stroke work directly into the piston of the hydraulic pump. The glass filament is wound around the steam cylinder for reinforcement. This technique has been successfully applied to naval guns, steam boilers, and rocket cases. The filament is wound in such a way that the sides of individual loops do not contact. Because of the filament's low thermal conductivity and restricted contact area the added conduction losses are negligible.

4. Water Inventory

A common water inventory serves four functions in the system: first, as the Rankine-cycle working fluid; second, as the energy-transfer medium coupling the power source and blood pump; third as the agent for heat transfer from the condenser to the blood pump; and fourth, as the lubricant for bearing surfaces. The entire power source/blood pump system is hermetically sealed so that no make-up water is required. Internal leakages from bearings and pistons are returned to the system via the sump pump. The condenser can be made to function in any orientation (lying, standing etc.) by providing a tapered inlet to a small-bore condenser; this design has been used in space power systems to solve the analogous zero-gravity condenser flow problem.

5. Engine Control

Primary control of the system is obtained by making the steam engine load-responsive, and not by dissipative controls. The engine speed is controlled by pressure variations in the high-pressure reservoir. Since the energy per expansion stroke is fixed, the engine must respond to fluctuations in power demand by changing its speed. Increased power demand by the blood pump drops the hydraulic reservoir pressure, which decreases the energy per stroke into the hydraulic pump. The difference between the energy of the expansion stroke and that absorbed by the hydraulic pump goes into speeding up the flywheel. Conversely, decreased power demand by the blood pump raises the hydraulic reservoir pressure and slows down the engine. A number of parameters are available for tailoring the response characteristics of the system. These include the flywheel inertia, the hydraulic accumulator pressure differential, and the accumulator volume. The possibility of stalling is prevented by bleeding the accumulator with a pressure relief valve. The maximum output is determined by the maximum rate of production of steam, which corresponds to 12 watts of shaft power output.

D. HYDRAULIC SUBSYSTEM

The hydraulic pump and reservoir are designed as a part of the power source, so that the power source can be coupled either to an assist pump or a total replacement pump. Based on experience with assist pump control systems, it does not seem realistic to expect the engine to synchronize directly and otherwise respond exactly to the circulatory demands (variable heart rate, stroke volume and systemic pressure) and still run efficiently. For

efficient operation, short-term mechanical energy storage is essential for assist devices and desirable for total heart replacements. This short-term energy storage is provided by the hydraulic reservoir, which is useful for several other reasons. It decouples the power source from the blood pump as far as possible, and allows the engine to operate efficiently regardless of blood pump power level. The hydraulic reservoir buffers the engine controls from the necessity of responding to fast transients and serves as part of the heat-transfer loop to carry the reject heat from the power source to the blood pump.

The hydraulic subsystem is made up of the following major components:

- hydraulic pump

- hydraulic accumulator (or high-pressure reservoir)

- low-pressure reservoir

- hydraulic lines coupling power source to the blood pump controls
- sump pump.

A fluid flow diagram of the proposed system is shown in Figure II-15. A section of the engine showing the sump and feedwater pumps is shown in Figure II-16. The hydraulic pump is located at the bottom of the engine housing. The feedwater and sump pumps are located on the side of the housing and driven by an eccentric off the flywheel drive. The high-pressure reservoir is located under the heat source. A pressure relief valve which bypasses the feed pump regulates the boiler-superheater pressure and prevents over-temperature of the heat source.

Figure II-15. Fluid Flow Diagram for Thermo Electron Circulatory Support System.

Figure II-16. Section of Engine Showing Sump and Feedwater Pumps.

The high-pressure accumulator operates at a mean pressure of about 80 psia. This pressure represents a compromise among the following factors:

- (1) The size of the hydraulic lines coupling the power source with the blood pump.
- (2) The temperature rise of the hydraulic fluid in dumping the reject heat.
- (3) The size and weight of the system pressurizer.
- (4) The pressure reservoir leakage.

The low-pressure reservoir operates at 15 psia. The condenser pressure at 140°F is 2.9 psia. The sump pump scavenges all leakage from pistons and bearings and pumps this water, along with the engine flow, up to the low-pressure reservoir. The hydraulic pump raises the fluid pressure from that in the low-pressure reservoir to that in the high-pressure reservoir which supplies the blood pumps. This constitutes the primary flow in the system. The flow through the sump and feedwater pumps is very much less than that through the hydraulic pump (about 1 to 5000). The feedwater pump operates between the high-pressure reservoir and the boiler.

E. REFERENCE POWER SOURCE SPECIFICATIONS

Specifications for the reference power source are given in Table II-1. The average output power available at the hydraulic reservoir is 4 watts. A maximum of 12 watts can be obtained. The maximum dose rate on the surface of the power source at the beginning of life is 50 mrem/hr. A three dimensional view of the implantable power source is shown in Figure II-17. Reference to Figures II-8 and II-10 may be helpful in visualizing the component arrangement shown in Figure II-17.

TABLE II-1
SPECIFICATIONS FOR REFERENCE
IMPLANTABLE POWER SOURCE

Size	4" in diameter x 3.5" long
Weight	4.5 lb
Fuel	Plutonium-238
Thermodynamic Converter	Rankine Cycle
Working Fluid	Water
Boiler Temperature	900° F
Efficiencies	
Ideal Rankine	32%
Practical Rankine	24%
Shaft (thermal to engine shaft)	20%
Thermal Energy Storage	LiCl-LiF Eutectic
Temperatures	
Fuel Capsule Surface	935° F
Boiler	850° F
Condenser	140° F
Pressure Levels	
Inlet	400 psia
High-Pressure Reservoir	80 psia
Low-Pressure Reservoir	15 psia
Condenser	2.9 psia
Power Levels	
Radioisotope	20 watts
Pump, resting	4 watts
Pump, peak	12 watts

Figure II-17. Implantable, Rankine-Cycle, Radioisotope Power Source for Circulatory Support Devices.

F. BLOOD HEAT EXCHANGER

Radioisotope fueled circulatory support systems create an additional heat load to the body from the reject heat of the thermodynamic converter. Although a number of alternatives have been proposed for dissipating this heat load, only heat exchange with the blood appears to be feasible. For the reference circulatory support system, the blood pump ventricle also serves as the heat exchanger surface. This has the advantage of requiring no additional prosthesis; or additional prosthetic interface with the blood. It also simplifies the surgical implantation. In addition the bulk blood temperatures are lower because the heat exchanger (i. e., blood pump) is placed in the highest possible blood flow. Obviously, the weight and volume of a separate heat exchanger are eliminated. Furthermore, the hydraulic fluid for actuating the pump can be used as the heat transfer fluid between the power source and blood pump(s).

Although characteristics of the reference power source enable it to power any pump capable of being driven hydraulically, the remainder of the circulatory support system will be designed with reference to a specific design. The TECO-CHMC assist pump will be used for this purpose. Before discussing the method of coupling the pump and the power source, pump function will be reviewed.

Connected between the apex of the left ventricle and the descending aorta, the pump accepts the blood volume ejected during ventricular systole, presenting a very low impedance to the left ventricle and allowing it to eject blood at significantly reduced cardiac work. The pump in turn ejects this blood volume into the descending

aorta during ventricular diastole. Two valves are required by the pump to perform this function. A cross section of the pump is shown in Figure II-18. In vivo studies with this device have demonstrated that it is capable of reducing left ventricular pressure by 60 to 90% in healthy animals with normal stroke volumes. Chronic in vivo studies have shown that the pump can function for periods exceeding four months without causing hemolysis or thrombosis or otherwise affecting the animals adversely. The attainment of successful chronic implants and significant lowering of left ventricular pressure make this device a logical choice for illustrating the operation of the blood pump portion of the circulatory support system.

The pump is also well suited as a vehicle for rejecting waste heat originating in the engine condenser. This can be accomplished by transferring the heat through the bladder wall across the pseudo-endothelial lining on the bladder and into the blood being pumped. The temperature gradient across the polyurethane wall of the bladder is 7°C and across the endothelial layer 1°C . These values are acceptable, as shown in studies by Pierce Foundation and Thermo Electron. A more detailed description of pump and heat exchanger function is given in Chapter III.

G. BLOOD PUMP CONTROLS

From the preceding discussion of the operation of the assist pump, it is apparent that pump systole must be synchronized with cardiac rhythm. Furthermore, aortic pressure varies considerably, and the driving force actuating the pump must follow these variations. If it does not, either a significant inefficiency will result from using

Figure II-18. TECO-CHMC Left Ventricular-Aortic Assist Pump

an unnecessarily high driving force, or the pump will not be able to eject blood when aortic pressure exceeds some level. Neither alternative is acceptable. Therefore the energy system must include some means of synchronizing the pump and matching its impedance to the driving pressure.

Assist blood pump controls are usually either "follower" or "counter pulse." A detailed discussion of blood pump controls is given in Chapter III. "Direct acting", "follower" and "inverter" assist pump controls are analyzed in Appendices F, G, and H, respectively.

Regardless of the details of the blood pump control, systole is usually terminated when a predetermined end-systolic volume is reached. This control function should be accomplished by a hydraulically actuated mechanism which senses pressure and position directly and avoids the use of dissipative fluidic controls. The overall efficiency of the control unit of 67% should be possible. It is important to note that unless great care is exercised in the design of this subsystem, serious inefficiencies can result in addition to improper control function.

CHAPTER III

PARAMETRIC ANALYSIS

A. INTRODUCTION

Because of the length and variety of topics considered in this chapter, some explanation of its organization is desirable. First, a general discussion of the design approach will be given. Next, the Rankine cycle will be reviewed and the practical problems of using such a vapor cycle will be considered. The performance of promising working fluids in a Rankine cycle will be analyzed as a function of inlet temperature, inlet pressure, cut-off pressure and condenser temperature. Primary emphasis is on superheated water and regenerative organic cycles. Potential engine configurations will be evaluated and the basis for choosing the single cylinder, directly coupled, flywheel return configuration will be explained. Fuel capsule design will be considered. The choice of thermal insulation, thermal energy storage material and component materials will be reviewed with data to support the design selections. The mechanical design features such as bearings, valves and rings will be discussed individually. Reliability will be projected on the basis of bench tests of Thermo Electron steam engines.

An evaluation of the candidate radioisotope fuels is presented. Factors considered include power density, half life, dose rate, shielding requirements and cost. Finally, weight and volume variations will be estimated.

B. DESIGN APPROACH

The purpose of this study is to establish a conceptual design for an implantable power source for circulatory support systems. The major emphasis is to be on overall approaches and combinations of approaches for subsystems in an attempt to identify the optimum system approach before becoming prematurely committed to specific approaches and mechanical designs. Therefore, the point of departure must be basic, and previous assumptions must be questioned. The general requirements of a circulatory support system are discussed in Appendix A.

A circulatory support system can take any one of three forms:

- A temporary assist device;
- A permanent assist device;
- A total heart replacement.

The first two are assist devices that work in conjunction with the left ventricle. Both types must maintain normal systemic circulation and not interfere with the function of the diseased heart. The third type of circulatory support system is a total heart replacement in which the diseased heart is excised and an artificial heart is implanted which provides the total circulation in both the systemic and pulmonary systems.

The object of the temporary assist device is to unload the left ventricle and maintain systemic circulation during the acute phase of a seizure. The intent is that the heart will be able to resume its normal function within a relatively short period. The load is reapplied to the left ventricle gradually to promote maximal myocardial healing. Care must be taken not to overload the heart to reduce the possibility of aneurysms. Eventually, the assist device is either removed (the patient gradually resuming normal

activity) or it is replaced with a permanent circulatory support system. This is either a permanent assist device or heart replacement. The temporary assist device is powered by an external power supply beside the patients hospital bed. Since it is either removed or replaced, an implantable power source is not required. Therefore, temporary assist devices will not be considered further in this study.

There is a fundamental difference between a permanent assist device and a total heart replacement which greatly influences the systems approach to each. The difference is that the assist device must operate in conjunction with a diseased heart to provide adequate systemic circulation while not causing undue trauma to the diseased left ventricle. This reduces itself to two requirements: 1) the varying blood demand of the systemic circuit will be controlled through the normal intact physiologic mechanisms which will be manifested by an increase in cardiac output from the incapacitated left ventricle; 2) the assist device must be synchronized with the beat of the heart to minimize trauma to the left ventricle which could cause it to fibrillate. Normal right heart and pulmonary function is required.

The total heart replacement, on the other hand, must provide normal cardiac output to both the systemic and pulmonary circuits. This can be accomplished by varying only stroke volume, only beat rate, or a combination of both. The question of what parameter should be used to determine the changing requirements for cardiac output is not fully resolved, although it is generally agreed that right atrial pressure is probably the best control parameter (essentially adhering to Starlings law of the heart). This can be accomplished with pulsatile flow at a beat rate and pressure-time relationship which are near normal physiologic values.

The power source output level is roughly the same for the assist pump or total heart replacement applications since the work

of the left ventricle is approximately 80 percent of the total heart work. This statement would not be valid if the controls to balance the systemic and pulmonary circuits require substantial power.

The fundamental difference in the implantable power source for the assist device and that for the heart replacement is one of control. In the case of the assist device, the control must be capable of following the heart under a wide range of conditions. The output of the heart replacement, however, need only correspond to the normal physiologic right atrial pressure-cardiac output relationship. There are certainly other differences. However, it is this difference which most affects the selection of a particular design approach for one or the other system. Because this effect is so significant, it will be necessary to consider the overall systems approach to each separately.

1. Requirements for a Permanent Assist Device

The following are some of the fundamental assumptions and requirements for permanent assist devices.

a. Assumptions

(1) Permanent assist devices will be only clinically applied to those patients with seriously incapacitated left ventricles. This is in contradistinction to temporary assist devices where the intent is to rehabilitate the left ventricle, which implies only limited damage to the myocardium.

(2) The severely scarred myocardium will, in all probability, have an impaired stroke volume with little flexibility for increased stroke volume.

(3) Increased cardiac output is accomplished primarily by increased heart rate.

(4) It will be assumed that all of the stroke volume of the left ventricle enters the assist device and, therefore, the work of the ventricle is regulated by the end diastolic pump pressure. This assumption is based upon the observation that pressure work consumes more oxygen than stroke volume work, and that full aortic pressure may prove too great a shock on the pathologic left ventricle.

(5) The surgeon (or cardiologist) will set the back pressure at the time of implantation. It will remain fixed, relying only on its inherent control system for safe operation under all reasonable conditions. It is assumed that the surgeon (or cardiologist) will want to set the back pressure so that the heart will perform as much work as it can comfortably accommodate. As the patient recovers, the capacity of the left ventricle should increase, primarily, by its ability to accommodate a greater stroke volume. However, as the patient recuperates, he will probably increase his activity level, so the two trends are compatible. It may be worthwhile to subsequently increase the fraction of work assumed by the left ventricle. Therefore, it would be advantageous to have easy access to this control, although it is not mandatory. This adjustment would be made by the surgeon (or cardiologist).

(6) Normal right heart and pulmonary function is assumed.

b. Functional Requirements for the Reference Design

(a) The permanent assist device will be synchronized with the heart and will probably pump in counterpulsation with the heart.

(b) Systemic pressure and total cardiac output will be maintained within normal physiologic limits by the intact autonomic control mechanisms.

(c) The stroke volume of the assist device must match that of the left ventricle on a beat by beat basis.

(d) Design parameters:

- 1) Stroke Volume: variable from 75 to 95 cc.
- 2) Peak Systolic Pressure Range: 120 mm to 180 mm Hg.
- 3) End Diastolic Pump Pressure: 20 mm to 80 mm Hg.
- 4) Beat Rate: Normal operation, 60 to 150 beats per minute (Short term excursions up to 200 beats per minute must be safely tolerated although normal efficient function is not required at these higher rates.)
The power source must operate effectively up to 150 beats per minute. It must be capable of safely reestablishing normal synchronized operation after a short excursion to 200 beats per minute.

(e) Variation from individual to individual will be provided for through the use of various models. However, this range of variation is not to be considered in the conceptual design, although it will be discussed in Task C, Problem Areas. For purposes of the conceptual design, a system will be designed for the typical male:

Body weight:	75 kgms (165 lbs)
Body surface:	1.73 meters ²
Body height:	5 ft, 10 in.

(f) The assist device must contain an integral heat sink and heat rejection sub-system to dissipate the heat generated by the radioisotope heat source.

2. Requirements for Total Heart Replacements

The following are some of the fundamental assumptions and requirements for total heart replacements. The specific items cited are only those which have an effect on the selection of the overall systems approach to be used with the heart replacement. It is not meant to be a complete specification for a heart replacement.

(a) Total heart replacement will be clinically applied to those patients who exhibit any of the following conditions:

- 1) a myocardium incapable of maintaining a normal sinus rhythm or not satisfactorily responsive to treatment with a pacemaker;
- 2) inadequate right heart function.

(b) Cardiac output will be varied so as to maintain normal right atrial pressure. Figure III-1 shows the normal variation of cardiac index in relation to right atrial pressure in a typical patient.

(c) Increased cardiac output should be provided in such a way as to resemble normal physiologic conditions in the vascular system. Pulsatile flow rather than continuous flow should be employed. Peak systemic and peak pulmonary pressures should be maintained near normal physiologic values so as not to interfere with the normal function of the pressoreceptors such as the carotid body. Increased cardiac output should be associated with increased beat rate in approximately a direct relationship.

(d) The cardiac output should be regulated by the right heart pump in the pulmonary circuit, with the left heart pump pumping whatever stroke volume is delivered to it from the pulmonary veins. Therefore, the signal from the left atrium for more or less cardiac output should cause an increase or decrease in the beat rate, based upon a fixed stroke volume in the right heart pump. Since the left and right pumps are directly coupled, the beat rate of both would be equal. However, the stroke volume of the left heart pump must be variable and responsive to the amount of blood returned to it by the pulmonary veins. The purpose of this requirement is to prevent damage to the pulmonary system, which has a much less flexible blood reservoir capacity than the systemic circuit. This control

may take the form of an independent left atrial pressure control or a cross-coupled control with the right side. The important point is that there be independent control of cardiac output in the pulmonary and systemic circuits, with the primary control based upon the demands of the pulmonary circuit.

(e) Gross variations from patient to patient will be provided by discreet models of overlapping output range. Vernier adjustment to accommodate each particular patient will be provided for by means of an adjustable control to vary the base beat rate. Normal physiologic variations would then be accommodated by automatically varying the beat rate around this base, or reference beat rate. Since it may prove beneficial to subsequently adjust this reference beat rate, it would be advantageous to have easy access to this control, although it is not mandatory.

(f) The heart replacement system must contain an integral heat sink and heat rejection sub-system to dissipate the heat generated by the radioisotope heat source.

3. Alternate System Approaches

There are many approaches that can be considered in attempting to meet these requirements. While on the one hand, one would like to be completely objective by considering all possible alternatives, there is a practical limit to one's capacity to do so. As a reasonable compromise, it is possible to start with broad overall concepts described in rather general terms becoming more specific as the number of approaches is narrowed to a manageable few.

4. System Approaches for the Permanent Assist Device

An implantable permanent assist device system consists of the following major components and sub-systems.

Figure III-1. Normal Variation of Cardiac Index in Relation to Right Atrial Pressure in a Typical Patient.

- (a) a blood pump
- (b) blood pump controls
- (c) coupling between the power source and the blood pump
- (d) heat sink and heat rejection sub-system
- (e) thermodynamic converter
- (f) thermodynamic converter control components
- (g) thermal insulation
- (h) thermal energy storage
- (i) radioisotope heat source
- (j) packaging, mounting, and support components

In the blood pump, many of the decisions are already made or are not relevant to the selection of the optimum system approach. Assume that the type of pump will be one that interconnects the apex of the left ventricle to the descending aorta. It is a through type pump, with two valves, and must have a stroke volume which is variable between 75 and 95 cc. The material in contact with the blood, or the type of valves, although very important to the proper function of the system, are not relevant to the question of determining the optimum approach to the overall system.

The blood pump controls and their relation to the engine controls, is a vital consideration which must be evaluated as a part of the total system. However, one requirement is clear - the controls must be responsive to the left ventricle, pumping the exact stroke volume presented to it in exact synchrony with the left ventricle.

The heat rejection and coupling subsystems are related and can be considered jointly. The blood is the optimum heat sink in

which to reject heat generated within the body. This is the approach the body uses itself to eliminate excess metabolically generated heat in vital organs, such as the heart, the liver, and the brain. The heat is generated in the power supply and dissipated into the blood in the blood pump. Therefore, there must be an interconnection or coupling which can safely transfer heat from the power source to the blood pump. The obvious choice is the coupling required to transmit power from the power source to the blood pumps. If this is a hydraulic coupling, it is feasible to use the hydraulic coupling to transfer the heat to the blood pump, and then to the blood, due to the relatively high specific heat of a liquid, and the high flow rates involved. If the interconnection is by means of a pneumatic coupling, i. e., gaseous fluid, then it is difficult to use the gaseous fluid also as the heat transfer agent, due primarily to the lower specific heat of gases as compared to liquids. In this case it is probably necessary to bring the blood to the power supply, which is a serious drawback because of the additional anastomosis, increased prosthetic area in contact with blood, and the complications encountered when one refuels by replacing the power supply. If the blood pump sub-assembly and power source are all one integral unit with direct coupling, then the problem of heat dissipation is no longer a problem but the problem of refueling or replacing the power supply is much worse. Therefore, there is much to recommend a hydraulic coupling between the power source and the blood pump sub-assembly, which also doubles as the heat transfer fluid.

The evaluation of the alternative thermodynamic converters is very complex, and is one of the fundamental aspects of this study. However, one of the basic decisions that validly can be made independently of the type of closed cycle thermodynamic converter, is that it will be reciprocating. This at least simplifies some aspects of the total system controls and the nature of the hydraulic coupling.

As mentioned earlier, the engine controls are intimately involved in the selection of the thermodynamic converter and their relationship to the blood pump controls. However, it is worth noting that through the use of a thermal energy storage system required by the wide peak to average power variation, the problem of controlling the temperature of the working fluid is greatly simplified.

Since the efficiency of low power level thermal systems hinges on minimizing heat losses, the thermal insulation has been singled out as a separate sub-system in the evaluation of the implantable power supply. Recent test results obtained at Thermo Electron with vacuum foil insulation under AEC sponsorship, unequivocally recommends it as the approach for the thermal insulation sub-system. Of course, along with this goes minimizing structural heat losses, fluid flow losses, and any other irreversibilities in the system.

The choice of thermal energy storage material depends directly on the results of the evaluation of the alternate thermodynamic converters. A material should be selected which has a melting point at the operating temperature of the thermodynamic working fluid. There are basically two classes of materials that should be considered: lithium, hydride and lithium halides or mixtures of various lithium halides.

Lithium hydride has a high latent heat, with a melting point of 1250°F, which is in the range of acceptable operating temperatures, possibly a little higher than ideally desired from the materials viewpoint. However, the greatest disadvantage of the LiH is the dissociation at the melting point with the subsequent containment of the hydrogen. For such small TES systems as the one under consideration, it may well turn out that an adequate container may well

weigh several times the weight of the LiH contained therein. Therefore, the advantage of the high latent heat of fusion may be lost. In addition, the released hydrogen must be "gettered" in the vacuum foil insulation to prevent its deterioration.

The lithium halides, and in particular mixtures of LiF, LiCl, or LiI also have high latent heats of fusion, are very stable and easy to contain in light weight containers, and offer a wide range of melting points (from the melting point of LiOH to well above that of LiH). Also, the lithium halides, in general, have a higher density than the LiH and therefore require a much smaller container. Due to the lighter, smaller containers, these will probably result in lower overall weight of the TES sub-system.

This fact, coupled with the flexibility for selecting a melting point over a wide range, recommends the mixed lithium halide salts as the primary TES material choice. The exact mixture will depend on the selection of the operating temperature of the working fluid for the thermodynamic converter.

The choice of the specific radioisotope does not affect the selection of the optimum system approach, other than from a biologic safety point of view. Therefore, the specific selection of the particular lithium halide salt can be postponed until a specific system has been selected.

The packaging, mounting and support aspects will become more apparent once a specific system is selected. Again, these aspects have little to do with the selection of the optimum system approach, with one possible exception. There are significant advantages to dividing the system into two parts: the implantable power source and the blood pump sub-assemblies. By so doing, it is possible to locate the blood pump sub-assembly in the thoracic

cavity in near proximity to the heart and major vessels to which it will be attached; and locate the implantable power supply in the abdominal cavity where there is more latitude in regard to the location, support, and refueling or replacement. The two can then be interconnected by means of a hydraulic cable, passing through the left hemi-diaphragm.

In summary, it is relatively clear that the following selections can be made for the permanent assist device components without further detailed evaluation:

Blood Pump - through type sac, two-valves, variable stroke volume, connecting the apex of the left ventricle to the descending aorta.

Blood Pump Controls - Completely responsive to the left ventricle, pumping the exact stroke volume presented to the pump in exact synchrony, either "in" or "out" of phase.

Coupling-Power Source to Blood Pump - Ideally should be hydraulic with a flow rate considerably less than cardiac output

Heat Sink and Heat Rejection Sub-System - By means of the hydraulic coupling to the blood in the blood pump.

Thermodynamic Converter - A reciprocating, closed cycle engine.

Engine Controls - Temperature of the working fluid controlled primarily by the TES system

Thermal Insulation - Vacuum foil.

Thermal Energy Storage Material - Mixed lithium halide salts.

Packaging, Mounting and Support - System divided into two

sub-assemblies, the implantable power source and the blood pump sub-systems, interconnected by a hydraulic connector.

The primary aspects which are still to be resolved are:

- a. Selection of the optimum reciprocating closed cycle thermodynamic converter.
- b. Selection of the overall system control concept (involving both blood pump, and engine controls, as well as the overall control considerations, such as stability and response).
- c. Selection of the radioisotope heat source.
- d. Selection of the best configuration of these sub-assemblies.

5. Systems Approach for the Total Heart Replacement

The major components and sub-systems are identical to those of the permanent assist device, and the evaluation of each is equally valid for the heart replacement. The major differences lie in the area of the overall systems controls. The total heart replacement will consist of two ventricles. Compared to the systemic system, the pulmonary system has limited capacity to store blood. The balancing of the pulmonary and systemic flows at different pressure levels is a major problem in total heart replacement blood controls.

C. RANKINE CYCLE

The Rankine cycle is a reversible cycle which corresponds more nearly than the Carnot cycle to the sequence of states assumed by the working fluid in practical engines. Consider the p-v diagram in Figure III-2. Starting at point 1, which corresponds to the compressed liquid after pumping into the boiler at condenser temperature, heat is supplied along the line 1,2 to raise it to the source temperature. From 2,3 the saturated liquid is converted reversibly to saturated vapor at the source temperature and pressure p_2 . The saturated vapor is next superheated along the segment 3,4. Superheating raises the mean effective temperature at which the heat is added (with a corresponding increase in thermal efficiency) without increasing pressure and confines the bulk of the expansion to the vapor region (important for engine efficiency). The superheated vapor is expanded adiabatically from 4 to 5. The mixture of vapor and liquid is then completely liquified at the condenser temperature at pressure p_1 , along the segment 5,6. The liquid is then compressed reversibly and adiabatically to the boiler pressure p_2 .

The theoretical efficiency of the reversible Rankine cycle is lower than that of a Carnot cycle operating between the same source and sink temperature. In order to achieve the Carnot cycle efficiency a thermodynamic cycle must supply all the heat at the source temperature, reject all heat at the sink temperature and be completely reversible. The basic reason that the theoretical efficiency of Rankine cycle is lower than that of the Carnot cycle is that not all the heat is supplied at the source temperature.

Why then, does one not abandon the Rankine cycle and choose a cycle with a higher theoretical efficiency (e.g., Carnot cycle) for the thermodynamic converter in an implantable power source?

Figure III-2. Pressure-specific Volume Diagram of Rankine Cycle with Superheat

The answer is that the practical Rankine cycle efficiency is usually higher than the practical Carnot cycle efficiency. Lee and Sears (Reference 1) have analyzed this problem and the data given in Table III-1 have been taken from their results.

TABLE III-1
COMPARISON OF RANKINE AND CARNOT STEAM CYCLES*

Source Temperature = 600°F Sink Temperature = 60°F η_x = Expansion Efficiency η_c = Compression Efficiency		
	CARNOT Steam Cycle	RANKINE Steam Cycle
Ratio of expansion work to compression work ($\eta_x = 1, \eta_c = 1$)	2.44	103
Cycle Efficiency ($\eta_x = 1, \eta_c = 1$)	0.51	0.41
Ratio of expansion work to compression work ($\eta_x = 0.8, \eta_c = 0.96$)	1.17	76
Cycle Efficiency ($\eta_x = 0.8, \eta_c = 0.96$)	0.13	0.33
*Data taken from J. F. Lee and F. W. Sears, "Thermodynamics," 2nd Ed., Addison Wesley, Reading, Mass. (1963)		

A striking fact from Table III-1 is that the ratio of expansion to compression work is very much greater for the Rankine cycle than the Carnot cycle. This is because the compression work in the Rankine cycle (i. e., pump work) is done on a liquid rather than on a gas (as in the Carnot and Stirling cycles). Therefore, the Rankine cycle compression work is small in comparison to the expansion work and the net work developed in each cycle is large compared to that of the Carnot cycle. Another advantage of the Rankine cycle is that the pumping of a liquid can be accomplished with high efficiency. However, this fact should not confuse the results. The Rankine cycle pump efficiency (i. e., η_c) could be dropped as low as 0.5 and the numerical results in Table III-1 would not be significantly different!

This comparison deserves sober reflection. This sample calculation demonstrates that high practical efficiencies are associated with large ratios of expansion to compression work. This point will be discussed again in Chapter V where a comparison analysis will be made between the Rankine and Stirling cycles.

Comparing power cycles on an ideal basis can be misleading. Thermal, friction and leakage irreversibilities are particularly difficult to minimize in a miniature engine such as will be required for the implantable power source. Thermo Electron's steam engines have successfully competed with Stirling engines at the kilowatt level. The Rankine cycle advantage should improve as the engine power drops and extraneous losses become relatively more important.

The foregoing discussion illustrates the reason why steam engines powered the industrial revolution. Stirling received his patent in 1817, but the hot air engine never became a serious competitor to steam engines during the 19th century. Modern advances in metallurgy and heat transfer have led to a renewed interest in the Stirling engine. However, it should not be overlooked that these

same advances also improve the performance of Rankine cycle engines. In particular, organic working fluids have the potential for giving significantly higher cycle efficiencies than water. This possibility will be discussed in the next section of this chapter.

As noted previously, the basic reason why the theoretical efficiency of the Rankine cycle is less than that of the Carnot cycle is that not all the heat is supplied at the source temperature. To improve the efficiency of the reversible Rankine cycle, the mean effective temperature at which heat is added must be increased. This may be accomplished by superheating, reheating and feed-water heating. It may also be accomplished by using a working fluid (such as some organics) with thermodynamic properties such that the expansion is roughly isothermal and a regenerator can be used to reversibly heat the liquid entering the boiler.

In practically all steam cycles the vapor is superheated to a temperature higher than that of the saturated vapor before it is expanded. Superheating is represented by the line segment 3,4 in Figure III-2. Superheating allows the highest temperature of the cycle to be raised without a corresponding increase in pressure. Superheating also improves the cycle efficiency by increasing the mean effective temperature at which heat is added. Most importantly, superheating allows the expansion to take place primarily in the vapor region. If the expansion is through a turbine, it is important to keep the moisture content low so that the wear on the turbine blades will not be excessive. In a reciprocating steam engine superheating is important to reduce cylinder condensation. Tests clearly show that allowing the expansion to go wet results in substantially lower engine efficiency.

Reheat is a solution of little interest for the reciprocating engine. It involves reheating the vapor after a partial expansion.

This gives a slight improvement in efficiency by increasing the mean temperature of heat addition. Reheat is interest in turbine expansion as a means of reducing moisture content. A reciprocating engine would have to be compound to use reheat.

Feedwater heat is a means of approximating a regenerative vapor cycle. This increases the mean effective temperature by raising the temperature at which the liquid water enters the boiler (i. e., by decreasing the amount of heat added at the lower temperatures). The regenerative vapor cycle has many practical and theoretical difficulties. However, it can be approximated by extracting and condensing steam as it expands and using the heat of condensation for preheating the feed water. This can be done by tapping off a turbine at different stages. However, it appears that a compound engine would be required to feedwater heat in a reciprocating configuration. The marginal improvement in efficiency would not seem to justify the additional mechanical complexity in such a miniature engine as required for the artificial heart.

1. Rankine Cycle Working Fluids

a. Ideal Rankine Cycle Fluid

An ideal working fluid for a Rankine cycle engine would have the following characteristics:

- (1) Thermally stable.
- (2) Critical temperature above the metallurgical limit.
- (3) Vapor pressure should be moderate at source temperature to avoid extreme stresses in the container walls.
- (4) Vapor pressure should not be too low at sink temperature. Otherwise non-condensable gas leakage and handling of high specific volumes become problems.

(5) The latent heat, or change in enthalpy during vaporization should be large in comparison with heat supplied to the liquid.

(6) The saturated vapor line on the temperature-entropy diagram should either be vertical or have a positive ds/dT slope so that isentropic expansion does not result in wet vapor. This is especially important if the cycle is regenerative.

(7) Non-corrosive with available materials.

(8) Non-toxic.

It is also desirable that the fluid be inexpensive and have good lubricating characteristics.

No single fluid has yet been found which fulfills all these specifications. Water satisfies, or nearly satisfies, specifications (1), (4), (5), and (8). However, it completely fails to satisfy (2) and (3). Mercury is satisfactory for items (2) and (3). It is nearly satisfactory for (4) and (5). A number of organic fluids satisfy all the specifications other than (1) and (5).

In the case of mercury, the condensing temperature would be 468°F at 1.0 psia. A pressure lower than this would not be practical. With an inlet temperature of 1250°F and condenser temperature of 468°F, the Carnot efficiency is reduced to 46 %. This compares with a Carnot efficiency of 67% for a condenser temperature of 100°F. Since about the same fraction of Carnot efficiency is obtained with mercury or water, water has a potential of obtaining a cycle efficiency 1.45 times that of mercury. In addition, mercury presents significant development problems without offering any apparent advantages over water as a working fluid. Although a mercury system could be developed, it would require more development effort than a comparable steam system, and would offer no compensating advantages.

Alkali metals atomically clean metal surfaces, which causes high wear rates on moving or sliding surfaces, and can even result in friction welds. Also, the vapor pressures of the alkali metals are low for the condenser temperature of the implantable power source. Although an alkali metal system could be developed, it would require more development effort than a comparable steam system. Water and organics will receive primary attention in the working fluid evaluation.

b. Cycle Parameters

For any thermodynamic engine, the cycle efficiency improves with increased temperature. Some of the factors in choosing the peak cycle temperature are:

(1) Stability of fluid (This fixes the maximum temperature a fluid can operate without decomposition.

(2) Efficiency of cycle.

(3) Heat loss: - - The higher the peak temperature in the system, the higher will be heat losses which reduce the overall efficiency.

Because of the opposite effects, a given power output implies that an intermediate temperature will optimize the overall system efficiency.

The peak pressure for a given temperature

(1) Cycle efficiency

(2) Wetness of exhaust

(3) Leakage of steam through the clearance between the piston and the cylinder.

The interaction of these factors suggests an optimum inlet pressure.

All the reject heat from the power source must be delivered to the blood. A 30° temperature across the blood pump bladder is assumed, thereby fixing the condenser temperature at 130°F. The condenser pressure is the saturation pressure of the fluid corresponding to the condenser temperature.

The release pressure is the pressure at which the exhaust valve actuates and the expanded vapor is connected to the condenser. It is fixed by --

(1) Expansion ratio

(2) Pressure difference between cut-off pressure and condenser pressure should be enough to drive the exhaust fluid out of the cylinder. The upper limit on the expansion ratio was set at 20:1.

c. Water Working Fluid

Water has many desirable properties for use in a Rankine cycle. Historically, most Rankine cycle engines have used water. Almost all of Thermo Electron's hardware experience has been with Rankine cycle engines using water as the working fluid (i. e. , steam engines). Water is not limited in operating temperature and can be used as a lubricant with graphite and carbide bearings. Therefore, water must be carefully evaluated for application in an implantable power source for circulatory support devices.

Initial calculations of ideal Rankine cycle efficiency were made without taking heat loss, leakage or release pressure into account. The indicator diagram for release pressure equal to the condenser pressure is given in Figure III-3. The tabulated results for the survey calculations for water are given in Tables III-2 and III-3. Graphs of ideal Rankine cycle efficiency for water as a function of inlet pressure, parametric in inlet temperature are given in Figures III-4 and III-5.

Figure III-3. Indicator Diagram with Ideal Inlet Valve and Cut-off Pressure Equal to Condenser Pressure

Figure III-4. Ideal Rankine Cycle Efficiency for Water Working Fluid and Condenser Temperature of 130°F

Figure III-5. Ideal Rankine Cycle Efficiency for Water Working Fluid and Condenser Temperature of 220°F

TABLE III-2.

SURVEY EFFICIENCY CALCULATIONS FOR WATER
AT CONDENSER PRESSURE OF 2.2 PSIA

Condenser Temperature - 130°F Condenser pressure - 2.2 psia
Release pressure - 2.2 psia Feed pump efficiency - 1.0

Inlet Temperature °F	Inlet Pressure (psia)	Efficiency (percent)	Exit Moisture (percent)	Mean [*] Effective Pressure (psia)
1250	1200	41.2	11.2	24.25
1250	1000	40.8	10.2	23.81
1250	800	40.3	8.5	23.2
1250	600	39.8	6.65	22.05
1250	400	38.0	3.8	20.9
1250	200	34.7	Superheated	18.3
850	1000	37.1	18.5	20.8
850	600	35	14.8	18.5
850	500	34.8	13.5	18.17
850	400	33.2	11.9	17.1
850	200	30	7.3	15.05
750	1000	36.5	21.5	20.0
750	600	34.05	17.8	17.75
750	500	33.4	16.0	17.25
750	400	32.35	14.5	16.5
750	200	29.2	9.5	14.3
650	1000	35.7	29.8	19.55
650	600	33.3	20	17.18
650	500	32.6	18.7	16.7
650	400	31.6	17	15.91
650	200	28.7	13.8	14.7

* Mean effective pressure is tabulated since it is a measure of engine size.

TABLE III-3.

SURVEY EFFICIENCY CALCULATIONS FOR WATER
AT A CONDENSER PRESSURE OF 17.2 PSIA

Condenser temperature---220°F
Condenser pressure -----17.2 psia
Release pressure ----- 17.2 psia
Feed pump efficiency ---- 1.0

Inlet Temperature °(F)	Inlet Pressure (psia)	Efficiency (percent)	Exit Moisture (percent)	Mean Effective Pressure (psia)
1250	1200	34.9	1.8	121.0
1250	1000	34.0	0.3	115.9
1250	800	33.1	Superheated	105.5
1250	600	31.7	"	100.0
1250	400	29.3	"	80
1250	200	25.2	"	60
850	1000	30.1	10.8	97.8
850	600	27.55	6.4	86.3
850	500	26.4	4.8	83.0
850	400	25.2	2.7	77.6
850	200	21.2	Superheated	55.8
750	1000	29.2	14.3	94.0
750	600	26.4	10.0	81.4
750	500	25.7	7.8	78.1
750	400	24.9	6.4	76.2
750	200	20.0	S	55.5
650	1000	28.7	19.9	95.6
650	600	25.75	12.8	78.65
650	500	24.7	11.2	74.7
650	400	23.7	9.21	70.2
650	200	19.65	5.3	61.2

In order to be more realistic, the effects of release pressure and the feed pump work were included in the efficiency calculations. The release pressure was limited by either a maximum expansion ratio of 20:1 or a minimum pressure difference of 4 psi across the exhaust valve (difference between release pressure and the condenser pressure). The tabulated results are given in Table III-4. The associated graph of Rankine cycle efficiency versus inlet pressure is given in Figure III-6.

d. Sample Calculation for Steam

Consider the temperature-entropy diagram for a Rankine cycle given in Figure III-7.

Figure III-7. Temperature-entropy diagram for superheated Rankine cycle.

Figure III-6 Rankine Cycle Efficiency for Water Working Fluid including the Effect of Cut-off and Feedwater Pump Efficiency of 0.6 (Release Limited by 20:1 Expansion Ratio or Minimum Pressure Differential of 4 psi between Release and Condenser Pressure)

TABLE III-4

EFFICIENCY OF WATER WORKING FLUID
INCLUDING RELEASE PRESSURE

Condenser pressure 2.2 psia
Condenser temperature 130°F
Feedpump efficiency 0.6

Inlet Temperature (°F)	Inlet Pressure (psia)	Efficiency	Mean Effective Pressure (psia)	Exit Moisture (percent)	Expansion Ratio	Release Pressure (psia)
1250	1200	35.7	172	Superheated	20.0:1	24
1250	1000	35.9	153	S "	20:1	21.7
1250	800	35.4	119.4	S "	20:1	16.6
1250	600	35.1	88.0	S "	20:1	12.5
1250	400	34.9	58.2	S "	20:1	8.3
1250	200	33.0	42.5	S "	13:1	7.2
850	1000	30.4	150	8.5	20:1	26.5
850	800	30.6	119.0	7.6	20:1	20
850	600	30.7	90.0	6.4	20:1	15
850	400	30.2	58.8	5.8	20:1	9.5
850	300	30.1	42.9	5.0	20:1	7.0
850	200	28.25	36.85	2.2	14.5:1	6.5
650	1000	28.1	158	16.5	20:1	30.8
650	800	28.2	124.2	14.5	20:1	24.0
650	600	28.5	98.2	12.9	20:1	17.5
650	400	28.2	60.1	11.3	20:1	11
650	200	24.65	34	9.8	16.15:1	6.5

Heat given by boiler = $(h_1 - h_5)$ B/lb

Feed water pump work = $W_p = h_5 - h_4 = v_4 \frac{(P \text{ boiler} - P \text{ cond.})}{\eta_{\text{Pump}}}$

Engine Output = $(h_1 - h_2) + v_2 (P \text{ cut-off} - P \text{ condenser}) - W_p$

Therefore,

$$\eta_{\text{Engine}} = \frac{(h_1 - h_2) + v_2 (P \text{ cut-off} - P \text{ condenser}) - W_p}{h_1 - h_5}$$

for the case of inlet steam conditions at 1000 psia, 850°F, one finds from the steam tables that

$$h_1 = 1419 \text{ BTU/lb}$$

$$v_1 = 0.721 \text{ cft/lb}$$

$$S_1 = 1.59 \text{ BTU/}^\circ\text{F lb}$$

$$\text{cond. temp} = T_4 = 130^\circ\text{F}$$

$$\text{" pressure} = P_4 = 2.22 \text{ psia}$$

$$S_2 = 1.59 \text{ BTU/}^\circ\text{F lb}$$

$$v_2 = 14.42 \text{ cft/lb}$$

(expansion ratio 20:1)

$$\therefore P_2 = 26.5 \text{ psia} = P \text{ cut-off } h_2 = 1079 \text{ BTU/lb}$$

$$\therefore v_2 (P \text{ cut-off} - P \text{ cond}) \quad \text{Moisture})_2 = 8.5\%$$

$$= \frac{14.42 \times (26.5 - 2.22)}{\frac{\text{ft} \cdot \text{lb}}{\text{BTU}}} \times 144 \frac{\text{psf}}{\text{psi}}$$

$$= 65 \text{ BTU/lb}$$

$$\therefore \text{ Pump Work} = v_4 \times (P_{\text{boiler}} - P_{\text{cond}}) \quad \eta_{\text{Pump}} = 60\%$$

$$= \frac{0.0162 \times (1000 - 2.22) \times 144}{778 \times 0.6}$$

$$= \frac{2.9}{0.6} \text{ B/lb} = 4.83 \text{ BTU/lb}$$

$$\therefore \text{ Net Work Output} = 1419 - 1079 + 65 - 4.83$$

$$= 400.17 \text{ BTU/lb}$$

$$h_5 = h_4 + 2.9$$

$$= 98^4 + 2.9 = 100.9$$

$$\therefore \eta = \frac{(400.17) \times 100}{1419 - 100.9} = 30.4\%$$

$$\text{Mean Effective Pressure} = \frac{402.1}{v_2} \times \frac{778}{144} = 150 \text{ psi}$$

Comparison of Figure III-4 and III-5 show that a large efficiency penalty is incurred if the condenser temperature of 220°F should be practical for rejecting heat to 100°F blood sink. The temperature drop across the blood heat exchanger is approximately 13°F and the temperature rise to transfer the reject heat from the power source to the blood pump is only about 6°F. The balance of the temperature drop between the condenser and blood is budgeted to miscellaneous temperature drops in the heat rejection path.

The cycle efficiency for water is insensitive to condenser temperature near 130°F and an increase in condenser temperature to 150°F would not seriously affect the Rankine cycle performance. However, increasing the condenser temperature elevates the power

source temperature and creates a problem of thermally insulating the power source from adjacent tissue.

An encapsulation technique that could be used to thermally insulate the power source and protect it from body fluids was developed for electrical heat exchangers as part of the NHI-sponsored contract, "Study of the Effects of Additional Endogenous Heat." It has since been successfully applied to radioisotope heat exchangers. The heat exchangers are encapsulated in layers of Fiberfrax thermal insulation, General Electric TRV-120 and Dow Corning Medical Grade 382 Silastic. A composite thickness of 0.150 should be adequate.

Practical considerations of cut-off (limited by 20:1 expansion ratio or minimum pressure differential of 4 psi between cut-off and condenser pressure) and feedwater pump efficiency of 0.6 have been incorporated into Figure III-6. Comparison of Figures III-5 and III-6 shows that release and feedwater factors do not degrade the Rankine cycle efficiency seriously.

More importantly, the Rankine cycle efficiency in the figures should be reduced by about 0.8 to account for mechanical inefficiency. Therefore, a Rankine engine using a water working fluid (i.e., a steam engine) should give practical cycle efficiencies greater than 20 percent over a range of reasonable temperatures and pressures.

e. Organic Working Fluids

Organic fluids were surveyed for application to Rankine engines. Thermal composition limits the practical maximum operating temperatures of Diphenyl oxide, Dowtherm A, Dowtherm E, Freon-113 and Freon-114 to less than 700°F. The low vapor pressure of Dowtherm A and diphenyl oxide make these fluids unattractive.

The most promising fluids were selected as:

1. Freon E-1
2. Monsanto CP-34
3. Monsanto CP-9

Thermodynamic data for the organic fluids were supplied by the vendors. All the organic working fluids require regeneration. The flow diagram of a regenerative - Rankine cycle engine is given in Figure III-8.

Initial calculations of ideal Rankine Cycle efficiency were made without taking heat loss leakage or release pressure into account. The results of the survey calculations are given in Tables III-5 through III-7.

TABLE III-5.

SURVEY EFFICIENCY CALCULATIONS OF FREON E-1 AT
A CONDENSER PRESSURE OF 23.6 PSIA

Condenser temperature---130°F
Condenser pressure ----- 23.6 psia
Cut-off pressure----- 23.6 psia

Inlet Temperature (°F)	Inlet Pressure (psia)	Efficiency	$\frac{\Delta H)_{\text{engine}}}{\Delta H)_{\text{regenerator}}}$ *	Mean Effective Pressure (psia)
800	550	34.4	0.322	82.7
600	550	34.1	0.305	80.4

* $\Delta H)_{\text{engine}}/\Delta H)_{\text{regenerator}}$ is a measure of the size of the regenerator. This ratio shows that FREON E-1 requires a large regenerator.

TABLE III-6

SURVEY EFFICIENCY CALCULATIONS OF MONSANTO
CP-34 AT A CONDENSER PRESSURE OF 5.7 PSIA

Condenser temperature	130°F
Condenser pressure	5.7 psia
Cut-off pressure	5.7 psia
Feedpump efficiency	1.0
Regenerator efficiency	1.0

Inlet Temperature (°F)	Pressure (psia)	Efficiency	$\frac{\Delta H^{\text{engine}}}{\Delta H^{\text{regenerator}}}$	Mean Effective Pressure (psia)
550	500	34.8	4.3	26.82
550	200	34.78	2.364	23.0

$\Delta H^{\text{engine}} / \Delta H^{\text{regenerator}}$ is high and hence a regenerator of comparatively small size will be satisfactory.

TABLE III-7

SURVEY EFFICIENCY CALCULATIONS OF MONSANTO
CP-9 AT A CONDENSER PRESSURE OF 0.5

Condenser pressure	0.5 psia
Condenser temperature	338°F
Feedpump efficiency	1.0
Regenerator efficiency	1.0

Inlet Temperature (°F)	Inlet Pressure (psia)	Efficiency	$\frac{\Delta H^{\text{engine}}}{\Delta H^{\text{regenerator}}}$	Mean Effective Pressure (psia)
650	25	23.8	0.433	2.01
650	50	27.8	0.495	2.4

Figure III-8. Flow Diagram of Regenerative-Rankine Cycle Engine

More detailed calculations included the effects of release pressure and feed pump work. A feed pump efficiency of 0.6 and a regenerator efficiency of 1 were assumed. The release pressure was limited by either a maximum expansion ratio of 20:1 or a minimum pressure difference of 4 psi across the exhaust valve (difference between release pressure and condenser pressure). The tabular results are given in Tables III-8 and III-9. Monsanto CP-9 is excluded from these more detailed calculations because of its lower cycle efficiency, inconveniently low condenser pressure and requirements for a large regenerator.

A comparison of FREON E-1 and Monsanto CP-34 working fluids is shown in Figure III-9. Monsanto CP-34 is of primary interest because it gives high efficiency at lower inlet pressures and requires a smaller regenerator. The cycle efficiencies of CP-34 and E-1 with an inlet temperature of 550°F are comparable to those of water with an inlet temperature of 900°F.

This lower inlet temperature is an extremely important factor in maximizing overall efficiency in a miniature engine. An inlet temperature of 550°F virtually eliminates thermal radiation losses and roughly halves the thermal conduction losses. If this were the only basis for choosing a working fluid for the Rankine cycle, the organics would be a clear choice over water.

However, other factors must be considered. In the reference design the working fluid also acts as a lubricant, hydraulic fluid and heat transfer medium. Vendor data on lubricating qualities and long term thermal stability for the organic fluids are extremely sketchy.

TABLE III-8.
EFFICIENCY OF FREON E-1 INCLUDING RELEASE PRESSURE

Condenser temperature ----130° F
 Condenser pressure -----23.6 psia
 Feed pump efficiency -----0.6
 Regenerator efficiency ---- 1.0

Inlet Temp. (°F)	Inlet Pressure (psia)	Efficiency	Mean Effective Pressure (psia)	$\frac{\Delta H_{\text{engine}}}{\Delta H_{\text{regenerator}}}$	Expansion Ratio	Release Pressure (psia)
550	800	30.8	150.0	0.243	20:1	52
550	600	30.6	107.2	0.2335	20:1	39
550	461.8	29.8	78.0	0.233	19.5:1	28.2
550	414.1	29.4	75.7	0.224	17.3:1	28.2
550	367.4	29.3	58.1	0.223	14.8:1	28.2
550	319.6	28.4	51.5	0.195	12.6:1	28.2
500	414.8	34.5	86.4	0.215	15.8:1	28.2
500	414.8	28.6	74.0	0.245	18.2:1	28.2

TABLE III-9
EFFICIENCY OF MONSANTO CP-34, INCLUDING RELEASE PRESSURE

Condenser temperature---- 130°F
Condenser pressure----- 5.68 psia
Feed pump efficiency----- 0.6
Regenerator efficiency ---- 1.0

Inlet Temp. (° F)	Inlet Pressure psia	Efficiency	Mean Effective Pressure (psia)	$\frac{\Delta H) \text{ engine}}{\Delta H) \text{ regenerator}}$	Expansion Ratio	Release Pressure (psia)
550	500	30.0	95	1.96	20:1	25
550	300*	32.0	58	1.75	20:1	15
550	200	30.0	35.8	1.34	14.4:1	10
550	150	28.0	31.5	1.1	13.1:1	10

* Calculated by interpolation.

Figure III-9. Comparison between FREON E-1 and Monsanto CP-34 Working Fluids

f. Fluorochemical Working Fluids

Ray and Ross (Reference 3) have recently made a theoretical assessment of fluorochemicals as working units for small Rankine cycle power units. Although the analysis was performed for turbine expansion, the cycle efficiencies apply equally well to reciprocating expansions. The four fluids evaluated were:

- 1) Hexafluorobenzene (HFB)
- 2) Dichlorotetrafluorobenzene (DCTFB)
- 3) Perfluorodecalin (PFD)
- 4) FC-75

Physical properties of the compounds are listed in Table III-10. The data on HFB, DCTFB and PFD were provided by Imperial Chemical Industries and Imperial Smelting Corporation while those of FC-75 are quoted from Minnesota Mining and Manufacturing Company technical literature. All these compounds are characterized by chemical inertness, high degree of thermal stability and non-inflammability. DCTFB may be somewhat less stable than the other three materials.

TABLE III-10
PHYSICAL PROPERTIES OF FLUOROCHEMICALS

	HFB	DCTFB	FC-75	PFD
Mol. wt.	186	219	416	462
B. P., °C	80	156	99-107	142
F. P., °C	5.2	2.8	-62 (Pour point, max.)	-10
Critical temp., °C	244	353	227	293
Critical pressure, atm.	32.6	52.5	15.9	15.0
Density, g/cm ³ at 25°C	1.607	1.660	1.770	1.946 (20°C)
Surface tension, dynes/cm at 25°C	22.6	28.9	13.5-18.0	—
Viscosity, centistokes at 25°C	0.56	0.87	0.65	—

All the fluids have temperature-entropy diagrams with the saturated vapor line being either vertical or having a positive ds/dT slope so that isentropic expansion does not result in "wet" vapor. This means that vapor which is initially unsaturated becomes superheated after expansion since the useful enthalpy drop is reduced. However, this difficulty is overcome by recovering heat from the superheated vapor leaving the cylinder in a regenerator and adding it to the boiler feed. This raises the mean effective temperature at which the heat is added so that cycle efficiency is improved. The flow diagram of a regenerative-Rankine cycle engine is given in Figure III-8. The Rankine cycle efficiencies calculated by Ray and Ross are given in Table III-11. Near perfect regenerator efficiency is assumed in column 3.

TABLE III-11
RANKINE CYCLE EFFICIENCIES*

Inlet temperature = 800°F
Condenser temperature = 140°F

Working Fluid	Efficiency Without Regenerator (Percent)	Efficiency With Regenerator (Percent)	Regenerator Load for 10 watt output Btu/hr
HFB	17.8	38.7	274
DCTFB	27.6	45.3	58.6
FC-75	13.5	39.8	228
PFD	13.1	41.5	206

* Mechanical efficiency of 1

The Rankine cycle efficiencies with regenerator tabulated in column 3 of Table III-11 are quite attractive. DCTFB is particularly interesting because of its high efficiency and low regenerator load. This implies that the cycle efficiency will not be sensitive to regenerator efficiency and that the regenerator volume will not be prohibitive. For example, PFD requires a regenerator about three times as large as that for DCTFB to give a comparable thermal efficiency.

These results indicate that fluorochemicals can give appreciably higher regenerative Rankine cycle efficiencies than water in an unmodified Rankine cycle engine. Although Rankine engines will probably use water, longer range and higher performance engines will certainly use organic working fluids in a modified Rankine cycle with regeneration.

g. Conclusion

The conclusion of this investigation is that organic working fluids are extremely promising and offer significant overall efficiency advantages over water. However, questions as to lubricating qualities and thermal stability need resolution before their long term application can be considered. Therefore water is chosen as the working fluid for the reference design power source. Water gives good cycle efficiency, its thermodynamic properties are well known, and there is a tremendous amount of practical experience available involving its use in Rankine engines (i. e. , steam engines).

D. ENGINE CONFIGURATIONS

A number of steam engine configurations were evaluated for application in the implantable power source. These configurations included:

- Slide valve engine
- Uniflow engine
- Compound engine
- Linear engine with spring return
- Rotary engine with flywheel return
- Multiple cylinders
- Linear engine with flywheel return

1. Slide Valve Engine

The slide valve is virtually an inverted box so that sliding contact between the valve and port face is steam tight. Admission and release of steam take place as the valve is reciprocated over the port face. Its advantages are 1) simple construction, 2) easily reversible, 3) variable expansion and 4) runs well over wide speed range if steam is not superheated. The slide valve's disadvantages are 1) high friction, 2) late cutoff, 3) large clearance volume and 4) difficult lubrication.

In addition, there is an irreversible loss due to steam being supplied and exhausted through a common port. A large transfer of heat occurs from steam to walls. The steam is thereby cooled and condensed while the walls are heated. This process is termed "cylinder condensation." The transfer of heat from the steam in its hot state to the wall and from the wall to the steam in its cold state is inherently irreversible. It follows that the enthalpy of the steam leaving the engine is greater than it would have been had the process been reversible. Therefore, the efficiency is reduced since the work output of the engine is less by the amount that the enthalpy is greater.

The simple slide valve can be improved by relief rings or by coiling it around the axis of reciprocation to form a piston valve. Other improvements over the slide valve are the Corliss valve and the drop valve. However, the fundamental irreversibility remains. For this reason, slide type valve engines were dropped from consideration for the present application.

2. Uniflow Engine

The uniflow engine reduces cylinder condensation by eliminating the reversal in the direction of steam flow between admission and exhaust in the slide valve engine. The engine obtains its name from the unidirectional flow of steam in the cylinder from inlet to the exhaust. An exhaust port is uncovered near the end of the piston stroke so that hot and cold steam are not alternately passed over the same surfaces as in the slide valve engine. This engine is well suited for using superheated steam (a necessity for high engine efficiency) and mechanical efficiency is high because of few moving parts. In practice, uniflow engines have proven efficient and responsive.

Although the terminology is less than precise, the term "uniflow" also implies the use of exhaust ports rather than valves. Therefore, the steam that is not exhausted through the port results in high compression on the return stroke. This is undesirable because it reduces the net work per stroke and makes the engine more sensitive to frictional, leakage and thermal losses. This high compression is unacceptable for the implantable power source application because a great deal of rotational energy would be required to return the piston. Either a massive flywheel or a high speed engine would be required. The former is distasteful because of size and weight while the latter would make the reliability problem more severe. Both would increase the gyroscopic moment of the power source.

Because of the foregoing problems, the uniflow engine with exhaust ports was eliminated from consideration for an implantable power source. Actually, the reference design, as are all of Thermo Electron's operational steam engines, is a uniflow engine with an exhaust valve which stays open the entire return stroke. This feature eliminates compression and allows the piston to be returned with a small flywheel. In addition, the net work output per cycle is not degraded and the engine remains relatively insensitive to irreversible losses. A detailed analysis of the uniflow engine for both the exhaust port and exhaust valve is given in Appendix K.

3. Compound Engine

If the expansion is divided into two parts, executed successively in separate cylinders, the change in temperature of the steam in one cylinder may be cut in half. The differences in temperature between the steam and walls is reduced since the wall surface to which the steam is exposed is not doubled (since the low pressure cylinder is of the same dimensions as a single cylinder engine while the higher pressure cylinder is much smaller). This engine modification is called compounding. The principle has been extended to as many as five expansions in series, although more than three is uncommon.

Compounding has demonstrated marked efficiency advantages in large engines. However, these advantages are purchased at the expense of mechanical complexity. Because of the miniature size of the engine and reliability considerations, compound engines did not seem well suited for such small engines as required for the implantable power source.

4. Linear Engine with Spring Return

The dynamics of a linear steam engine with spring return is described in Appendix J. In this configuration the steam piston is coupled directly to the hydraulic piston. (There is no rotary motion.)

On the power stroke part of the cycle energy is stored in a spring. This energy is recovered when it is used to return the piston on the upstroke. This configuration would be workable except for the requirement that the engine have a maximum to minimum speed ratio of 3:1 (for physiological reasons).

This objection can be simply illustrated. If the engine stroke and spring index are constant, the energy stored in the spring during the power stroke will be constant. This means that the duration of the return stroke which follows will be constant. This duration can be fixed such that it is one-half of the total minimum speed cycle time. The cycle time at maximum speed will be one-third of the cycle time at minimum speed. Therefore, the duration of the return stroke should be about one-sixth of the minimum cycle time. Since the return stroke has been already set at one-half the minimum cycle time, this is an impossible situation. Alternatively, the spring could be chosen so that the return stroke duration is always one-half the maximum cycle time of one-sixth the minimum cycle time. However, the engine will be running at a mean RPM which is slightly above the minimum. This means that most of the time the return stroke duration will be about one-fifth the power stroke duration. If the situation is analysed further, it can be shown that the piston will rapidly accelerate during the return stroke to a maximum speed at top dead center at which time the steam inlet will open to begin the power stroke. The sequence would require the engine to work against its own inertia and would be very inefficient.

Another solution would be to vary the engine stroke and/or spring index with engine RPM. However, this would require complex controls which are undesirable.

5. Rotary Engine with Flywheel Return

Another possible engine configuration is a rotary engine with a flywheel return. In this configuration the steam piston is connected directly to a flywheel via a crankshaft. The purpose of the flywheel is to store a certain amount of energy on the power stroke and then use this stored energy to displace the piston on the return stroke.

This configuration requires at least two bearings (at the ends of the crankshaft) that will transmit the entire energy of the power stroke. It would be preferable, however, to have an engine configuration that would not put full engine load on bearings. This would improve both bearing life and overall engine reliability.

6. Linear Engine with Flywheel Return

The reference design configuration is a linear engine with a flywheel return. The purpose of the flywheel is as before -- to store a fraction of the energy from the power stroke to use for the return stroke. The hydraulic pump is coupled directly to the steam piston--no bearings are required. The stroke of the steam piston is the same as the stroke of the pump. The flywheel is turned by an eccentric which is connected via a crankshaft to the hydraulic piston. Therefore, the only rotary bearings are between the hydraulic pump and flywheel. The average load on these bearings is significantly lower than the load on the bearings of a rotary engine configuration. Therefore, the linear engine with flywheel return has the dual advantage of low bearing load and the dynamic response of a rotary engine with flywheel return.

7. Multiple Cylinders

Other configurations using multiple cylinders were investigated. Three cylinders could be used with pistons which are each 120° out of phase with the other. This would eliminate the need for a flywheel

since at least one piston would be in its power stroke at all times and would return the other pistons. However, this system would require more parts and have higher thermal losses through the engine structure. In addition it would be difficult to fit three cylinders with no flywheel into the same space as one cylinder with a flywheel.

8. Conclusion

After a review and evaluation of basic steam engine designs, a superheated, linear, flywheel return, uniflow Rankine engine with poppet exhaust valve was chosen as the most promising for the thermodynamic converter in an implantable, radioisotope-fueled, power source. The valving principle is described in detail in Chapter II.

The slide valve engine is inherently inefficient. The uniflow engine (with exhaust ports) requires too large a flywheel. The compound engine is too complex mechanically. The linear engine with spring return is difficult to control efficiently over a wide power range. The rotary engine with flywheel return had undesirable rotary bearing loads. Multiple cylinders are more complex and offer additional paths for thermal loss.

The basic design selected is essentially that used in Thermo Electronics operational engines. However, the direct coupling between the steam and hydraulic pump piston greatly reduce the average rotary bearing load. The flywheel return is chosen for reasons of non-dissipative dynamic response.

E. THERMAL INSULATION

The most effective thermal insulators are either fibrous, opacified powder and vacuum foil. Microquartz and Min-K are commercial examples of fiber insulations and Santocel is an example of an opacified powder. The Linde Super Insulation and Thermo Electron multiple foil insulation are well-known vacuum foil insulations. A useful thermal insulation must not only impede heat transfer by solid conduction, gas conduction, and radiation; but it must also exhibit chemical, physical and mechanical stability. A comparison of the characteristics of various thermal insulations is given in Table III-12.

1. Fiber Insulation

Min-K is a proprietary composition of Johns Manville and has been widely used in the SNAP radioisotope thermoelectric generators. Although its thermal insulation properties are not outstanding, they are adequate for many generator applications. This material is available in either powder or block forms. The compacted Min-K is load bearing and can be machined to loose tolerances. However, Min-K is quite difficult to outgas and is not suitable for static vacuum operation. When used in thermoelectric generators, Min-K is outgassed and then backfilled with an inert atmosphere, usually argon.

An example of powder insulation is Santocel, a product of Monsanto Chemical Company. This material is a fluffy white powder composed primarily of silicon dioxide. As with other insulations, pressure has a decided effect on this product's thermal conductivity. This is illustrated in Table III-13.

TABLE III-12
THERMAL INSULATION COMPARISON

Insulation	Maximum Operable Temperature (°F)	Apparent Thermal Conductivity Between Boundary Temperatures of 100°F and 1500°F 10^3 Btu/hr - ft - °F
Fibrous potassium titanate Non-vacuum (1)	2200	39
Min-K (2)	1800	29
Opacified Powder Vacuum at 500 μ (3)	2200	7.5
LINDE Super Insulation Vacuum at 10 μ (4)	2200	0.90
Thermo Electron Vacuum foil insulation Vacuum at 1 μ (5)	3000	0.17

- (1) H. C. Gullledge, I. E. C. 52, 2, 117 (1960)
- (2) Vendor bulletin
- (3) A Wechsler, AFOSR Conf., Cambridge, Mass (1961)
- (4) D. Wang, AFOSR Conf., Cambridge, Mass. (1961)
- (5) Phase I Final Report, Vacuum Foil - Type Insulation for
Radioisotope Power Systems, Thermo Electron Corporation,
ALO-3434-6.

Table III-13

EFFECT OF AIR PRESSURE ON
SANTOCCEL THERMAL CONDUCTIVITY

Pressure (mm Hg)	Thermal Conductivity Btu 3 In/Hr-Ft ² - °F
100	0.087
10	0.058
1	0.037
0.1	0.022
0.01	0.012
0.001	0.009

The thermal conductivity of Santocel can be appreciably reduced by grinding and combining it with metal powders, usually aluminum. Although this opacified powder is an effective insulator under vacuum conditions it is extremely difficult to outgas and maintain a hard vacuum under static conditions.

Fiber insulations have their lowest conductivities under vacuum conditions. However, fiber insulations such as Min-K and Santocel are infamous for outgassing in vacuum. It did not appear reasonable that the vacuum thermal conductivities of the fiber insulations could be maintained for the life of the power source. Fiber insulations are normally backfilled with an inert atmosphere which increases their thermal conductivity.

Most of the older radioisotope thermoelectric generators (RTGs) use Min-K fiber insulation backfilled with argon which is compatible with the more efficient telluride semiconductor thermocouples. Telluride thermoelements vaporize unduly in a vacuum near their maximum operating temperature so that they are usually run in an inert and/or reducing atmosphere. Since the

telluride thermoelements are unsuitable for vacuum operation, a vacuum seal must be made if vacuum foil thermal insulation is used. Typically for RTGs of several watts output, the thermal losses down the vacuum seal are much higher than through the foil stack. Since the thermocouples are self supporting, argon filled thermal insulation is reasonably satisfactory. However, if a reciprocating engine is used as the thermodynamic converter; the cylinder wall, itself, can serve as the vacuum seal between heat source and power source container temperatures. Thus, there is no vacuum seal loss penalty in the case of the reciprocating engine compared to that of the thermoelectric generator.

2. Vacuum Foil Insulation

The vacuum foil insulations are arranged in layers located perpendicular to the direction of heat flow and form radiation shields to reflect the radiant heat back to the heat source. A separating layer is used to prevent physical contact between the foils and direct conduction paths between the foils. Since some contact occurs between the foils and the separators, the separating layer must exhibit a high resistance to conduction heat transfer. This high thermal resistance is established primarily by surface contact resistances between the foil and separator materials. To prevent gaseous conduction across the interfaces, vacuum foil insulation must be operated in a vacuum. By proper design of the separator, the conduction component of heat transfer through the foils can be made small compared with the radiation component.

Linde originally developed vacuum foil insulations for cryogenic applications under the trade name, Super Insulation. However, subsequent work has extended the useful operating temperature of their product to 2200°F. Linde uses fibrous mats of glass, at low temperatures, or quartz, at high temperatures to

separate the foils. Thin filaments are used to reduce the area of the contact points, thus increasing the length of conduction paths through the fibers.

Vacuum foil insulation has been further developed by Thermo Electron Corporation. This development was an essential part of thermionic converter technology. Not only has the maximum operating temperature been extended to 1700°C , but the thermal conductance of the insulation has been significantly reduced.

The insulation that Thermo Electron is developing (AEC Contract No. AT(30-1)-3634), consists of thin metal foils separated by thin layers of high-purity oxides in granular form. The metal foils serve as thermal radiation shields, and the oxide layers provide thermal resistance barriers to prevent conduction between the metal foils. Typical heat flux test results are shown in Figure III-10.

The program accomplishments are as follows:

The chemical compatibility of high-temperature refractory insulation material combinations was investigated as a function of time and operating temperature.

Direct measurements were made of the heat-transfer characteristics of both planar and cylindrical insulation samples assembled using the most promising material combinations. Typical data are shown in Figure III-11.

The effect of pressure-loading on planar insulation performance was investigated. Sample results are given in Figure III-12.

An analytical and experimental investigation of edge losses was undertaken to identify the heat-transfer paths through

Figure III-10. Planar Heat Flux Test Results
(Mo-ThO₂ Insulation Spray Coated, Zero Loading)

Figure III-11. Comparison of Cylindrical and Planar Ta-ThO₂ Heat Flux Data (20, 40, and 80 Layer Samples).

Figure III-12. Heat Flux Versus Applied Pressure
(W-ThO₂, 40 and 100 layer Samples)

current edge designs and to suggest improvements in these designs.

Direct measurements were made of the heat losses associated with certain insulation support structures.

A series of tests was made to study the variations in insulation performance with time and to monitor the outgassing rate of the insulation over a prolonged period.

Preliminary designs were prepared for a static structure test assembly and an insulation demonstration system.

The reference design requires the heat source to be supported by the vacuum foil insulation. However, the pressure-loading on the insulation will be rather modest (less than 1 psi). Figure III-10 shows that the increase in thermal conductance due to this low pressure-loading is nominal.

3. Conclusion

From Table III-12, it is apparent that the Thermo Electron vacuum foil insulation developed under AEC sponsorship is superior to any other foil, powder or fiber insulation. There are several factors which dictate the use of vacuum foil insulation in implantable power sources for circulatory support devices:

- a. Reduction of radioisotope fuel inventory is mandatory because of cost, radiation and additional endogenous heat considerations. (Therefore the thermal insulation must be of the highest quality.)
- b. Power source volume must be minimized to permit implantation. (Vacuum foil insulation thicknesses of one-tenth of an inch should be adequate.)

- c. If the cylinder housing the reciprocating piston is used as a vacuum seal, there is no thermal loss penalty in making the vacuum seal above the unavoidable conduction loss along the cylinder wall.
- d. Vacuum foil insulation can be readily outgassed and can maintain high vacuum for the required times after "pinch off."
- e. The layered foils form an elastic unit that has good compressive recovery which is desirable for absorbing shock and vibration loads.

F. THERMAL ENERGY STORAGE

Heat can be stored at high temperature in the heat of fusion of a material. By storing thermal energy at the operating temperature of the thermodynamic converter (Rankine cycle engine) to provide for short term peak power demands, the average power supplied by the radioisotope can be minimized. The attendant savings in cost, radiation exposure and additional endogenous heat are significant.

The amount of thermal energy storage (TES) required depends on the load profile. However, thermal storage of 20 to 30 watt-hours should be ample for most activity schedules. The reference design described in Chapter II has 48 watt-hours of thermal energy storage which gives a comfortable safety factor. The additional TES material minimizes the problem of heat source over-temperature and also reduces the surface dose rates on the power source.

Thermal energy will be stored for use in a thermodynamic engine, which operates most efficiently when interacting with a heat source at a constant temperature. This requirement implies that an isothermal phase change or a reversible chemical reaction must be used to store thermal energy. Reversible chemical reactions invariably exhibit very small heats of reaction per unit mass and are rejected in the interests of low weight and volume. For the same reason, solid-phase transformations are also rejected. Evaporation and sublimation are accompanied by high energy release rates but suffer from the fact that they are associated with high pressures when the volume is held constant or with high volumes when the pressure is fixed. In addition, the process is isothermal only at constant pressure. These considerations essentially eliminate evaporation and sublimation from further consideration.

Melting of a solid is isothermal and the accompanying volume changes are very small. Several materials exhibit heats of fusion

high enough to allow the design of a fairly attractive system. The remainder of this discussion will consider only devices using fusion to store thermal energy.

The number of possible materials can be limited by specifying an acceptable temperature range. The melting point of the thermal energy storage material determines the radioisotope operating temperature as well as the working fluid inlet temperature to the thermodynamic converter. High temperatures are desirable because the cycle efficiency of the engine increases with increasing working fluid inlet temperature. However, thermal radiation heat losses from the engine heat and thermal conduction losses through the engine structure also increase with temperature. Chemical interactions and materials problems in general are aggravated with increasing temperature. Materials were surveyed between a maximum of 1550°F and minimum of 840°F. The maximum was chosen because of materials limitation and the minimum because cycle efficiency must be maintained.

Within the temperature range specified above, a large number of materials are available. Some of these have high latent heats of fusion, but their densities vary widely. As a result, the materials with high storage capacity per unit weight are not always the best in terms of storage capacity per unit volume. Weight and volume are more important in the case where the thermal energy storage device is rechargeable than when the device is used in conjunction with a radioisotope heat source. The material used obviously has to be stable at its operating temperature within this particular environment. It must not decompose or interact with its container.

1. Evaluation of Materials

A compilation of elements and compounds with melting points in the 840 - 1550°F temperature range is shown in Table III-14.

TABLE III-14

MELTING POINT AND ENERGY STORAGE CAPACITY
FOR MATERIALS MELTING BETWEEN 840°F AND 1550°F

TES Material	Storage Capacity		Melting Point, °F
	cal/gm	cal/cc	
LiF	248	643	1550
LiH	610	357	1270
FeCl ₂	81	240	1245
V ₂ O ₅	86	290	1235
Al	93	250	1220
Sb ₄ O ₆	45	230	1210
Mg	86	140	1205
Sb	39	260	1170
LiCl	112	231	1140
CdCl ₂	29	117	1055
NaCN	100	200	1047
LiB _r	48.4	167	1020
PbCl ₂	21	122	910
BeCl ₂	56	106	895
LiOH	208	298	885
LiI	26.2	106	875
B ₂ O ₂	75	140	840
Te	33	216	840
LiF/LiCl	165	362	930
Eutectic			

This table also includes the latent heat of the materials as a function of unit weight and unit volume. These same data are shown graphically in Figures III-13 and III-14.

From Table III-14 it is apparent that lithium hydride has the highest gravimetric thermal storage capacity. However, both lithium fluoride and lithium fluoride/lithium chloride eutectic have higher volumetric thermal storage capacities.

The fluoride and hydroxide fall at the opposite ends of the temperature range (1250°F and 885°F, respectively). The alkali metal halides all have very similar physical and chemical properties. Also, the halides of any one alkali metal differ chemically and physically only in degree. As a consequence of this similarity, these salts are completely soluble, and the thermal properties of the resulting solutions are well-behaved functions of the mol-fractions of the two components. The melting point of a mixture of halides, when plotted versus mol-fraction, forms a concave curve, i.e., the depression of the melting point caused by the addition of a lower-melting compound is greater than proportional. For purposes of the present discussion, a conservative estimate of the behavior of LiI-LiF mixtures was made by assuming a linear relationship between mol-fraction and melting point. This line is shown in Figure III-13 joining the melting points of LiI and LiF. Inspection of Figure III-13 shows that, except for LiOH and LiH, mixtures of LiF-LiI have the highest storage capacities per unit volume of any of the compounds listed.

Figure III-14 is a plot of the energy storage capacity per unit weight for the substances listed in Table III-14. Again a line has been drawn to indicate the storage capacity of LiF-LiI mixtures. LiH and LiOH again show considerably higher capacities than any other substances, while mixtures of LiF-LiI are equal to or better than the remaining substances. Because of the obvious weight and volume

Figure III-13. Energy Storage Capacity Per Unit Volume versus Melting Point, °F, for Various Materials

Figure III-14. Energy Storage Capacity Per Unit Weight versus Melting Point, °F, for Various Materials

advantages of the lithium compounds, the remainder of this discussion will be confined to them.

Use of a composite material at any composition other than that of the eutectic involves a temperature stability problem. All compositions other than the eutectic melt over a temperature range. This means that the heat source temperature would vary. A small variation is not serious, but a large variation in source temperature would likely create severe control problems.

2. Physical and Chemical Properties of Selected Lithium Compounds

a. Lithium Hydride

Lithium hydride has an outstanding gravimetric thermal storage capacity. While all metal hydrides are somewhat unstable, LiH is distinguished from the other light metal hydrides by its relative stability. The hydrides of Na, K, and Ca all decompose more readily, as do the complex borohydrides of the type $M(BH_4)$.

At elevated temperature LiH dissociates to give lithium metal (melting point $180^\circ C$) and hydrogen. The pressure of hydrogen in equilibrium with LiH at its melting point ($1250^\circ F$) is 30 torr. In principle, this dissociation can be arrested by providing an atmosphere of hydrogen over the LiH. However, to do this requires containment of the hydrogen, and hydrogen permeates most materials rapidly at $1250^\circ F$. The degradation of vacuum foil insulation can probably be solved by "gettering" the insulation package. However this complicates system design and fabrication.

b. Lithium Hydroxide

Like lithium hydride, lithium hydroxide is somewhat unstable and tends to dissociate according to the reaction $2 LiOH \rightarrow Li_2O + H_2O$. At the melting point ($950^\circ F$) the associated free energy change is $+15.5 \text{ k cal/mol}$, from which the equilibrium pressure of water above

LiOH at 950°F is calculated to be 2.4×10^{-2} torr. This is a low pressure and can easily be tolerated. However, containment requires that there be no sink for the products of the above reaction - that is, that neither LiO₂ nor H₂O react with the container. While nickel and the noble metals will probably contain the water, there is no data on reactions with LiO₂.

c. The Halides of Lithium

The halides of lithium are among the most stable compounds known. Table III-15 is a compilation of the free energies of formation of these compounds at 950°F.

TABLE III-15
FREE ENERGIES OF FORMATION OF LITHIUM HALIDES

Compound	ΔF k cal/mol
LiF	129
LiCl	85
LiBr	74
LiI	62

The fact that lithium halides can be contained in nickel containers is shown by computing the free energy change for the reactions shown below:

All free energy changes are large positive numbers, indicating that the reactions shown will not take place spontaneously.

The vapor pressures of the halides of lithium are shown in Table III-16 at their respective melting points.

TABLE III-16
VAPOR PRESSURE OF HALIDES OF LITHIUM

Li F	at	848°C	2.1×10^{-2} torr
Li Cl	at	614°C	2.1×10^{-2} torr
Li Br	at	550°C	8.6×10^{-3} torr
Li I	at	469°C	4.2×10^{-4} torr

The above values clearly indicate that at operating temperature these materials present no pressure containment problem.

d. Mixed Halides of Lithium

An initial survey of the literature revealed relatively little data on the mixed systems of the halides of lithium. However, the data on LiCl-LiF given in Figure III-15 shows a eutectic at 30.5 mol per cent LiCl with a melting point of 935°F. This material has a high heat of fusion (362 cal/cc) and a convenient melting point (935°F). The heat of fusion and melting point have been confirmed by measurements at Thermo Electron. This material has proven easy to contain in 304 SS and has a volumetric heat storage compatibility equal to that of lithium hydride.

e. Hydrogen Containment

The containment of lithium hydride presents a difficult problem due to hydrogen diffusion. Studies by TRW (Ref. 3) that tungsten is the high temperature material resistant to hydrogen diffusion. Tungsten, however, presents some fabrication problems. The next alternative would be a composite of Haynes 25 base alloy metal with

Figure III-15 System LiCl - LiF. Solid line, experimental, dashed line, Clausius - Clapeyron limit.

a solarmic glass coating, which is some 70 times more permeable to hydrogen than is tungsten.

The optimum TES design calls for a cylindrical container. For this shape, the diffusion of hydrogen is expressed mathematically by:

$$\frac{dq}{dt} = \frac{2 \pi L K}{\ln \frac{b}{a}} [p_1 - p_2] \quad (\text{III-1})$$

where dq/dt is the hydrogen flux in cm^3 (STP) per hour, L is the effective cylinder length in cm, k is the permeability of the container material in cm^3 (STP) $\text{hr}^{-1} \text{cm}^{-1} \text{atm}^{-1/2}$, b and a are the container outside and inside diameters, respectively, in inches, and p_1 and p_2 are the inside and outside hydrogen pressures, respectively, in atm. Values of k from the above report for LiH at its melting point are:

$$\begin{aligned}\text{Tungsten: } k &= 1.8 \times 10^{-5} \text{ cm}^3 (\text{STP}) \text{ hr}^{-1} \text{ cm}^{-1} \text{ atm}^{-\frac{1}{2}} \\ \text{Haynes/Solaramic: } k &= 1.3 \times 10^{-3} \text{ cm}^3 (\text{STP}) \text{ hr}^{-1} \text{ cm}^{-1} \text{ atm}^{-\frac{1}{2}}\end{aligned}$$

The tungsten is obviously better for resistance to hydrogen diffusion. Current techniques indicate that tungsten is workable for this application. A sample calculation indicates the magnitude of the hydrogen loss. Consider a TES volume which provides 240 watt-hours of energy. The total TES cylinder length, 8.9 cm, was used as the effective length L to simplify the calculations. For LiH at its melting point, the hydrogen dissociation pressure is 28 mm Hg. Thus $p_1 = 0.037$ atm, a constant. The outside pressure, p_2 , depends on the gas buildup. Initially it is extremely low ($< 10^{-4}$ mm Hg). Assuming $p_2 = 0$ initially, the value of p_2 at any time is

$$p_2 = \frac{nR_o T}{V} \quad (\text{III-2})$$

where n is the number of gram-mols of hydrogen in the void space, R_o is the universal gas constant, T is the void temperature (assumed to be the average of the TES surface temperature and the sink temperature) and V is the void volume. If the hydrogen flux is expressed in gram-mols per hour and TES design dimensions are inserted into Equations III-1 and III-2, the resulting differential equation is:

$$\frac{dn}{dt} = \frac{4.5 \times 10^{-8}}{\ln(1 + 0.606\delta)} [0.1925 - 15.7 \sqrt{n}] \quad (\text{III-3})$$

where δ is the container thickness in inches. This equation has been solved by integration for various container thicknesses. The results are shown graphically in Figure III-16. It can be seen that hydrogen diffusion is not a problem in terms of loss of thermal

Figure III-16. Pressure Build-up in Vacuum Insulation
Void Space

storage capacity. After 700 hours the percent loss of hydrogen by weight is considerably less than one percent. However, a major problem is evident from the standpoint of pressure rise in the void volume within 40 layers of vacuum foil thermal insulation. After only 100 hours, the pressure is up to 5 mm Hg. The vacuum foil insulation requires pressure below 10^{-4} mm Hg to operate properly.

The data in Figure III-14 overstate the problem because the hydrogen can escape to other voids within the power source. However, even if the other voids are a thousand times higher volume, the magnitude of the hydrogen pressure buildup is fatal to the use of vacuum insulation unless the insulation is "gettered" (i. e., combined chemically).

f. Hydrogen Exclusion from Vacuum Foil Insulation

Assuming that hydrogen leaves the lithium hydride container at a low enough rate to be acceptable from a thermal energy storage viewpoint; this rate would still, most likely, be unacceptable from the point of view of the vacuum foil thermal insulation. Thus, hydrogen must be excluded from the vacuum foil insulation.

One means of doing this is to take advantage of the easy diffusion of hydrogen and bleed it out of the insulating space. Palladium has a high solubility for hydrogen and is quite permeable to it, even at body temperatures.⁴ The permeation rate is pressure dependent, however, and it seems unlikely that a useful rate could be achieved with the very low pressure head available, about 10^{-6} torr. Furthermore, the most permeable palladium membranes are very susceptible to poisoning and polariaation, either of which could reduce the rate substantially.

A better approach appears to be to combine the leaked hydrogen with a "getter" which will keep its effective pressure very low. Here

the significant constraint is that the getter must be active towards small traces of hydrogen at a temperature low enough so that the pressure of hydrogen over it is still less than 10^{-6} torr. Such a material is calcium. Moeller⁵ reports that calcium absorbs hydrogen rapidly at 150° to 300°C forming the stoichiometric saline hydride CaH_2 , with a small decrease in volume. The dissociation pressure of H_2 above CaH_2 at 200°C is $<10^{-7}$ torr, which is acceptably low. The charge of lithium hydride is anticipated to be about 62 moles; consider a degradation of 1.6% or 1 mole to be acceptable. Then, since $2 \text{LiH} \rightarrow 2 \text{Li} + \text{H}_2$, but $\text{H}_2 + \text{Ca} \rightarrow \text{CaH}_2$ half a mole of calcium, or 20 gms of calcium, will be sufficient to combine with all the hydrogen released and still keep the pressure in the insulation down below 10^{-7} torr. Twenty grams is less than 1 ounce and seems to be an acceptable weight penalty.

It should be possible to contain a (solid) film of calcium at a point in the insulation having a temperature of 200°C . The calcium will incidentally also getter oxygen and nitrogen to form the stable compounds CaO and Ca_3N_2 . However, no problem is anticipated in excluding O_2 and N_2 from the insulation space.

Niobium appears to be a promising bonding agent for the hydrogen. However the rates of compound formation are unknown.

G. RADIOISOTOPE FUEL CAPSULE

The problems associated with designing and fabricating the fuel capsule depend on the choice of radioisotope fuel. Thulium and promethium oxides are high temperature, high stability fuel forms and present no unusual compatability or fabrication problems. In order to reduce surface contamination, a double encapsulation should be used.

Plutonium-238 (Pu-238) is an ideal fuel from the point of view of half life and power density. However an inexpensive, high power density, high temperature, stable fuel form with a low neutron output does not appear to exist. The metal could become molten in a fire. The nitride melts at a high temperature, but is pyrophoric. The oxide is satisfactory from the viewpoints of power density, container compatability, stability and melting point. However it requires an isotopic depletion of the oxygen in order to eliminate the alpha-neutron reactions typical of low atomic number materials. The second problem with using an alpha emitter such as Pu-238 is the buildup of helium pressure. The helium results from the attachment of electrons to arrested alpha particles.

The detailed design of the fuel capsules is beyond the scope of present study. However, because of the generally favorable properties of Pu-238, some further consideration will be given to the design of a Pu-238 fuel capsule. The capsule components are 1) Pu-238 in some fuel form yet to be specified, 2) void volume for helium buildup, 3) liner capsule for materials compatability (may not be required, 4) high strength, high temperature encapsulation container, and 5) photon shield.

The design of the fuel capsule shown in Figure III-17 is conventional except that the tantalum shield is placed only around the fuel and not around the fuel and void volume. This configuration

Figure III-17. Haynes 25 Outer Capsule of 25 Watt Plutonium-238 Heat Source

decreases the shield weight by approximately one half. The fuel volume is sized for plutonium oxide. Depleted oxygen is necessary to reduce the α - n reactions to the level of spontaneous fission. A small hole, or controlled leak, releases the helium from the fuel volume to the void volume. The void volume is equal to the fuel volume. Although no optimization of the void volume was made, this choice is typical of SNAP radioisotope thermoelectric generators design. The encapsulation material is Haynes 25 with a wall thickness of 0.100 inch. This material is corrosion resistant and has good strength at high temperature.

Most of the gamma radiation from a Pu-238 source is low energy. Tantalum is used to shield the low energy photon radiation from Pu-238 and daughters of the Pu-236 impurity. Tantalum is an effective gamma shield because of its high density and high atomic number. No credit for neutron attenuation other than geometry. Little can be done to shield the high energy gamma and neutron radiation without the heat source becoming prohibitively heavy.

The 16 and 24 watt Pu-238 capsules presently used in the joint NHI/AEC Additional Endogenous Heat Study were designed as typical fuel capsules for an implantable power source. Although the fuel is metal, the design of these capsules is relevant to the present study.

The fuel capsule was designed for the 24 watt source, and the same design then used for both the 24 and 16 watt sources. The capsules were designed for a five-year life, although the life will be limited in practice to no more than two years. The design consists of an inner thin-walled, welded tantalum capsule containing the metallic plutonium fuel. This inner capsule is sealed in a second, thick-walled, welded Haynes-25 capsule with full penetration welds. The capsule design is as shown in Figure III- 17. The inner tantalum container is compatible with plutonium metal up to 400°C

and also includes a void volume equal to 1.1 times the fuel volume for helium gas pressure build-up. (for the 25-watt capsule) while the outer container is the high strength container that resists the internal pressure. The ultimate strength of the Haynes at 400°F is 1.25×10^5 psi, and the yield strength is 4.5×10^4 psi. The maximum design stress due to internal pressure at five years is 3.3×10^4 psi at 423°C. A trade-off among the variables of power, physical size, weight and capsule container stresses was made. It is feasible to encapsulate the required 25 thermal watts of plutonium-238 metal in a capsule approximately 1.54 inches O. D. by 0.60 inches long. For the purpose of this study, a right cylindrical geometry was assumed.

Several different capsules were considered, each with different cap and wall thicknesses, and cap to wall thickness ratios. Included were several capsules whose O. D. and length were varied. The capsule chosen, falls within the size, weight and thermal output suitable for fueling an implantable power source. Pressure within the capsule is caused by the buildup of helium resulting from the alpha decay of the radioactive material. Proper design of a capsule involves careful consideration of the pressure, time, temperature relationships so that sufficient space (void volume) is designed into the capsule. It is convenient to calculate this pressure, using one cm^3 of void volume, then the pressure for any void volume can be calculated by ratio. The required equation is:

$$P = \left(\frac{w}{M} \right) \left(\frac{RT_o}{V_v} \right) \left(e^{-\lambda t} \right) \left(1 - e^{-\lambda t} \right)$$

where: P = pressure in atmospheres
 w/M = mole fraction active material
 R = gas constant, 82.0 units

T_o = maximum temperature at the design point, $^{\circ}\text{K}$

t = half-life fraction

λ = decay constant, 0.693

V_v = void volume

The values inserted into the equation (except for T_o and t) must be determined. T_o was chosen as slightly above the interrupted blood flow temperature of 700°F , i. e., 700°K . This value is, therefore, conservative and higher than is to be expected for any extended period of time. The value of t was chosen as 5 years and is assumed to be the maximum capsule lifetime. Substituting values into the equation, one obtains:

$$P = \left(\frac{44}{238} \right) \times \left(\frac{82 \times 700}{1} \right) \times \left(e^{-(.693 \times .058)} \right) \left(1 - e^{-(.693 \times .058)} \right)$$
$$= 407 \text{ Atm or } 5980 \text{ Psi}$$

The stress in the capsule wall and cap which are a function of the capsule pressure are calculated from the following equations.

$$S_w = \delta P$$

Where S_w = wall stress, psi

$$S_c = \frac{K P r_1^2}{t_1^2}$$

S_c = cap stress, psi

K = constant, 0.75

r_1 = capsule inside radius

r_o = capsule outside radius

t_1 = cap thickness

$$\delta = \frac{(r_o/r_1)^2 + 1}{(r_o/r_1)^2 - 1}$$

These equations are taken from Marks' "Mechanical Engineers Handbook."

The values of capsule stresses at $T = 700^{\circ}\text{K}$ and $t = 5$ years for the initial capsule design are:

$$S_w = 14,000 \text{ psi}$$

$$S_c = 24,000 \text{ psi}$$

Figure III-18 shows the stress time plot which was taken from extrapolated data. The capsule would not be expected to accumulate 2% elongation in the cap after more than five years at 700°F . It also would not rupture before five years at 700°F from stress alone. The safety factor for 2% creep in five years is about 1.6. The safety factor for rupture in five years is about 2.4. (Note: Operation above 800°F is not permissible because of fuel-liner compatibility.)

Since the capsule will be opened at some time prior to five years after closing, the capsule strength is more than adequate for this application.

Haynes-25 metal for the outer container was chosen because it has oxidation resistance at the temperatures involved and meets the requirements of compatibility. Tantalum was chosen for the inner capsule for compatibility with the fuel form and the outer container. The materials and weld closure for these capsules are typical of those used for other Pu-238 Fuel Capsules designed and fabricated at Mound Laboratory for other applications with more severe environmental conditions.

Figure III-18. Elongation and Rupture Time for Haynes-25 at 700° F
(Extrapolated Data)

H. HEAT SOURCE SUBSYSTEM

The heat source subsystem consists of the following major components:

- Radioisotope Fuel Capsule
- Thermal Energy Storage Material and Container
- Vacuum Foil Insulation
- Boiler

The boiler is actually part of the engine subsystem, but is cocooned within the heat source subsystem. All the other subsystem components have been considered previously.

The heat source subassembly of the reference design described in Chapter II is based on the successfully demonstrated thermal energy prototype shown in Figure III-19. This prototype unit was constructed as part of the NHI sponsored Thermal Energy Storage contract. Although the prototype TES is sized for electrical recharge and is several times the TES volume required for a radioisotope fueled implantable power source, the choice of TES material (LiF/LiCl eutectic) and thermal insulation (vacuum foil) are identical to those of the reference design given in Chapter II.

Some notable features of the TES prototype design are:

The thermal storage material is a eutectic mixture of LiF and LiCl contained in a 304 stainless steel (total material volume of 475 cc).

The thermal insulation is a right cylinder of 40 layers of nickel foil (each 0.5 mil thick) separated by zirconium oxide particles.

Cooling tubes, calorimeter and guard heater are incorporated into the design for test purposes.

The entire unit is attached to a vacuum system (oil diffusion pump) to provide a vacuum of less than 10^{-4} mm Hg for testing.

The TES prototype has demonstrated the feasibility of the reference design heat source subassembly. Hardware testing has:

Demonstrated thermal energy storage of 140 watt-hours and subsequent release rates of 20 watts for approximately 7 hours.

Shown the effectiveness of thermal insulation using vacuum foils (see Figure II-9).

Demonstrated that transient TES surface temperatures are stable enough for good engine control under anticipated heating and cooling rates. The TES surface temperature as a function of time, parametric in power level, is given in Figures III-20 and III-21 for the cases of cooling and heating, respectively. Since a radioisotope fueled implantable power source requires much less TES material and the transient heating and cooling rates will be much smaller than those given in Figure III-20 and III-21, it is clear that the temperature stability of the heat source will not be a problem.

Confirmed the fabrication and short-term containment of the LiCl-LiF eutectic.

The heat source subassembly of the reference design would require the following modifications to the prototype TES unit:

Reduction of TES volume by factor of four.

Fabrication of TES container with tapered hole to receive radioisotope fuel capsule.

Brazing of the monotube boiler around TES container.

"Pinching off" foil insulation package so that static vacuum is maintained.

Figure III-19. TES Prototype Unit

Figure III-20. TES Surface Temperature vs Time (Cooling)

Figure III-21. TES Surface Temperature vs Time (Heating)

The measured insulation loss of 2.7 watts at the melting point of the LiF/LiCl eutectic (see Figure II-9) is higher than would be experienced with the radioisotope fueled heat source. The surface area at high temperature in the prototype test is 68.2 square inches which is almost double that of the reference design. Taking the different surface areas into account gives a normalized insulation loss of 1.5 watts.

The thermal insulation loss can be reduced further by 1) use of more foils, 2) refinement of vacuum foil edge construction and 3) lower emissivity foils. Based on modest improvements by these three factors, the direct thermal loss through the vacuum foil thermal insulation should be no more than one watt. The AEC is currently sponsoring work at Thermo Electron (Contract No. AT(30-1)-3634) to improve vacuum foil thermal insulation for radioisotope power systems.

I. THERMAL LOSSES

The smaller the power supply, the higher the relative thermal losses. The thermal loss from the heat source subassembly have been discussed previously. On the basis of prototype tests and reasonable extrapolations in vacuum foil insulation technology, it should be possible to reduce the direct heat source insulation loss to below one watt.

In analyzing the total heat loss of the power source, three types of thermal transfer must be considered: conduction, convection, and radiation. Because of the relatively low operating temperature, conduction losses will predominate.

The conduction heat paths are along the engine cylinder, the engine exhaust pipe and cylinder support. Conduction loss through the steam piston and shaft due to piston ring contact are negligible. An electrical analog of the engine conduction losses is shown in Figure III-22. In the model, temperature is analogous to voltage, heat flux is analogous to current, and the invers of thermal conductance is analogous to electrical resistance. The heat source (boiler) temperature is 900°F and the heat sink (condenser) temperature is 140°F . Heat conduction through the piston rings has been neglected. The quantities are defined as follows:

T_{EX}	$\equiv$	temperature of exhaust pipe at outlet
T_{B}	$\equiv$	boiler temperature
T_{c}	$\equiv$	condenser temperature
R_{ins}	$\equiv$	thermal resistance of vacuum foil insulation
R_{uc}	$\equiv$	thermal resistance of upper cylinder
R_{LC}	$\equiv$	thermal resistance of lower cylinder

- R_{EX} $\equiv$ thermal resistance of exhaust pipe
 $R_{cyl\ sup}$ $\equiv$ thermal resistance of cylinder support
 ΣQ $\equiv$ total conduction heat loss
 Q_{ins} $\equiv$ heat flux through vacuum foil insulation
 Q_{uc} $\equiv$ heat flux through upper cylinder
 Q_{LC} $\equiv$ heat flux through lower cylinder
 Q_{EX} $\equiv$ heat flux through exhaust pipe
 $Q_{cyl\ sup}$ $\equiv$ heat flux through cylinder support

Figure III-22. Conduction Analog with Constraints.

Titanium alloy is used as the basic structural material of the engine. It was chosen because of its high strength-to-weight ratio, low thermal conductivity, compatibility with water, machinability, fabricability, and demonstrated reliability in engines.

Because of the relative magnitude of the thermal resistances, R_{ins} and R_{EX} effectively approach infinity and consequently Q_{ins} and Q_{EX} approach zero. It can be shown that the constraint $T_{EX} = 200^{\circ}F$ (the temperature of the exhausted steam) insures that the cylinder wall temperature is always hotter than the steam it encloses. This minimizes convection losses. Table III-17 summarizes the results of the analog analysis.

TABLE III-17
ENGINE CONDUCTION LOSSES

$R \equiv \text{length of heat flow} / (\text{conductivity} \times \text{cross sectional area})$				
R_{ins}	$\rightarrow$	∞		
R_{uc}	=	$490 \frac{^{\circ}F}{\text{watt}}$	(.010" thick cylinder wall)	
R_{LC}	=	$42 \frac{^{\circ}F}{\text{watt}}$	(.070" thick cylinder wall)	
R_{EX}	$\rightarrow$	∞		
$R_{cyl \sup}$	$\cong$	0		
ΣQ	=	Q_{uc}	=	$Q_{LC} = Q_{cyl \sup} = 1.43 \text{ watts}$

Convection losses will occur within the steam cylinder due to the temperature difference between the steam and the cylinder wall-piston and in the exhaust line due to the temperature difference between the exhausted steam and the exhaust pipe-piston. The coefficient of convection heat transfer is difficult to determine except by experiment. In the case of the steam engine cylinder,

the steam temperature is a function of time and this further complicates the problem. The cylinder wall and exhaust outlet should be hotter than or at least the same temperature as the steam in order to obtain a high engine efficiency. The cylinder wall temperature can be adjusted over a wide range by adjusting the relative thermal conductances of the engine components. The convection loss is estimated as less than 0.5 watts.

The radiation losses will be small because 1) the inlet steam temperature is only 900°F ; 2) the cylinder heat has restricted area; 3) the cylinder is plated with low emissivity chromium and 4) the cylinder is jacketed with vacuum foil insulation. Figure III-23 shows one place where radiation will occur — within the steam engine cylinder. Most of the radiation will occur between surface 1 and surface 4 and between surface 2 and surface 3. This is because the view factor and temperature difference is the highest between these surfaces. A complication is that surfaces 3 and 4 are not always exposed because of the motion of the piston. Based on the cylinder dimensions and emissivity of chromium, the radiation loss is approximately 0.4 watts.

Figure III-23. Radiation Within Steam Cylinder

The total thermal losses within the implantable power source for the reference design are summarized in Table III-18.

TABLE III-18

IMPLANTABLE POWER SOURCE THERMAL LOSSES

Loss Mechanism	Thermal Loss (watts)
Vacuum foil insulation	1.0
Engine conduction	1.4
Engine radiation	0.4
Engine convection	<u>0.5</u>
TOTAL	3.3

Thermal loss studies indicate that the overall efficiency of the engine at 900°F is approximately the same as that of a comparable engine operating at 1250°F. This apparent anomaly arises because, although the higher inlet temperature increases the cycle efficiency by approximately 10 percent, it also increases thermal losses. In addition, the structural materials required at these temperatures have higher thermal conductivity than titanium so that no improvement in overall efficiency is achieved with increased temperature. The higher temperature also raises more difficult problems with materials, insulation, TES and fuel containment. Comparable performance and efficiency can be realized more easily with a low-temperature system than with a high-temperature system.

A more detailed analysis of thermal losses is given in Appendix L.

J. BEARINGS

The average bearing load of the reference design is minimized by directly coupling the steam and hydraulic pistons. Thus the primary engine loads are not transmitted through the journal bearings. The journal bearings need only take the loads associated with accelerating the flywheel. These loads are small because the flywheel need only supply enough energy to return the power piston and crack the inlet valve. The exhaust valve stays open during the entire return stroke so that the flywheel does not have to supply energy to do compression work.

The bulk of the discussion in this section centers around the reference design which uses water as the working fluid. Since the working fluid doubles as a lubricant in all the Rankine cycle concepts, much of this section is devoted to water lubricated bearings. However, the legitimate emphasis on water systems should not obscure the high potential of organic systems.

As discussed previously, organic Rankine engines with regenerators are quite efficient. They are also operated at low temperatures which decreases thermal losses. Another important point is that many organic working fluids also act as lubricants or can dissolve lubricants without detriment to the cycle efficiency. Common refrigeration systems use organic working fluids with dissolved lubricants. These sealed motor-compressor units often last more than ten years. Paxton has constructed a Rankine cycle automobile engine in which the working fluid also acts as the lubricant. Much of this information is proprietary and is difficult to evaluate. However, organic working fluids doubling as lubricants hold a great deal of promise for future implantable power source engines.

The Rankine cycle engine power source, as described herein, requires three journal bearings. Journal bearings are used at both ends of the shaft that joins the hydraulic pump piston to the outside of the flywheel eccentric. The flywheel rotates from the third journal bearing which is connected to the axis of rotation of the eccentric. The eccentric delivers power to the feedwater and sump pumps. As the eccentric swings around, it propels the pump shafts by making contact during part of the rotation cycle. The pumps are returned by a spring. The contact points between the eccentric and pump shafts constitute two more bearing surfaces. The rings of the steam piston, pump pistons, and high pressure reservoir piston constitute the surfaces of what are essentially guide bearings (although rings and liners are also present to minimize leakage).

Oil or grease lubricated bearings would require sealing from a water working fluid. If hydrocarbon lubricated bearings are not sealed, they will run dry. In addition, the oil or grease would contaminate the water in the system and possibly plug water lines or cause valves to leak. For the long operational life required for the implantable power source, it appears highly unlikely that hydrocarbon lubricated bearings can be adequately sealed to prevent contamination and maintenance.

Water lubricated bearings are of three types: hydrostatic, hydrodynamic, and water fed. The water used for lubrication would be drawn from the common system water supply. Since water would be used for both lubrication and cycle working fluid there would be no sealing problem. Water leaked from the bearings would be scavenged by the sump pump. However, water is not as good a lubricant as oil or grease because it is less viscous. Compensation would have to be made for this fact.

Figure III-24 shows a plot of coefficient of friction vs. characteristic number for journal bearings. The ranges for hydrostatic and hydrodynamic bearings are indicated and the characteristic number is defined. The characteristic number is the critical factor in determining the feasibility of a particular bearing design.

Hydrostatic bearings have a low characteristic number. Since the viscosity of the lubricant (water) and the engine RPM are predetermined, a low characteristic number can be obtained only by a high bearing pressure, p . Therefore, water must be pumped to the journal bearings under high pressure. The journals will then ride atop a layer of water. Hydrostatic action means that a large volume of fluid will have to be pumped to the bearings and then scavenged from the central cavity of the capsule. Therefore the power to maintain hydrostatic bearings would be a significant parasitic loss. Since maximum efficiency and simplicity are very important, the hydrostatic bearing system is undesirable.

In hydrodynamic bearings, the bearing shaft rides atop a film of lubricant formed by hydrodynamic action. The characteristic number required is higher than that for hydrostatic bearings. Therefore, for the same bearing pressure, p , the hydrodynamic bearings require a higher Zn . The viscosity, Z , is predetermined so that the magnitude of Zn must be determined by the bearing shaft velocity, n . However, the magnitude of n required for hydrodynamic bearings using water lubricant is much higher than practical design levels. An alternate solution would be to reduce the bearing pressure in order to reduce proportionally the required bearing shaft velocity. However, this would require bearing areas much too large to be practical. Because of these problems, hydrodynamic bearings have been eliminated as a solution.

- a-b Range of Hydrostatic or Thin-Film Lubrication
- b-c Boundary Lubrication or Transition Between Hydrostatic and Hydrodynamic Lubrication.
- c-d Range of Hydrodynamic or Thick-Film Lubrication
- Z = Absolute viscosity of lubricant (centipoise)
- n = Relative RPM of bearing and journal
- p = Bearing pressure (psia)

Figure III-24. Hydrostatic and Hydrodynamic Bearing Ranges

Water-fed bearings are the third type of water-lubricated bearings to be considered. In order for water-fed bearings to operate efficiently, several requirements must be fulfilled: (a) the bearing must be smooth and hard to reduce friction and prevent wear, (b) water must readily adhere to the surface of the bearing material and thus lubricate it; (c) adequate water must be supplied to the bearings to keep them wet. From these requirements it is seen that proper function of water-fed bearings depends quite sensitively on the material used. Graphite, carbides, and borides are outstanding bearing materials for use in water.

Graphite, although not as hard as any boride or carbide, substantially reduces friction when used in water-fed journal bearings. Two effects are observed as the journal rotates. First, the water adheres to the graphite surface forming a thin film of lubrication. Second, the surface layer of graphite slides over the graphite layer beneath it giving a roller bearing effect. Graphite bearings will require testing to establish their suitability for the power source application. Preliminary water lubricated graphite bearing tests at Thermo Electron have been encouraging.

Outgassed carbide or boride bearings are quite hard. It is probable that these bearings could achieve a life of five or even ten years. All the requirements for water-fed bearings can be fulfilled. Water-fed carbide or boride bearings are attractive because:

- (1) These materials can be made very smooth.
- (2) Carbides and borides are extremely hard and do not distort, thus allowing the bearing surface to remain smooth under load.
- (3) The area of bearing surface contact can be made small.

(4) Water will adhere to the surface of the material and form a thin-film promoting lubrication.

(5) Carbides and borides may be outgassed to better enable water to adhere to the bearing surface, especially at high temperatures.

Many carbide and boride compounds are available for bearing application. Among them are tungsten carbide, niobium carbide, tantalum carbide, boron carbide, titanium boride, and zirconium boride. Tungsten carbide approaches the hardness of diamond. It has high compressive strength and is oxidation resistant up to 1200°F . There is some evidence to show that dissimilar materials in contact with each other sometimes produce lower friction. There is also evidence that at temperatures above 1000°C , borides or carbides on graphite yield low coefficients of friction. However, the bearings in the heart power supply will be operating at a temperature well below 1000°C . Therefore the best type of journal bearings for this application appear to be water-fed carbide or boride bearings. Water feed could be provided by the hydraulic pump. The volume of water required per pump cycle would be a negligible fraction of the total pump volume.

As mentioned previously, the flywheel eccentric propels the feedwater and sump pump pistons by making contact during part of the rotation cycle. A hard wear resistant material is needed for the contact points. The best material for this application appears to be either a carbide or a boride compound. The flywheel eccentric and the tips of the pumps shafts could be made of this material. Water feed could be provided by pump piston leakage.

The leakage analysis given in Appendix M shows that the steam piston requires rings to limit leakage. The pistons of the hydraulic pump feedwater pump, sump pump, and high pressure

reservoir do not need rings. Leakage can be limited by using a simple straight piston inside of a cylinder liner.

The rings and liners are bearing surfaces. The friction between the rings and steam engine cylinder and between the liners and pistons of the other pumps must be small to promote a high mechanical efficiency.

First consider the friction of the rings and liners. As described in Chapter II, the steam rings will be graphite and will bear against a chromium plated cylinder. The liners will also be graphite and will be mated to steel pistons. The coefficient of friction developed between graphite and its mating material depends largely upon the type of film and interface built between and on the mating surfaces. This is influenced by the graphite grade, mating material, surface speed, load and fluid environment. When a hydrodynamic film is maintained, as in the case of the liners, the coefficient of friction can be less than .01. When this film is broken and boundary lubrication exists, as is the case with the steam piston rings, the coefficient of friction is between .01 and 3 depending on the mating surface and operating temperature. For a mating surface of chromium and an average operating temperature of 550° F the coefficient of friction will be between .01 and .1. Even under these conditions galling or welding is not possible with graphite. Coefficient of friction is normally higher during early periods of operation when film and interface are being developed. Break-in periods may require hours or only seconds, depending on operating conditions encountered.

The wear rate behaves like the coefficient of friction. It is initially high and then slows to an insignificant amount once the graphite is "broken in". Experience with the larger steam engines has shown that a liner and ring life of five years is likely.

Other properties which recommend graphite as a component in mechanical seals are:

1. Dimensional stability over a wide range of temperatures
2. Low permeability
3. Resistance to thermal shock
4. Easily fabricated by molding and machining.

K. FLYWHEEL

The purpose of the flywheel is both to smooth the dynamic response of the engine and to return the steam piston on the upstroke. By adjusting the flywheel inertia together with the high pressure reservoir compliance, the desired natural frequency can be achieved (see the section on engine controls). In order to return the steam piston, the kinetic energy stored in the flywheel must be sufficient to overcome the friction of the moving engine parts and open the intake valve. This will require less than 20 percent of the total energy of the power stroke.

In analyzing the flywheel, the following parameters are defined:

Q_o	$\equiv$	nominal flow rate of hydraulic pump
V_o	$\equiv$	nominal hydraulic reservoir stroke volume for one heart beat
P_o	$\equiv$	nominal hydraulic reservoir pressure
r_o	$\equiv$	mean heart rate (beats/sec.)
f_o	$\equiv$	mean frequency of piston (cycles/sec.)
I	$\equiv$	inertance of flywheel as seen by fluid
I'	$\equiv$	rotary mechanical inertia as seen by the engine
A_H	$\equiv$	area of hydraulic pump piston
S_H	$\equiv$	stroke of hydraulic piston

Mean energy in $I \geq .2 \times$ energy in one power stroke or

$$\frac{1}{2} I Q_o^2 \geq .2 \frac{r_o}{f_o} P_o V_o, \quad I \geq \frac{.4 r_o P_o V_o}{f_o Q_o^2} = \frac{.4 P_o}{f_o r_o V_o}$$

$$I = \frac{4\pi^2 I'}{S_H^2 A_H^2}$$

The following parameter values have been chosen for the reference design:

$$\begin{aligned} r_o &= (70/60) \text{ beats/sec.} & f_o &= (500/60) \text{ cycles/sec.} \\ V_o &= 0.3 \text{ in.}^3 & P_o &= 80 \text{ psia} \\ S_H &= .45 \text{ in.} & A_H &= .11 \text{ in.}^2 \end{aligned}$$

Using these parameter values $I' \geq 6.8 \times 10^{-4} \text{ in.-lbs.-sec}^2$

The maximum outside radius of the flywheel is 1.3 inches in the reference design. The inside radius can be between 1.05 inches and 0.8 inches. The thickness can vary from .15 inches to .3 inches. The material used for the flywheel can vary in density from that of aluminum (.1 lb/in³) to that of tungsten (.7 lb/in³). From all the given ranges, the absolute range of flywheel inertia is:

$$1 \times 10^{-4} \text{ in. -lbs. -sec.}^2 \leq I' \leq 21 \times 10^{-4} \text{ in. -lbs. -sec.}^2$$

The minimum flywheel inertia falls within this range. It could be increased up to $21 \times 10^{-4} \text{ in. -lbs. -sec.}^2$ to adjust the natural frequency if this is desirable.

The flywheel, as a rotating body, will exert an opposing force when the direction of its axis of rotation is changed. Its behavior can be stated mathematically: The time rate of change of the angular momentum, $\overline{H}_O$, of any system of particles, with respect to a point 0 fixed in space, is equal to the moment of the external forces, $\overline{M}_O$, acting on the system about that point. Figure III-25 illustrates this principle.

Figure III-25. Gyroscopic Effect of Flywheel ($\overline{M}_O = \dot{\overline{H}}_O$)

$$\overline{M}_O = \dot{\overline{H}}_O = I' \omega_o \omega_r$$

In this equation I' is the inertia of the flywheel, ω_o is the mean angular frequency of flywheel rotation about its axis, and ω_r

is the angular frequency of rotation of the flywheel axis. It will be assumed that the maximum ω_r encountered is 2π radians/sec. However, the angular velocities encountered while aboard certain transportation vehicles (automobiles, trains, airplanes, etc.) may be higher. This question requires further investigation. The mean engine frequency is the same as that of the flywheel, $\omega_o = 52.3$ radians/sec. The possible range of flywheel inertia is 6.8×10^{-4} in.-lbs.-sec.² $\leq I' \leq 21 \times 10^{-4}$ in.-lbs.-sec.² Using this range, the range of moment of external forces is $.224 \text{ in.-lbs.} \leq M_o \leq .69$. Figure III-26 shows what this result means with respect to the implanted capsule. It is clear that the flywheel inertia, I' , should be kept as small as possible. The flywheel moment produces two equal forces F_1 and F_2 of a couple acting through moment arm r ($r = 4 \text{ in.}$).

Figure III-26. Flywheel Moment Applied to Implanted Capsule

$$M_o = F_1 r = F_2 r$$

The range of forces $F_1 = F_2 = F$ is $.055 \text{ lbs} \leq F \leq .17 \text{ lbs}$ which should not concern the recipient of an implantable power source.

L. MATERIALS

The implantable power supply must be constructed of suitable materials in order to function properly. Materials must be chosen for the following parts categories:

- Engine Parts
- Bearings
- Piston Rings and Liners
- Radioisotope fuel capsule
- TES Capsule
- Power package encapsulation.

The criteria for choosing the materials for each group will be discussed separately.

1. Engine Components

Materials used in the engine parts should have the following characteristics: high strength, low thermal conductivity, low emissivity, sufficiently high recommended maximum temperature, weldability, and low weight. It would be difficult to obtain one material which would be the best in all of these categories. However, 6-4 titanium, which is used in Thermo Electron's present steam engines, is an excellent material in many respects. It has a thermal conductivity which is less than half that of stainless steel. It is strong, weldable, and of sufficiently low weight. Its aging temperature is 900°F to 1000°F. The maximum temperature of the engine (boiler temperature) is 900°F. Consequently, the material can be expected to hold its properties even at the highest expected temperature. A summary of the properties of 6-4 titanium are given in Table III-19.

TABLE III-19
 PROPERTIES OF 6-4 TITANIUM ALLOY⁷

<u>Composition, %</u>	6 Aluminum, 4 Vanadium
<u>Physical Properties</u>	
Density, lb/cu. in.	0.160
Melting Point, °F	2800-3000
Thermal Conductivity (212°F) Btu/hr/sq. /ft. /°F/ft.	4.3
Coefficient of Thermal Exp. (68-1650°F), per °F	5.8×10^{-6}
Specific Heat (212°F) Btu/lb/°F	0.135
Electric Resistance (68°F) microhm-cm	176
Magnetic Permeability (20 oersteds)	1.00005
<u>Mechanical Properties</u>	
Modulus of Elasticity in Tension, psi	$15-17.5 \times 10^6$
Tensile Strength, 1000 psi	
Room Temp.	145-180
800°F	120-130
Yield Strength 1000 psi	
Room Temp.	135-175
800°F	95-105
Elongation (in 2 in.), %	
Room Temp.	5-10
800°F	7-9
Compr. Yield Strength, 1000 psi	
Room Temp.	170-190
800°F	100-120
Hardness (Rockwell)	C30-40
Impact Strength (Charpy), ft.-lb.	15-25 (bar)
Shear Strength, 1000 psi	100-110
<u>Thermal Treatment</u>	
Annealing Temp, °F	1275-1400
Stress Relieving Temp., °F	1000-1200
Solution Temp., °F	1700 ± 50

TABLE III-19 (continued)

<u>Thermal Treatment</u> (continued)	
Aging Temp. , °F	900 - 1000
<u>Fabricating Properties</u>	
Forging Temp. , °F	1450 - 1750
Working Temp. , °F	Room to 1200
Formability	good
Weldability	good
<u>Corrosion Resistance:</u> Superior resistance to nitric acid, moist chlorine, chlorine solutions, chlorinated organic compounds and inorganic chloride solutions. Excellent resistance to sea water and most chloride salt solutions - unchallenged by other structural metals. Resists impingement pitting attack.	
<u>Available Forms:</u> Billet, bar, plate, forgings, extrusions, sheet, strip, tube, wire	
<u>Uses:</u> Most widely used titanium alloy; for parts requiring high strength, formability, weldability.	

2. Bearings

Water-fed bearings will be used in the implantable power supply. The material used in this application should have the following characteristics: smoothness and hardness to reduce friction and prevent wear and retention of these properties at high temperature, capability of forming a surrounding water film for lubrication, and minimum life of five years. Graphite and carbides are candidates for this application. Outgassed tungsten carbide is potential material for this application because: (1) The material can be made very smooth. (2) Tungsten carbide is very hard and does not distort allowing the bearing surface to remain smooth even under load. (3) The area of bearing contact can be made small and will

remain so under load because of the hardness of the material. (4) Water will adhere to the surface of the material and form a thin-film promoting lubrication. (5) Outgassing the material keeps it smooth at high temperatures and enables water to adhere better to the surface, especially at high temperature. (6) Because of its properties, the material will give a long bearing life. (7) The material is easily obtained. A summary of the properties of a typical grade of tungsten carbide is given in Table III-20. The properties of various tungsten carbide compositions are given in Table III-21.

TABLE III-20

TYPICAL PROPERTIES OF TUNGSTEN CARBIDE

Melting Point (°C)	Room Temp. Hardness (kg/mm ²)	Softening Point (°C)	Grain Size (microns)	Friction Minimum (°C)	Mean Grain (microns)
2,870	2,400	815	0.8	830	1.0

3. Piston Rings and Liners

The piston rings and cylinder liners will be made of a graphite. The properties which make graphite an excellent material for this application are: low wear rate, ability to form a hydrodynamic film, low coefficient of friction, dimensional stability over a wide range of temperatures, low permeability, resistance to thermal shock, and easy fabrication by molding and machining. The rings will be mated with a chromium plated cylinder and the liners will be mated with steel pistons. A summary of the physical properties of the graphite which is being used for rings and liners in Thermo Electron's existing steam engines is given in Table III-22.

TABLE III-21
PROPERTIES OF VARIOUS TUNGSTEN CARBIDE COMPOSITIONS

Composition, percent	Specific Gravity	Hardness Rockwell A	Transverse Rupture Strength k psi	Young's Modulus psi x 10 ⁶
WC + 3 Co	15.25	92.7	170	97.5
WC + 4.5 Co	15.05	92.3	200	90.5
WC + 6 Co	14.85	90-92	225	88
WX + 9 Co	14.6	90	275	----
WC + 13 Co	14.15	89	300	80
WC + 10 TaC + 6 Co	14.7	91	220	----
WC + 12 TiC + 6 Co	11.2	92	160	----
WC + 10 TzC + 15 TiC + 8 Co	11.7	92	165	72
WC + 15 TaC + 15 TiC + 15 Co	11.4	90	190	67
WC + 2 TaC + 1 CbC + 20 Co	13.4	86	380	----
WC + 6 TaC + 11 TiC + 2 CbC + 9 Co	11.1	92	250	----

TABLE III-22

PHYSICAL PROPERTIES OF RING AND LINER GRAPHITE

Apparent Density (gm/cc)	Hardness Shore Scleroscope	Compress Strength psi	Transverse Strength psi	Tensile Strength psi
1.8	90	37,500	11,000	8,000
Modulus of Elasticity $\times 10^6$	Temp. Limit Neutron Atmosphere ($^{\circ}\text{F}$)	Coefficient of Thermal Expansion (in/in/ $^{\circ}\text{F} \times 10^{-6}$)		Permeability (Darcies $\times 10^{-6}$)
3.1	1200	2.2		0.3
Porosity (Vol. %)	Moldable to size	Dry Coefficient of Friction (Carbon on Steel)		
		at 70°F		at 500°F
2	fair	.04 - .45		.01 - .3

4. Radioisotope Fuel Capsule

The radioisotope fuel capsule is subject to extreme thermal, mechanical, and chemical forces. It must be strong enough to remain intact in case of an accident. If such an accident dumps the capsule into salt water, it must not corrode. A high maximum recommended temperature is also required. In this particular case, the capsule will be at 900°F maximum temperature. One material which has been used often in the past for fuel capsules is Haynes 25. Because of its properties, this material is a good choice for the implantable power source fuel capsule material. A listing of the properties of Haynes 25 is given in Table III-23.

5. Thermal Energy Storage Capsule

The thermal energy storage (TES) material will be lithium chloride/lithium fluoride eutectic. The container should not react chemically with the TES material. It should be strong and capable

TABLE III-23
PROPERTIES OF HAYNES 25⁷

<u>Composition, %</u>	.05-.15C, 1. -2. Mn, 19. -21. Cr, 9. -11. Ni, 14. -16. W, 3. Fe, 1. Si, bal Co
<u>Physical Properties</u>	
Density, lb/cu. in	0.33
Melting Temp., °F	2425-2570
Thermal Conductivity (1300°F)	
Btu/hr/sq. ft/°F/ft	13.1
Coefficient of Thermal Exp. 9.4 x 10 ⁻⁶	
(70-1800°F), per °F	
Specific Heat (77 - 1300°F),	
Btu/lb./°F	0.09 (80-212°F)
Electric Resistance, microhm	188.7
cm.	
<u>Mechanical Properties</u>	
Modulus of Elasticity in Tension,	
psi	
Room Temp.	34.2 x 10 ⁶
1200°F	27.4 x 10 ⁶
1400°F	26.3 x 10 ⁶
Tensile Strength, 1000 psi	
Room Temp.	146
1600°F	47
Yield Strength (.2% offset)	
1000 psi	
Room Temp.	67
1600°F	35
Elongation (in 2 in.) %	
Room Temp.	64
1600°F	30
Rupture Strength (1500°F), 1500 psi	
10 hr.	30
100 hr.	22
1000 hr.	18
<u>Corrosion Resistance</u>	
In sea water	good
In air	good
<u>Recommended Maximum Temp.</u>	930°F

TABLE III-23 (continued)

<u>Weldability</u>	good
<u>Available Forms</u>	plate, sheet, bar, wire, tubing
<u>Uses</u>	high temperature applications requiring strength and corrosion resistance.

of remaining at a sustained temperature of 900°F. Compatibility tests run at Thermo Electron indicate that 304 stainless steel is a suitable material for this application. A good choice for the eutectic capsule material is stainless steel. A summary of the important properties of 304 stainless steel are given in Table III-24.

TABLE III-24

PROPERTIES OF 304 STAINLESS STEEL⁷

<u>Composition, %</u>	.08 max C. 2. max Mn. 1. max Si, .045 max P, .030 max S, 18. -20. Cr, 8. -12. Ni
<u>Physical Properties</u>	
Density, lb/cu in.	0.29
Melting Point Range, °F	2550 - 2650
Thermal Conductivity (212°F)	
Btu/hr/sq. ft./°F/ft	9.4
Coefficient of Thermal Exp. (32-212°F), per °F	9.6×10^{-6}
Specific Heat (32-212°F), Btu/lb/°F	0.12
Electric Resistance (Room Temp.), microhm-cm	72
<u>Mechanical Properties</u>	
Modulus of Elasticity in Tension, psi	28×10^6
Tensile Strength, 1000 psi	
Annealed	85
Cold Worked	110

TABLE III-24 (Continued)

<u>Mechanical Properties (Continued)</u>	
Yield Strength, 1000 psi	
Annealed	35
Cold Worked	75
Elongation (in 2 in.), %	
Annealed	60
Cold Worked	60
Red. of Area % (Annealed)	70
Hardness	
Annealed	149
Cold Worked	240
Impact Strength (Izod), ft-lb	
Annealed	110
Cold Worked	90
Endurance Limit, 1000 psi	
(Annealed)	34
Creep Strength (1% ext in	
1000 hr), psi	
1000°F	17,000
1300°F	4,000
1500°F	1,200
<u>Fabricating Properties</u>	
Annealing Temp., °F	1850-2050
Forging Temp. (start), °F	2100-2300
Machinability Index	50
Weldability	Excellent
<u>Corrosion Resistance</u>	Excellent atmosphere resistance
<u>Available Forms</u>	Sheet, strip, plate, bar, wire, forgings, tubing
<u>Uses</u>	General purpose & welded construction

In spite of the hydrogen diffusion problem, the high gravimetric thermal storage capacity of lithium hydride make it an interesting TES material. Tungsten is the choice for the lithium hydride capsule material because of its low permeability to hydrogen. Tungsten's important properties are listed in Table III-25

TABLE III-25
SUMMARY OF TUNGSTEN PROPERTIES⁷

<u>Physical Properties</u>	
Density, lb/cu.in	0.70
Melting Point, °F	6170
Thermal Conductivity (212°F)	
Btu/hr/sq.ft./°F/ft	96.6
Coefficient of Thermal Exp.	
(70°F), per °F	2.5×10^{-6}
Specific Heat (70°F), Btu/lb/°F	0.034
Electric Resistance (70°F)	
microhm-cm	5.48
<u>Mechanical Properties</u>	
Modulus of Elasticity in Tension, psi	
Room Temp.	59×10^6
1600°F	55×10^6
Tensile Strength, 1000 psi	
Room Temp	220
1600°F	66
2400°F	45
Hardness (VHN)	
Cold Worked	480
Stress Relieved	450
Recrystallized	350
<u>Fabricating Properties</u>	
Working Temp., °F	700 - 3000
Annealing Temp., °F	
Stress Relief	≈1700
Recrystallization	≈2900
Machinability	Difficult but can be done with carbide tools
Joining	Weldable by TIG or resistance; can be brazed
<u>Corrosion Resistance</u>	Oxidizes rapidly above 1300°F; resistant to HF and HCl; attacked hot HN O_3 ; resistant to hot metals
<u>Available Forms</u>	Sheet, rod, wire, powder, plate, foil, tube

TABLE III-25 (Continued)

<u>Uses</u>	Lamp filaments, electrical contacts, electronic tubes, X-ray tubes, reactors, high temperature structures
<u>Permeability Constant (permeability to hydrogen), $\frac{\text{cm}^3 \text{ (STP) hr}^{-1} \text{cm}^{-1} \text{atm}^{-1/2}}$</u>	1.8×10^{-5}

6. Power Package Encapsulation

The power package encapsulation will consist of an inner metal can, a thin layer of Fiberfrax thermal insulation, a layer of General Electric RTV-120 and an outer layer of Dow Corning medical grade 382 Silastic. The can should be strong without being heavy. Since only the encapsulation layers will separate it from body juices, the container material should be resistant to corrosion in salt water and non-toxic to the body. Titanium is a good material for this application. It has a high strength to weight ratio and is resistant to corrosion by salt water. It has also been used as a prosthetic material. Properties of 6-4 titanium alloy are given in Table III-19. The two layers of GE RTV-120 and Dow Corning Silastic will be the same as those used successfully to coat TECO's heart assist device, implantable heat exchanger, and blood irradiator. The properties of the RTV and Silastic coatings are given in Tables III-26 and III-27, respectively.

TABLE III-26
GENERAL ELECTRIC RTV-120 PROPERTIES

<u>Cured Properties</u>	
Specific Gravity	1.07
Hardness, Shore A Durometer	30
Tensile Strength, psi	350
Elongation, %	400
Tear Resistance, die B, lb/in.	.45
Brittle Point, °F	below -90°F
Linear Shrinkage, %	1.0
<u>Thermal Properties</u>	
Thermal Conductivity (200°F), Btu/hr-ft ² - °F	0.12
Coefficient of Thermal Exp. (0-350°F), in/in-°F	<15 x 10 ⁻⁵
<u>Electrical Properties</u>	
Dielectric Strength, volts/mil 0.075" thick	500
Dielectric Constant (60 Hz)	2.8
Dissipation Factor (60 Hz)	.0026
Volume Resistivity, Ohm-cm	3 x 10 ¹⁵
<u>Uses</u>	sealing and extreme temperature applications

TABLE III-27
 PROPERTIES OF DOW CORNING MEDICAL
 GRADE 382 SILASTIC

<u>Cured Properties</u> (after vulcanizing for 24 hours at 77°F)*	
Hardness, Shore A Durometer	43
Tensile Strength, psi	400
Elongation, %	160
Brittle Point, °F	below -100°F
Linear Shrinkage, %	
after 3 da. at 77°F	0.4
after 6 da. at 77°F	0.6
after 14 da. at 77°F	0.7
Water Absorption, %, after	
7 days immersion at 77°F	0.4
Thermal Conductivity,	
calories/cm-°C-sec.	0.525×10^{-3}
Volume Coefficient of Thermal	
Exp. (0-100°C). per °C	7.50×10^{-4}
*Test specimens 0.075 inch thick, tested at room temperature.	
<u>Cured Properties</u> (catalyzed with 0.5% Catalyst M)	
Hardness, Shore A Durometer	
(minimum) 1 hr. cure	25
24 hr. cure	35 to 55
Tensile Strength, psi	
(minimum) 24 hr. cure	300
Elongation, %	
(minimum) 24 hr. cure	100

M RELIABILITY

Historically, reciprocating steam engines have proven to be the most reliable of power producers. The Ford Museum at Dearborn, Michigan displays a steam engine that was in commercial use for 87 years. Some of Newcomen's steam engines built in the early 18th century are still operable.

Reliability is difficult to quantitatively discuss at the conceptual design stage. Clearly, the power source is complex and there is little room for redundancy. Therefore, individual component failure rates must be extremely low.

Since the most critical components are in the engine power train, Thermo Electron's test experience of kindred engines can give some insight into the problem. The run times for present engines are limited by the life of the seal between the steam and lubricant. However, the reference design described in Chapter II uses the working fluid as the lubricant so that seal life, per se, will no longer be a problem.

The 300 watt steam engine power supply built for the United States Army was designed for a package life of 1000 hours. The two tests on Pistons and exhaust valves were terminated after 600 hours due to valve leakage. Since this test, the exhaust valves have been redesigned to give increased life. The steam piston did not have rings and worked within a graphite liner. Two intake valves were tested for 1000 hours without any significant signs of wear. Over 3000 hours have been logged on the crankcase without any sign of component wear. All of these tests were run at 3600 RPM. The reference design steam engine for the implantable power source operates at a mean speed on only 600 RPM. Making the conservative assumption that an engine operating at 3600 RPM

should run 6 times as long at 600 RPM, the state-of-the-art of steam engines closely related to that of the reference design is given in Table III- 28.

TABLE III-28
TEST LIVES OF THERMO ELECTRON STEAM
ENGINE COMPONENTS
(300 Watt Package)

Component	Actual Life at 3600 RPM (hours)	Normalized Life at 600 RPM (hours)
Pistons	600	3600
Exhaust Valves	600	3600
Intake Valves	1000	6000
Crankcase	3000	18,000

The test results in Table III-28 are particularly significant because the 300 watt engine efficiency was measured at 17.6 per cent. Its bore is roughly double that of the reference design, so that no large scale factors need to be considered. In addition, the valve mechanism, component materials and operating temperature of the 300 watt test engine are essentially the same as that of the reference design.

Reliability is not so much in the number of parts as in running them conservatively. Conservative operation means low temperature, light loadings and wet lubrication of all moving parts. Assuming that working fluid lubrication will replace oil lubrication with no penalty in operating life, one sees that state-of-the-art steam engines are, in actual hardware, quite reliable. Therefore, a design life of 40,000 hours does not seem insurmountable.

N. FUEL EVALUATION

Ideally, the radioisotope fuel should have 1) a long half-life; 2) a high power density; 3) a stable, high temperature fuel form; 4) compatible encapsulation material; 5) high availability; 6) low cost and 7) require little shielding. Neither of the candidate fuels (Pu-238, Pm-147, and Tm-171) have all these specifications, although Pu-238 clearly has superior half-life and power density.

Comparison of the cost and availability of radioisotope fuels is a difficult task beyond the scope of the present study. However, a few comments can be made. First is that the comparison of cost and availability should be projected at least into the 1975-1980 period. Second is that the radioisotope demand for fueling circulatory support devices is potentially so large that special single purpose facilities devoted exclusively to the economic production of a single radioisotope should be considered. After the initiation of routine implantations of circulatory support systems, the annual sales volume of radioisotope fuel could well exceed a billion dollars.

The candidate radioisotope fuels have been evaluated with respect to cost, radiation, and weight. The dose rates were calculated as a function of distance, shield material, shield thickness and impurity level for the isotopes plutonium-238 (pu-238), promethium-147 (Pm-147) and thulium-171 (Tm-171). A comparison of the radiation and weight of a 25-watt heat source is shown in Table III-29. Each source configuration was assumed to be a right circular cylinder with a length-to-diameter ratio of one.

The latter two radioisotopes, in principle pose much lesser shielding problems than plutonium, but in practice the available

TABLE III-29
RADIATION AND WEIGHT COMPARISON OF 25 THERMAL WATT
RADIOISOTOPE HEAT SOURCES*

Fuel	Impurity	Assumed Implant Life (yrs)	Power (Beginning of Life)+ (watts)	Uranium Shield Thickness (cm)	Midplane Dose Rate at 5 cm (mrem/hr)	Heat Source Weight (lb)
Pu-238	0	5	26	0.1	72**	0.36
				0.5	56**	0.95
				1.0	43**	2.26
Pm-147	0	2.67	50	0.1	470	0.54
				0.5	0.4	1.67
				1.0	1×10^{-5}	3.85
	Pm-146 (0.25×10^{-6})	2.67	50	0.1	2000	0.54
				0.5	700	1.67
				1.0	240	3.85
	0	1.9	50	0.1	0.7	0.79
				0.5	6×10^{-10}	2.10
				1.0	1×10^{-18}	4.55
Tm-171	Tm-170 (1×10^{-6})	1.9	50	0.1	7.4	0.79
				0.5	1.9	2.10
				1.0	0.32	4.55
	Tm-170	1.9	50	0.1	74	0.79
				0.5	19	2.10

TABLE III-29 Continued

RADIATION AND WEIGHT COMPARISON OF 25 THERMAL WATT
RADIOISOTOPE HEAT SOURCES*

Fuel	Impurity	Assumed Implant Life (yrs)	Power (Beginning of Life) [†] (watts)	Uranium Shield Thickness (cm)	Midplane Dose Rate at 5 cm (mrem/hr)	Heat Source Weight (lb)
	(1×10^{-5})			1.0	3.2	4.55
	Tm-170	1.9	50	0.1	740	0.79
				0.5	190	2.10
	(1×10^{-4})			1.0	32	4.55

* Rated at the end of design life

** Neutron dose rate assumed unattenuated by uranium

[†] The isotopes with shorter half-lives have higher initial power levels to ensure adequate power at the end of life.

forms of each contain impurities (Pm-146, Pm-148, and Tm-170) which contribute large amounts of penetrating radiation, and greatly increase the shielding requirements. Separation of the deleterious impurities is not likely to be possible at an acceptable cost, since isotopes of an element are chemically identical, and differ only physically. The latter isotopes also give heavier packages when shielded, even if pure, because of lower power densities and because more of the radioisotope is needed to compensate for their short half-lives compared to plutonium.

The neutron and gamma dose rates from a 26 watt Pu-238 heat source is given as function of distance from the curved surface of the source in Figures III-27 through III-29. Uranium thickness of 10, 5 and 1 millimeters shield the source in Figures III-27, III-28 and III-29, respectively. The pronounced buildups of gamma radiation after five years from the daughters of impurity Pu-236 (assuming 1 PPM) is shown on each of these figures. Figure III-29 also shows Mound Laboratory measurements of the dose rates from the 25 watt Pu 238 source currently used in the NHI study of additional endogenous heat.

These dose rates from plutonium-238 are large compared to those to which the healthy population is normally exposed. However, they would have to be continued for years to accumulate as much absorbed radiation as that used in short-term therapeutic radiation treatment. In a test of acute local radiation effects a radioisotope-heated blood-cooled prosthesis has been implanted successfully in the thoracic aorta of a dog, where it subjects the blood and surrounding tissue to a dose rate an order of magnitude higher than those tabulated for plutonium. At the time of writing after six months of exposure the dog is healthy and neither heat nor radiation damage is detectable in its blood. The operation

Figure III-27. Calculated Dose Rates from 26 Watt Pu-238 Source
Shielded with 10 Millimeters of Uranium

Figure III-28. Calculated Dose Rates from 26 Watt Pu-238 Source
Shielded with 5 Millimeters of Uranium

Figure III-29. Calculated Dose Rates from 26 Watt Pu-238 Source Shielded with 1 Millimeter of Uranium. (25 watt "Mound" Curves are measurements from capsule used in the NHI study of additional endogenous heat.)

of a life support device such as those considered here is not accomplished without risks to individual users, but the device is designed to increase the life expectancy of a group of users.

A cost comparison of these three radioisotopes is given in Table III-30. This comparison assumes that the residual fuel will be reprocessed after one implantation period, which is taken as 5 years for Pu-238, but only as a half-life for Pm-147 and Tm-171. In the third column, note that the total energy per initial thermal watt is far greater for Pu-238 than either Pm-147 or Tm-171. Pu-238 also has the advantage of a higher energy utilization factor, which is the ratio of useful energy to total energy released during the period of implantation. The fifth column is the product of the third and fourth columns and gives the effective energy per initial thermal watt. This number is then multiplied by the cost per thermal watt to yield the figure of merit, dollars per watt year of useful life, in the last column. It is seen that, on this basis, Pu-238 is the least expensive in addition to being the lightest in weight. If economic impurity levels are considered, it is probably the most desirable from the standpoint of radiation. In addition, there are clear indications that present gamma radiation levels can be further reduced. When this same comparison is repeated on the basis of a 5-year life for all three fuels, Pu-238 is the undisputed choice.

Tritium deserves serious consideration as a candidate fuel. It has a desirable half-life (12.3 years) and the problem of shielding the low energy bremsstrahlung from the low energy beta decay is trivial. Tritium could not be contained as a gas. However, as lithium tritide, it would have a power density of 0.11 watt per cubic centimeter. Although low, it is still useable.

TABLE III-30
COST COMPARISON OF CANDIDATE FUELS

Fuel	Implantation Period (yrs)	Total Energy Per Initial Thermal Watt (watt-yrs)	Energy Utilization Factor	Effective Energy Per Initial Thermal Watt (watt-yrs)	Cost \$/ (watt)	Cost \$/ (watt-yr)
Pu-238	5	128.5	0.934	120	1000	8.3
					500	4.2
Pm-147	2.67	3.86	0.53	2.05	200	98
					50	24
Tm-171	1.9	2.74	0.53	1.45	4000	2800
					200	140

The critical problem with using tritium fuel is the high helium release rate. Helium containment is difficult with Pu-238 where one helium atom represents approximately 5 meV. For tritium each helium atom would represent only 6 keV. Thus the helium buildup problem is three orders of magnitude more severe. The only possibility of using tritium as a fuel would be to leak the helium gas while absolutely containing the tritium. Quartz might be a suitable window material for the helium. This type of problem has a high probability for being solved. The helium could probably be released directly from the power source into the body since the leak rate would be so small.

Another problem is the stability of the fuel form. Lithium hydride has an appreciable vapor pressure of hydrogen at its melting point. The containment of this hydrogen is quite difficult. The same difficulty might be expected with tritium when compounded with lithium. Although the containment would probably not be difficult from a heat source viewpoint, it could be impossible from a radiological safety viewpoint. Studies of radiation stability at high temperature would be highly desirable.

The availability of tritium would probably be more straightforward than that of Pu-238. The cost is not clear. Approximately 200 neutrons would have to be invested in tritium to supply the same energy as one neutron invested in Pu-238. This is a difficult cost factor to overcome. Tritium will probably not become of major interest unless the radiation from Pu-238 spontaneous fission neutrons proves unacceptable. Although this is not expected to be the case, the merits of tritium as a fuel for implantable power sources should be carefully considered.

O. WEIGHT AND VOLUME VARIATION OF POWER SOURCE

The components weights of the Rankine cycle, implantable power source described in Chapter II are tabulated in Table III-31. It is apparent that the heat source subassembly constitutes over half of the power source weight. The shield and thermal energy storage account for a large fraction of the heat source subassembly weights. Therefore, the weight and volume of the power source are sensitive to the dose rate and load profile constraints assumed in the design. The reference design reflects a moderate amount of shielding and ample thermal storage. The shield should keep the gamma dose rate less than the fission neutron dose rate for almost 5 years (assuming an initial level of one PPM Pu-236).

The weight and volume variations of the implantable power source as a function of mean hydraulic output are given in Table III-32. These characteristics were scaled from the reference design described in Chapter II which used Pu-238 in the oxide (depleted) form as the fuel. The maximum dose rates on the power source surface are also listed since they are pertinent to the weight and volume trade-offs. Although the tabulated values are design dependent and not highly accurate, the relative weights and volumes should be significant.

Plutonium-238 gives the highest weight power source. The power source weight for Pm-147 and Tm-171 heat sources may be estimated by referring to Table III-29 and adjusting for the differences in heat source weights. Of course, the impurity level and dose rate must also be specified.

The estimated minimum weights and volumes of circulatory support system components are given in Table III-33. These estimates are based on the reference design of a Rankine cycle implantable power source (fueled with Plutonium-238 oxide)

TABLE III-31

COMPONENT WEIGHTS OF IMPLANTABLE POWER SOURCE^{*}
(Pounds)

<u>Heat Source Subassembly</u>	
Plutonium-238 oxide fuel	.12
Tantalum shield	.56
Fuel encapsulation	.35
LiF/LiCl eutectic	.57
TES encapsulation	.66
Vacuum foil insulation	.11
Subtotal	2.37
<u>Rankine Cycle Engine Subassembly</u>	
Cylinder, piston and support structure	.15
Pumps (Hydraulic, feedwater and sump)	.22
Flywheel	.12
Boiler	.11
Subtotal	.60
<u>Hydraulic Subassembly</u>	
Pressurizer cylinders	.10
Pressurizer pistons	.03
Pressurizer springs	.20
Subtotal	.33
<u>Miscellaneous</u>	
Canister	.58
Encapsulation	.51
Plumbing, etc.	.10
Subtotal	1.19
TOTAL	4.49

* Reference design implantable power source described in Chapter II.

TABLE III-32

WEIGHT AND VOLUME VARIATION OF
IMPLANTABLE POWER SOURCE*

Mean Hydraulic Power (watts)	Power Source Volume (cc)	Power Source Weight (pounds)	Surface Dose Rate (mrem/hr)
2	860	3.52	26
3	930	4.02	35
4 ⁺	1000	4.49	44
5	1070	5.01	53
6	1050	5.57	64

* Fuel is Plutonium-238 oxide.

+ Reference design

TABLE III-33

MINIMUM WEIGHTS AND VOLUMES OF
THERMO ELECTRON
CIRCULATORY SUPPORT SYSTEM COMPONENTS

Implantable Power Source ¹	Volume (cc)	Mass (kg)	Max. Surface ² Dose Rate (mrem/hr)
Reference Design ³			
(1.2 mm Ta shield)	950	1.92	60
(6 mm Ta shield)	1000	2.04	44
(12 mm Ta shield)	1060	2.24	28
Compact packaging ⁴			
(1.2 mm Ta shield)	800	1.79	110
(6 mm Ta shield)	850	1.91	80
(12 mm Ta shield)	910	2.11	50
Projected Minimum ⁵			
(1.2 mm Ta shield)	600	1.45	128
(6 mm Ta shield)	650	1.57	88
(12 mm Ta shield)	710	1.77	56
<u>Blood Pump Controls⁶</u>			
Present design	150	.13	
Projected minimum	120	.11	

TABLE III-33 (Continued)

MINIMUM WEIGHTS AND VOLUMES OF
THERMO ELECTRON
CIRCULATORY SUPPORT SYSTEM COMPONENTS

	<u>Volume (cc)</u>	<u>Mass (kg)</u>	<u>Max. Surface² Dose Rate (mrem/hr)</u>
<u>CHMC-TECO Assist Pump⁷</u>			
Present hardware	300	. 12	
Projected minimum	280	. 12	
<u>Total Circulatory Support System⁸</u>			
Reference design power source	1450	2. 30	
Compact packaging	1300	2. 17	
Projected minimum	1050	1. 82	

1. Rankine cycle engine fueled with Plutonium-238 oxide (includes shielding, void volume, hydraulic pump, hydraulic accumulator and power source encapsulation).
2. Sum of neutron and gamma dose rates at beginning of life (1 PPM Pu-236 impurity).
3. Reference design described in Chapter II.
4. Reduced TES and thermal insulation.
5. Improved efficiency, reduced TES, more compact engine and optimized structure.
6. Counterpulse controls.
7. Hardware device undergoing in vivo testing.
8. Total of power source, blood pump and blood pump controls.

described in Chapter II as well as the CHMC-TECO left-ventricle-to-aorta assist pump. The assist pump numbers are accurate since they are based on in vivo hardware.

The blood pump control unit contains a volume sensor, trigger mechanism, hydraulic valve and pressure transducer. The pressure transducer is bulky since it must expel the stroke volume of the blood pump. This unit could be integrated into either the blood pump or the power source, or placed along the hydraulic lines connecting them. There may be placement advantages to the latter alternative with the control unit located right below the diaphragm.

A primary consideration in reducing the power source volume is the radiation dose rate to the immediately adjacent tissue. Limited radiation attenuation by shielding can be achieved without the heat source becoming prohibitively heavy. However, the geometric attenuation close to the heat source is rapid. Therefore it seems reasonable to trade a modest increase in power source volume and weight for a significant reduction in surface dose rate. It may well be that the power source volume will not be dictated by minimum component size, but rather by the power source shape that will minimize surface dose rate and still be implantable.

Based on Thermic Electron's experience within vivo heat exchangers and blood pumps, implanting a one liter power source in the abdomen does not seem unreasonable. However the problems of diaphragm penetration, power source volume, shape, and suspension should be evaluated in the next development phase by in vivo testing of dummy components.

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CHAPTER IV.

BLOOD HEAT EXCHANGER

The blood heat exchanger concept is simply to use the blood pump for this purpose. The rejected heat from the steam engine will be transferred to the blood in the following sequence: First, the reject heat from the steam condensing will be transferred into the hydraulic accumulator or reservoir. This heat is carried to a heat exchanger located in the control unit, where it is transferred to the trapped coupling fluid. The sloshing due to the pump bladder readily mixes the coupling fluid, which transfers the heat across the pump bladder with its autologous interface into the blood. Reference to Figure II-15 may be helpful in tracing the reject heat path.

The reject heat from the steam engine is exchanged with the blood, since this appears to be the only feasible method of rejecting waste heat from the power source and conversion mechanism. The blood in turn rejects heat to the outside of the body by means of the usual physiological mechanisms of perspiration, radiation, and convection-conduction. Since blood must pass through the heat exchanger, either the heat exchanger should be located in an area of high blood flow (an artery, vein, or the blood pump), or prosthetic tubes must be relied upon to bring blood to the condenser.

The most natural way to reject the heat in or adjacent to the blood pump or pumps of a circulatory assist device is to use the artificial ventricle surface area as the heat exchanger surface. This has the advantage of requiring no additional prostheses, or additional prosthetic surface in contact with the blood. It also simplifies the surgical implantation (anastomosis)

Before calculating the temperature drop through the blood pump bladder, it is useful to review the major types of left ventricular assist devices. The assist pump being developed jointly by Thermo Electron (TECO) and Children's Hospital Medical Center (CHMC) will be described in some detail.

A. TYPES OF LEFT-VENTRICULAR ASSIST DEVICES

Three major current concepts of mechanical assistance for a failing left ventricle are illustrated in Figure IV-1. This composite schematic drawing illustrates, from top to bottom, the in-series, in-parallel, and left-ventricle-to-aorta types. The principal characteristics of each type are listed in the following paragraphs.

1. In-Series Type

This valveless assist pump is designed to accept blood from the ascending aorta and to eject it into the descending aorta, relying upon small differences in orifice size to determine direction of flow, as well as upon the directional flow arising in the left ventricle. Further, the device requires occlusion (or transection) of the ascending aorta just distal to the pump's proximal orifice. Finally, it requires the presence of a competent aortic valve in order to prevent ventricular volume overload during ventricular diastole. This pump was developed by Avco and the Kantrowitz brothers.

2. In-Parallel Type

This device, developed by the Statham and DeBakey groups, has two valves: one at the inflow and one at the outflow of the pump chamber. Unlike the in-series type, which is totally implantable within the chest, the in-parallel type is designed to be implanted with only its connecting

TYPES OF LEFT VENTRICULAR ASSIST DEVICES

Figure IV-1. Model IV Assist Device Cross-Sectional Views

arms within the chest, and the pump sitting outside the chest wall. Similarly, unlike the in-series type, the surgery required for implantation is relatively simple, compared with the considerable surgical procedure involved in implanting the in-series type. In contrast to the in-series type, the pump is drawing from a low-pressure reservoir (the left atrium) and must compete for filling with the relatively compliant left ventricle and the mitral valve orifice, which offers no significant resistance to flow. Thus considerable negative-pressure (-30 to -60 mm Hg) is required for filling the pump from the left atrium, and, because of slow filling, it can only be activated effectively every 2nd, 3rd, or 4th cardiac cycle. Like the in-series device, it requires a competent aortic valve in order not to defeat its own purpose in relieving the left ventricle of pressure-volume work. It is of note that this device, like either of the other two, requires the capture of a major part of blood flow in order to reduce left ventricular pressures significantly: better than 85 - 90%. This pump, in its present form, does not require an implantable power supply and is discussed here only to the extent that, should an implantable version of it be developed, it could be powered by the proposed system.

3. Left-Ventricle-to-Aorta Type

This pump (TECO-CHMC), like the in-parallel type, lies in a parallel position to the flow of blood through the ventricle to the aorta. However, since it is capable of accepting blood from the left ventricle during both low- and high-pressure portions of the cardiac cycle, it operates in series, rather than in parallel, since the aortic valve never opens. Like the in-parallel type, there are two valves, at the inflow and outflow ends of the pumping chamber. Like the in-series

type, since it fills primarily during what would otherwise be a high-pressure portion of the cardiac cycle (left ventricular contraction), it fills readily and may be activated with every cardiac cycle. Unlike either of the other two types, this device does not require a competent aortic valve, since volume overload in the left ventricle (aortic regurgitation) is easily accepted by the pump. Decreased left ventricular pressure results in this pump to the extent that it is able to accept the full stroke volume of the left ventricle, an eventuality which is not characteristic of either of the other two types of device.

B. CHARACTERISTICS

All three types of device tend to maintain aortic mean pressure constant, while increasing the pulse pressure widths during pump activation (counter-pulsation). Pulmonary arterial, right ventricular and right atrial pressures do not change significantly; neither does left atrial pressure unless it is elevated. In the latter eventuality, it is somewhat reduced by activation of the pump, in decreasing order of effectiveness, by the left-ventricle-to-aorta, in-parallel, and in-series types. Total cardiac output does not change significantly during activation of the pumps, though there is strong suggestion that a total increase in flow occurs during pump activation, particularly with the left-ventricle-to-aorta device and also the in-parallel device.

All three devices share the common feature of being powered by a fluid under pressure. The application of driving pressure must take place in synchronization with the heartbeat and release at the end of pump systole. This basic common feature allows the power supply design evolved around one pump to be used, with modifications, with all other pumps.

The TECO-CHMC left ventricle-to-aorta assist device has performed in a promising manner in regard to:

Lowering left ventricular pressure (60 - 90%)

Blood flow (up to 6 liters/min)

Blood compatibility

Ease of surgical implantation and removal.

The CHMC-TECO assist device is a two-valved pump which removes blood from the left ventricle at low pressures and ejects blood into the descending aorta at systemic pressure. The inflow tube is introduced into the left ventricle through the apex of the heart. Cross-sectional views of this assist device are given in Figure IV-2. A photograph of the pump components is shown in Figure IV-3. The assembled device is pictured in Figure IV-4. The CHMC-TECO assist device characteristics are given in Table IV-1. After visual inspection of the components and the assembled device, it is tested with a helium mass spectrometer and with 20 psig water pressure for leaks. The pumping performance of the device is tested on a mock circulation loop. The assembled device is sterilized by autoclaving.

C. AUTOLOGOUS LINING ON FLOCKED-DACRON SUBSTRATES

Experiments for developing a blood-compatible pseudo-endothelium surface (i.e., autologous lining) inside the CHMC-TECO assist device have been encouraging. Uniform flocked surfaces with a relatively tangled mat of Dacron fibers have been formed on all parts of the assist device, excluding the disc valves. Deposited fibrin is organized by trapped blood macrophages followed by further differentiation of the surface layers of cells into pseudo-endothelium.

Figure IV-2. Cross-Sectional Views of CHMC-TECO Assist Device Components

Figure IV-3. Photograph of CHMC-TECO Assist Device Components

Figure IV-4. Picture of Assembled TECO-CHMC Blood Pump

TABLE IV-1

CHMC-TECO ASSIST DEVICE CHARACTERISTICS

(Model IV)

SIZE:	External Volume -----	300 cc
	Internal Volume -----	114 cc
	Total Chamber Volume -----	86 cc
	Maximum Stroke Volume -----	60 cc
	Overall Length -----	18.5 cm
	Maximum Housing Width -----	6 cm
	Axial Flow Length -----	23.4 cm
WEIGHT:	Dry -----	222 gm
	Wet -----	336 gm

Present experience indicates that this interface is blood-compatible in regard to clotting, hemolysis and denaturing protein.

It appears that anticoagulation is desirable (for at least 60 days) after pump implantation to prevent a thick fibrin buildup on the flocked-Dacron matrix. The advantages of thin fibrin buildup are better adhesion, greater compliance and lower stresses. The thin fibrin deposit also decreases the temperature drop when the pump bladder is also used as a heat exchanger.

D. TEMPERATURE DROP THROUGH BLOOD PUMP BLADDER

The temperature drop across the blood pump bladder is given by

$$\Delta T = \frac{qt}{kS}$$

where,

ΔT = temperature drop across wall

q = heat flux through wall

t = wall thickness

k = wall thermal conductivity

S = total surface area

The thickness of the ventricle depends on the stress level allowed in the bladder. The stress level can be determined from Figure IV-5 where

r = local ventricle end radius

R = flat-section radius

t = uniform silastic thickness

p = internal pressure

σ = stress in silastic

Let us perform a force balance along the periphery of R:

$$2 \sigma t (2\pi R) = 2 r p (2\pi R)$$

Then:

$$\sigma = \frac{rp}{t}$$

$$\max r \approx 1.0''$$

$$\max p \approx 200 \text{ mm Hg or about } 4 \text{ psi}$$

If

$$t = 0.020'', \sigma_{\max} = 200 \text{ psi}$$

$$t = 0.040'', \sigma_{\max} = 100 \text{ psi}$$

$$t = 0.800'', \sigma_{\max} = 50 \text{ psi}$$

Suitable polyurethane rubbers (DuPont) have tensile strengths of about 1700 psi and tear strengths of about 1000 psi. The 0.030" -thick polyurethane bladders currently used have more than adequate strength.

The thermal conductivity of the various grades of polyurethane varies somewhat, but a conservative (low) value is:

$$k :: 5 \times 10^{-4} \frac{\text{cal}}{\text{cm sec}^\circ\text{C}}$$

$$3 \times 10^{-3} \frac{\text{watts}}{\text{in.}^\circ\text{F}}$$

For $r \ll R$ in Figure IV-5, the ventricular area is approximately

$$\begin{aligned} S &= 2 (\pi R^2) + (2\pi R) (\pi r) \\ &= 2 \pi R^2 + 2 \pi^2 R r \end{aligned}$$

For the compressed ventricle, with a stroke volume of 4.88 in.³ (giving 6000 liters/min cardiac output at 75 bpm) and a compressed

Figure IV-5. Ventricle Model Showing Important Parameters

volume of 2.5 in.³:

$$\text{Compressed Volume} = 2\pi R^2 + \pi^2 R r^2$$

Let

$$r = 0.15 \text{ inch}$$

Then

$$R = 1.55 \text{ inches}$$

and

$$\begin{aligned} S &= 15.1 + 4.6 \\ &= 19.7 \text{ in.}^2 \text{ (plus valves and connections to valves)} \end{aligned}$$

If a mean of 25 watts is dissipated, then

$$\Delta T = \frac{25 (0.030)^\circ \text{F}}{(3 \times 10^{-3}) (19.7)}, \quad t \text{ in inches, } = 12.7^\circ \text{F}$$

The accuracy of this calculation has been confirmed at Thermo Electron by an experiment using a hydraulically driven blood pump. This temperature drop across the bladder is certainly acceptable from an engine efficiency standpoint, and since this fluid does not come in contact with tissue, there is no incompatibility from a biologic standpoint.

There will be an additional temperature drop across the autologous lining on the bladder. However, this temperature drop should be small because the higher thermal conductivity of tissue is roughly double that of the bladder material, and the tissue thickness should be less than one-third that of the bladder. Typically, the added temperature drop across the bladder due to the pseudo-endothelium would be approximately 2°F.

This problem has been treated in greater detail in Thermo Electron Report No. TE 66-67, "Six Month Technical Progress Report, Study of the Effects of Additional Endogenous Heat." January 1967.

The average forced-convection heat-transfer coefficient and boundary-layer temperature drop were calculated based on certain mean velocity and flow assumptions for the ventricle. With the assumptions made, a maximum temperature drop of 4.85°F during diastole was calculated.

An analysis of natural convection alone was made, and yielded temperature drops of 11°F and 23°F , which are quite high. Therefore, natural convection alone is not adequate, fortunately, forced-convection conditions will exist for virtually all of the cycle or beat.

It is of interest to determine what temperature changes occur at the end of diastole and the end of systole. Over these parts of the cycle, mean blood flow and velocity are lower than at other parts of the cycle. The transient production analysis is encouraging because it indicates that during these short intervals (each probably less than $1/10$ of the cycle) the local blood temperature at the ventricle surface can be kept at acceptable levels by proper control of the variables.

These analyses show that heat transfer across the blood pump bladder is feasible with respect to engine efficiency. The heat-transfer fluid will be insulated from the surrounding tissue by the encapsulating material. The maximum temperature drop across the blood pump bladder will be larger during periods of peak activity. However, the duration of these spikes will be short, and the thermal capacity of the system will act as a filter to reduce their amplitude.

Dr. Rawson of the John B. Pierce Laboratory has reported (NHI Review, Dec. 1967) that tissue in contact with an implanted heated surface has been able to tolerate 1° to 2°C temperature rise for indefinite time periods. In the light of this evidence, temperature increases of 1/4 to 1/6 that amount are expected to be tolerated well.

CHAPTER V

CONTROLS

Attempts to couple the power source and blood pump subsystems which were independently designed demonstrated the necessity of a "systems" approach to the selection of the optimum controls for circulatory support systems. The system control is a function of the engine and blood pump controls and their interaction. The physiological requirements are discussed in Appendix A.

The subject is introduced with a description of the external control system currently used to pneumatically drive the CHMC-TECO assist pump. This relatively simple counterpulse control system has been well tolerated by experimental animals. It has several apparent drawbacks for an implantable control system. These are 1) size, 2) weight, 3) efficiency, 4) pneumatic rather than hydraulic drive and 5) "triggering" is off an ECG "R" wave rather than a pressure pulse. The problem of size alone characterizes the difficulties. The volume of the implantable control system must be three orders of magnitude smaller than the external unit.

After a general discussion of the dynamics and controls of artificial heart systems, alternative approaches to control system design are discussed. Next, the engine control concept for the reference design is explained and the dynamic behavior of the system is analyzed.

Appendixes F (Direct Acting Control System), G (Controls for a Left Ventricular Assist Device), H (Inverter Blood Pump Control) and I (Differential Equation for Engine Response) give detailed analyses of specific aspects of the control problem.

A. EXTERNAL PNEUMATIC CONTROL SYSTEM

A block diagram of the external control system currently used to pneumatically drive the CHMC-TECO assist device is given in Figure V-1. This control system supplies compressed air at the proper pressure to eject the pump stroke volume to the pre-set value of assist pump end-systolic volume. Then the ventricle ejects its stroke volume into the pump at a low pressure which can be selected at the control system. The filled or end-diastolic volume of the pump varies with the ventricular stroke volume, but the end-systolic volume is maintained to ensure that the ventricular stroke volume is ejected each beat.

A volume sensor in the control system is used to monitor assist pump volume. This volume signal is also compared with the pump end-systolic volume reference signal to terminate pump systole. The recovery period between the end of pump systole and the next R-wave is maintained at a constant and selected value by automatically varying the air pressure which actuates the pump.

An ECG amplifier, "R"-wave detector and cardiometer are used with an oscilloscope to monitor the ECG and to supply the R-wave trigger signal to the control system. A variable delay is provided to adjust the time between the R-wave and the beginning of assist pump systole.

Since the delay and recovery time are fixed, the assist pump systolic duration decreases with increasing heart rate. The air pressure increases automatically to achieve this recovery time at the higher rate over a few heartbeats. Beat-to-beat increases in

PRESENT CONTROL SYSTEM FUNCTIONS

3-A

Figure V-1. Block Diagram of CHMC-TECO Pneumatic Control System

stroke volume or aortic pressure are compensated for by the longer systolic durations required to reach the same end-systolic volume

Filling pressure is controlled by valves at the control system. It can be monitored and controlled from -30 mm Hg to +200 mm Hg, thus regulating the work output of the heart.

Ventricular arrhythmias cannot trigger assist device ejection during the delay and ejection phase of pump operation. Simultaneous ventricular and assist device ejection with arrhythmias will result in ventricular pressures about the same as unassisted peak ventricular pressures.

The control system and assist device function together automatically to maintain cardiac output and the desired left ventricle pressure over a wide range of aortic pressures and heart rates. Ventricular pressures can be reduced to levels presumably low enough for a diseased left ventricle to pump all of the required cardiac output, reducing atrial pressures to normal levels.

B. COUNTERPULSE CONTROL SYSTEM

Although there are a number of ways that a counterpulse control system keying off the filling pressure pulse can be designed, such a control system usually contains the following components

Pressure transducer (e.g., piston-driven from hydraulic reservoir) for ejecting blood from pumps.

Pressure sensor (e.g., blood pump bladder) for triggering hydraulic reservoir

Volume sensor (e.g., rod with limited travel) for terminating pump systole

Hydraulic valve to switch drive piston cylinder from high to low pressure or vice versa (e. g. , spool valve).

Pilot valve to switch pressure levels across spool valve

The sequence of counterpulse blood pump logic can be mechanized in several different ways. Regardless of the particular hydraulic component arrangement, the usual logic sequence of the counterpulse control system is:

- (1) The hydraulic valve connects the high-pressure reservoir (accumulator) to the drive piston, which ejects whatever blood volume is within the pump.
- (2) A volume sensor detects that the blood pump bladder is almost completely collapsed, so that it cuts off the high-pressure supply to the drive piston to the low-pressure reservoir.
- (3) Left ventricle ejects into the blood pump at low pressure and returns the drive piston into position for the next stroke. Note that the position to which the drive piston is returned varies from beat to beat with the stroke volume. As the drive piston is returned the hydraulic fluid behind it is expelled into the low-pressure reservoir.
- (4) The trailing edge of the filling pulse is compared with a reference pressure. When the trailing edge of the filling pulse drops below the reference pressure, the drive piston cylinder is switched (i. e. , "triggered") from the low- to the high-pressure reservoir, thus completing the cycle.

C. DYNAMICS AND CONTROL OF ARTIFICIAL HEART SYSTEMS

The flow of information and consequent action in the normal cardiovascular system, suggested qualitatively in Figures V-2 and V-3, is extremely complicated and not well understood. (Figures V-2 and V-3 are taken from Physiology, Ed. E. E. Selkurt, Little, Brown and Company, Boston, 1966.) An artificial heart assist or complete replacement implies that some of the normal information or action based thereupon is missing. Thus, the agglomerate behavior of a normal system is not sufficient on which to base the final design of an artificial device. Rather it is necessary to understand how the individual signals and actions behave when isolated. Further, data describing the steady-state behavior is not sufficient, even if the various signals and actions are measured individually. The unsteady or transitory behavior determines whether the subject can stand up without fainting, for example, and even whether his partially artificial cardiovascular system is dynamically stable.

Sufficient data for the final design of a heart assist device or complete artificial replacement probably will not be available before such devices are implanted. In fact, use of artificial devices themselves is perhaps the best way to find the necessary data. The isolation of each function required for complete description of the system is not easily achieved without some mechanical assist to keep the system functioning. Early models of assist or replacement devices should be viewed primarily as experimental tools to collect physiological data, rather than as possible panaceas per se. Fastest progress will result from designing flexible experiments which can be and are operated to achieve a variety of data. Each new design should, of course, take cognizance of the data already accumulated.

Figure V-2. Operation of the Arterial Pressoreceptor Reflex in Controlling Systemic Arterial Blood Pressure

Figure V-3. Control of the Cardiovascular System

A large amount of steady-state data is presently available, in spite of its insufficiency. Only a small but particularly relevant fraction is mentioned here.

The pumping characteristics of the heart are presented in Appendixes A and B. This blood flow behavior results from the matching of the characteristics of the heart with those of the circulatory system, with numerous information flows. It is probably best viewed as the characteristic of the heart rather than the circulatory system. The circulatory system has a systemic resistance which is modulated by stimuli of the sympathetic vaso-constrictor and vasodilator nerves and various local control actions. Thus, this resistance is represented by a family of characteristics, each characteristic representing different stimulation. The intersection of the appropriate characteristic with the heart characteristic represents the consequent steady-state equilibrium behavior.

The heart characteristic results from a combination of factors. Nerve-stimulated veno-constriction is sensed directly by the heart, and influences the heart indirectly through the medula and hypothalamus. These latter organs also monitor a variety of other signals, and influence the heart through the vagus and cardioaccelerator nerves. Adrenal stimulation of the heart is more direct.

The heart could be replaced with an artificial device which achieved the same characteristic, although obviously by quite different methods. The vagus and cardioaccelerator nerves would be inoperative, and adrenal stimulation would also be missing. In its place, the atrial and system pressures can be utilized, plus the

actual cardiac output. The question now is whether the system would operate in fact as though an intact healthy physiological heart were present.

The answer is not clear, because of the cross-coupling signal flows between the systemic and heart portions of the normal system. If the initiation of a response is largely through the vasculature, the system should work nicely. The artificial heart responds properly to changes in system resistance. If the initiation of a response is largely through the heart itself, however, the severing of the normal nerves precludes proper response.

The actual situation probably is somewhere between these two extremes. Some control would be lost, though the extent depends on compensatory mechanisms in the remaining information transmission, processing and muscle organs. Possibly dynamic response would be damaged more than steady-state behavior. The recent heart transplants strongly suggest that loss of the vagus and cardioacceleratory nerves can be tolerated (though adrenal stimulation may still be significant).

To repeat: It is necessary to isolate the various parts of the normal system, by severing nerves or immobilizing muscles, etc., to achieve the data necessary for logical design. Some in vitro measurements of system resistance without nervous control might be helpful. Most useful information, however, probably will come from experiments in vivo with actual heart assist or replacement devices. The total replacement gives results which are easier to interpret, and which are more useful for designing an assist device

as well as a better total replacement. This requires that the device work well enough to permit the desired data to be taken.

The above considerations neglect dynamic effects which could be extremely important. Consider a small compliance for the atrial system and a large compliance for the venous system, in the fashion of the famed "Windkessle" model, as shown in the electrical analogy of Figure V-4. This model is unstable. The negative conductance of the heart is greater than the positive conductance of the system, given an equivalent negative conductance or resistance causing a non-oscillatory instability.

In fact, however, the arterial system has considerable inertance in addition to its small compliance. If we neglect the latter, the equivalent model of Figure V-5 results. The two resistances now are in series, giving a net positive resistance. This model is stable.

To resolve the question of stability, the arterial system is best modeled as an ideal bilateral delay line which permits waves to travel back and forth without absorption or dispersion. The venous compliance is so large as to be virtually infinite. The equivalent circuit now is shown in Figure V-6, along with two graphical calculations^{*} for a large and small characteristic impedances of the arterial system. The characteristic impedance is the square root of the ratio of fluid inertance per unit length to the compliance of the arterial wall per unit length. The arterial system with the larger characteristic impedance (more nearly inertive case) is stable, whereas the system with the smaller characteristics impedance (more nearly compliant case) is unstable.

* The graphical calculation is an especially simple example from the widely practiced method of characteristics. One of several texts on the subject is "An Introduction to the Method of Characteristics," Abbott.

Figure V-4. Unstable Windkessle Model

Figure V-5. Stable Modified Windkessle Model

(a) ELECTRICAL ANALOGY

(b) UNSTABLE BEHAVIOR WITH SMALL CHARACTERISTIC IMPEDANCE
(HIGHLY COMPLIANT ARTERIES)

(c) STABLE BEHAVIOR WITH LARGE CHARACTERISTIC IMPEDANCE
(STIFFER ARTERIES)

Figure V-6. Bilateral Wave Model of Arterial Dynamics

Normal physiological systems are stable, of course, like the first of the delay-line models above. (Note that the system becomes increasingly stable with age, since the arterial compliance diminishes.) Care must be taken in the design of an artificial heart replacement or assist device, however particularly if the natural heart characteristic is simulated. Compliance introduced inadvertently could introduce instability.

All the dynamic analyses above presume that the steady-state characteristics of the heart and the system resistance still apply under dynamic conditions. This presumption probably is significantly in error, and the answers might be misleading.

The conclusion is that data on the dynamics of the cardiovascular system, as well as steady-state, is needed for the logical design of artificial heart assist and heart replacement devices. The data preferably is taken with the heart isolated from the rest of the system.

D. ALTERNATIVE APPROACHES TO CONTROL SYSTEM

There are at least six different approaches to the control system for a permanent assist device system, as outlined in Table V-1. The first of these, the direct mechanical coupling of the engine and blood pumps, requires the complete replacement of the entire system including the blood pumps at the time of refueling. This must be considered an objection, at least until such time as it is proven that the mean life of the implantable power source is greater than the life expectancy of the patient. Another objection to this system is the fact that the entire unit must be located in one area. Since the entire unit will not

TABLE V-1

ALTERNATE APPROACHES TO CONTROL SYSTEM

APPROACH	PRIMARY ADVANTAGES	PRIMARY PROBLEMS
I. Direct mechanical coupling of engine and blood pumps. Direct control of engine output on a beat-by-beat basis.	<ol style="list-style-type: none"> 1. Straightforward and relatively simple. 2. Compact. 3. Low total system weight. 	<ol style="list-style-type: none"> 1. Must be located in the thoracic cavity. 2. Requires complete replacement, including the blood pump, at the time of refueling. 3. Complex engine controls particularly so for an assist device.
II. Direct hydraulic coupling of engine and blood pumps. Direct control of engine output on a beat-by-beat basis.	<ol style="list-style-type: none"> 1. Permits subdivision into two units— the power source and the blood pumps; interconnected by a flexible hydraulic coupling. 2. Permits easy replacement of power source 3. Provides a means of dissipating heat to the blood. 	<ol style="list-style-type: none"> 1. Difficult to synchronize with the biologic heart. (Applicable only to the assist device.)
III. Same as II, but used with a pacemaker, either fixed or variable rate. (Applicable only to assist device.)	<ol style="list-style-type: none"> 1. All advantages of II above. 2. Can be synchronized with natural heart. 	<ol style="list-style-type: none"> 1. Requires a surgical procedure to induce heart block.
IV. Maintain a near constant hydraulic reservoir pressure, indepently controlling the engine and blood pumps from this reference.	<ol style="list-style-type: none"> 1. Maximum decoupling of engine subsystem from hydraulic-blood pump subsystem. 2. All other advantages of II above. 	<ol style="list-style-type: none"> 1. Same peak systolic pressure every beat, or else a complex control is required.

TABLE V-1 (continued)

ALTERNATE APPROACHES TO CONTROL SYSTEM

APPROACH	PRIMARY ADVANTAGES	PRIMARY PROBLEMS
V. Hydraulic reservoir pressure varies directly with peak systolic pressure with independent controls to maintain cardiac output.	<ol style="list-style-type: none"> 1. All advantages of II above. 2. Very efficient. 3. Quite physiologic. 	<ol style="list-style-type: none"> 1. Impossibly complex control system.
VI. Engine with high inertial load; variable stroke positive displacement hydraulic pump which unloads during pump diastole.	<ol style="list-style-type: none"> 1. All of the advantages of III above. 2. Reasonable control devices. 3. Straightforward approach with acceptable physiologic characteristics. 	<ol style="list-style-type: none"> 1. Requires either a large or high speed flywheel. 2. High gyroscopic moments and forces associated with the high inertial flywheel.

fit in the thoracic cavity, it would have to be located in the abdominal cavity, which, in turn, would require at least one major arterial graft extending from the apex of the left ventricle, through the diaphragm, and into the blood pump. A seal would have to be provided around the periphery of this arterial graft where it penetrated the diaphragm. This would be rather difficult to accomplish surgically. Lastly, the unit would be difficult to synchronize with the left ventricle because of the direct mechanical coupling. Therefore, this approach can be dropped from further consideration as an assist device. However, as a total heart replacement, it may have more merit.

The second approach consists of a direct hydraulic coupling between the power source and the blood pump, where the output is controlled on a beat-by-beat basis. This approach circumvents the main objections to the direct mechanical coupling approach in that it permits the implantation of the blood pump subassembly in the thoracic cavity and the power source subassembly in the abdominal cavity, with a small diameter hydraulic cable connecting the two. There is a reasonably high probability that the blood pump can be located in the thoracic cavity, preferably just above the left hemidiaphragm and under the (left) lobe of the lung. A flat, disc-shaped profile would facilitate the implantation and the compatibility with the lung. Although there is ample room in the abdominal cavity for the power source, the hydraulic cable will have to penetrate through the diaphragm to interconnect the two subassemblies.

With the power supply located in the abdominal cavity, refueling should be simplified considerably. At the time of refueling, the surgeon would open the abdominal cavity, connect a new power supply,

switch over from the old to the new unit, remove the spent unit, secure the new unit, and close the abdominal cavity. As there would be no reason to enter the thoracic cavity, the patient should recover quickly since the only healing would be the adominal incision.

Synchronization of the direct acting, hydraulically coupled system is difficult. The engine must respond directly to the requirements of the heart and systemic circulatory system on a beat-by-beat basis.

The third approach shown on Table V-1 is the same system as that just described, except that the control system is based upon a primary signal from a pacemaker which is made a part of the system. The pacemaker would pace the heart at a controlled periodic rate. The same signal could be used to synchronize the power source, with fail-safe features in the event the heart failed to respond to a particular stimulus. The pacemaker could either be a fixed or variable rate type. The fixed rate pacemaker would imply a wider range of stroke volume, while a variable rate pacemaker would require additional engine controls to effect rate changes.

The disadvantage to this system is the fact that it will be necessary to surgically induce heart block in all patients including those with a normal sinus rhythm. Of course, considering the profile of patients who will be treated with permanent assist devices, it may turn out that many will require pacing in any event. Recall that these patients are those who have not properly responded to treatment with a temporary left ventricular assist device to the extent of being capable of resuming normal heart function. Data does not exist

on the number and symptoms of these patients since, at the present time, they all die. This illustrates the necessity for retaining those system approaches with maximum flexibility in order to adapt for patients' needs, once these needs are better understood.

A comparison of the relative merits of approaches II and III requires a more detailed evaluation and systems design of each in order to choose between the two. Therefore, both must be retained for further consideration.

Approach No. IV from Table V-1 consists of a system that includes a hydraulic reservoir between the power source and the blood pumps. The main attribute of this approach is the degree of decoupling between the blood pump and the power source. The addition of a high pressure reservoir permits the separate optimization of the engine and the blood pumps with minimum requirements for feedback control. This is possible while still retaining all of the advantages of approach II above.

In particular, this feature allows choosing the rpm, stroke, cutoff of the expansion and other engine features, relatively independent of the blood pumps. Control of the engine is based upon maintaining the high pressure reservoir at near constant pressure. Self-regulation of the engine should be possible. That is, the engine speed would increase, thus increasing the power output as the pressure in the reservoir began to fall, due to a greater demand from the blood pumps.

One blood pump control problem that must be faced is blood pump stall in the event of high peak systolic pressure. With a fixed pressure level in the hydraulic reservoir and a fixed mechanical

advantage in the coupling with the blood pump, the maximum systolic pressure against which the blood pump can eject blood is fixed. If the blood pump stalls, the patient would be highly distressed. However, the stalling would not necessarily be fatal since the peak systolic pressure would probably drop and pumping could resume.

One approach to circumventing this difficulty is to set the reservoir pressure so high that it will not stall even under the highest systemic pressures and control down to any lesser pressure level. The disadvantage to this approach is that it is dissipative and results in an extremely low system efficiency (average) since the majority of time the systemic pressures will be considerably less than this maximum. The blood pump control design in Appendix G could be used in a system with a high pressure reservoir.

Approach V is a variation on the previous approach, except the basic difficulty attributable to the constant pressure reservoir has been eliminated. In this system, the pressure in the hydraulic reservoir varies directly with peak systolic pressure. In principle, this solves all of the problems of the previous approach. It would be very efficient under all physiologic conditions, and, in fact, would have very acceptable physiologic characteristics. However, it requires a more complex control system.

The final approach shown in Table V-1 evolved as a result of the effort on the blood pump controls. This system is ideal from that standpoint. The system employs a positive displacement, variable stroke volume hydraulic pump directly coupled to the blood pump. All of the blood pump control is accomplished on the hydraulic pump,

which can be done simply and in a very physiologic manner. During pump diastole, the engine would be completely unloaded except for its inertia. Therefore, a high inertial engine system would be required (i. e., high speed, flywheel) in order to keep engine speed variations within acceptable limits. The hydraulic pump would draw whatever power it needed for each stroke on a beat-by-beat basis from the flywheel. Engine control would be accomplished by averaging the flywheel speed over several beats to detect changes in the average flywheel speed. Centrifugally controlled valves would effect engine power changes to maintain near constant average flywheel speed. This system is discussed in more detail in Appendix F.

This system would be efficient and have a very straightforward control system which would have very acceptable physiologic characteristics. The system could be easily developed and should prove to be very reliable.

Although this system would be efficient, the "Achilles Heel" to this approach is the effects of the high inertial flywheel. The gyroscopic movements and resultant forces are sufficiently high with this system as to make it prohibitive. Not only would rapid accelerations in a plane or car be disastrous, but even turning a corner too rapidly would severely jolt the patient. In spite of the otherwise attractive aspects of this approach, it must be discarded because of this fundamental problem, which is an inherent characteristic of the system.

After evaluating various systems approaches, it reduces itself to a selection between two basic approaches: that with a direct acting hydraulically coupled power source and blood pump, versus a system with a high pressure reservoir. Pacemakers could be used with

either system. On the basis of Thermo Electron's experience with steam engines, blood pumps and blood pump controls the high pressure reservoir (accumulator) approach seems most feasible.

E. ENGINE CONTROL

The engine and blood pump controls are separated by a hydraulic reservoir (i. e., hydraulic accumulator) interface. In order for the blood pump to have the correct systolic duration, the hydraulic reservoir pressure must be maintained at a relatively constant pressure. The degree of constancy will be determined by the engine control system. During periods of high demand (i. e., increased blood pump rate or stroke volume), the engine speed must increase so that more fluid will be pumped into the hydraulic reservoir to maintain its pressure. During periods of low demand, the engine response must be such that its speed will decrease to keep reservoir pressure relatively constant and prevent engine stall.

A block diagram of the engine - high pressure reservoir - blood pump system is shown in Figure V-7. The blood pump, depending on the heart rate and stroke volume, drains flow out of the high pressure reservoir Q_{out} . The hydraulic pump, depending on engine RPM, supplies flow into the high pressure reservoir -- Q_{in} . The high pressure reservoir is both a summer and an integrator. The reservoir sums Q_{out} and Q_{in} to determine $\Delta Q = Q_{out} - Q_{in}$. Since Q_{out} and Q_{in} are functions of time ΔQ is also a function of time. The integral of ΔQ with time determines the instantaneous pressure in the high pressure reservoir. The instantaneous pressure in the high pressure reservoir is actually the instantaneous load on the engine which,

Figure V-7. Engine - High Pressure Reservoir-Blood Pump System

Figure V-8. Hydraulic Negative Feedback Engine Control System

depending on the resistance, compliance, and inertance of the entire system, causes the engine to adjust its speed. The block diagram of the engine control system is redrawn in Figure V-8. The control system constitutes a negative feedback system in which the hydraulic reservoir functions as the error detector or subtractor. The error signal is $\Delta Q = Q_{\text{out}} - Q_{\text{in}}$

The best way to describe the system is to formulate the differential equation which describes the energy transfers occurring within the system. One engine cycle involves many non-linearities, as testified by the indicator diagram of the engine. To simplify the analysis, the average energy per cycle will be used rather than instantaneous energy transfers. That is, the d^2w/dt^2 of the engine and its flywheel are assumed to be zero within one cycle. This assumption will be valid if the energy stored in the flywheel is such that the difference between the duration of piston downstroke and piston upstroke is small. The derivation of the system differential equation is given in Appendix I.

F. DYNAMIC BEHAVIOR OF SYSTEM

A compliant (accumulator or reservoir) coupling between the engine and heart pump allows the engine to work continuously, during both heart systole and diastole, permitting minimum weight and size and maximum efficiency. The compliance acts as a buffer storage of energy. An inertive coupling could perform the same basic function, but is more complicated to implement and has undesirable dynamic effects.

Some inertia (almost certainly in the form of a flywheel) is necessary merely to return the steam piston after each power stroke. This inertia also acts in conjunction with the compliance (of the reservoir) to determine the principal features of the dynamics of the entire artificial heart assist device. Proper selection of the parameters gives highly desired characteristics, including dynamic self-regulation without elaborate controls. Thus it is appropriate to investigate the dynamic behavior of the system.

The dynamic behavior is influenced by the physiological control system of the remaining natural cardiac system, which responds to external stimuli in a manner not understood in quantitative detail. Thus any dynamic analysis based on present data will not yield precise results. Consequently the exactness of predictive results is not significantly decreased by simplifying assumptions about the mechanical portion of the device. These assumptions introduce errors of the order of less than 5 or 10 percent. Only simple analyses are appropriate at this point. When cleverly done, they also reveal more starkly the principal features of a design.

The mean frequency of the steam piston (600 rpm) was selected for efficiency, compactness and reliability. This frequency is about ten times the nominal heart rate and only very rarely falls below five times the actual heart rate. Little error in the overall dynamic analysis results from assuming that the engine and water pump combination produces a steady flow of water at a rate proportional to the speed of the engine. Moreover, this assumption vastly simplifies the analysis and clarifies the meaning of the results; it is made here.

The system is shown in Figure V-9. The "inertia" results principally from the flywheel, although other moving parts, including those in the water pump and the water itself, contribute to the kinetic energy. The "compliance" is associated principally with the spring-loaded reservoir or accumulator, although some contribution may be made by the tubing. The electrical analogy is based on the analogous behavior of pressure and voltage on one hand, and volume fluid flow and electric current on the other hand. Note that all velocities and flows (or currents) are referred to the water flows. This implies transformational couplings (not shown) between the flywheel speed and water pump flow, etc. The "resistance" includes frictional losses in the steam piston, flywheel, and water pump, plus most of the load of the feedwater pump, plus the mean effective orifice loss of the check valves in the water pump.

The mechanical heart pump presumably contains a fluid-mechanical transformer, which couples low flow and high pressure on the engine side to high (cardiac) flow and low (systolic) pressure on the physiological side. This permits the engine to be small, light and efficient at the smaller expense of the size, weight, and inefficiency associated with the transformer itself. The heart pump also contains inertance (mass) which actually can improve the dynamics by inhibiting rapid motions. This energetically conservative filtering action might be purposely augmented (by a venturi, for example) to slow down the mechanical systole. Some dissipative filtering may also be necessary, partly to accommodate a range of systolic pressures and partly to cushion the end of systole. This filtering can be accomplished by a

Figure V-9. Power Coupling with Approximate Circuit Model

variable orifice such as in a fluid control valve. Cushioning might also be accomplished by conservative springs, however, although complications are introduced. The electric circuit model for the mechanical heart shows the transformer, inductance (inductance) and modulated resistance features.

The boiler design with its thermal energy storage and near-equilibrium output temperature implies that the effective force on the piston is determined largely by the pressure produced by the feed-water pump. A local control loop can provide stability and ensure that the quantity of water in the boiler stays within limits. Beyond this it is possible either to hold the steam pressure nearly constant, regardless of the mean heart rate at the time of interest, or to attach a larger feedback loop to adjust the steam pressure in the face of load (heart) changes. The first possibility has the virtue of simplicity, whereas the second has the potential virtues of improved physiological matching and possibly increased power efficiency. A logical decision between the two must await physiological data not currently available. The best approach probably is to design and test the first option in vivo, and then if the physiological data warrants it, append the larger feedback control. This control may well be unnecessary.

The essential dynamic behavior of the core system is thus best represented by assuming a constant steam pressure. This produces a mean effective piston force (represented by its transformed equivalent pressure source) which is virtually constant, independent of flow (speed). Since the dynamics of the physiological systems are not known quantitatively, and the details of the mechanical heart pump

are not specified, it will be assumed that the physiological system demands a specified flow, Q , which changes with time. The response of the power supply unit, namely, the flow or speed (of the flywheel, etc.), and the reservoir pressure, can be computed. If the reservoir pressure changes drastically, the specified flow will not actually result in vivo. Interpretation for a variety of situations is fairly direct. The dynamics are largely controlled by the compliance and inertance in nearly any circumstance.

1. Static Characteristics

The static characteristics of interest are the water pressure and compliance volume, which correspond to various steady cardiac outputs. In Figure V-10 the resistance characteristic is plotted (non-quantitatively) on the left. A given steady flow Q_0 specifies a given steady pressure, p_0 , which in turn indicates the volume of the spring-loaded reservoir, as shown on the right. The useful volume of the water in the compliance is presumed to be proportional to the pressure applied by the linear spring. The resistance characteristic actually will not be so linear, however, and is much more difficult to determine, requiring actual experiment. Both characteristics can be modified within limits by appropriate design changes.

The fundamental attribute of this simple system is that increased flow produces somewhat decreased pressure. Thus its virtue, aside from simplicity, is absolute stability. Its fault is that the matching with the physiological system probably is not as nearly ideal as possible. The extent of this fault probably is not great, since the most natural control of cardiac output appears to be associated with modulated

Figure V-10. Steady-State Characteristics of Engine-Pump and Water Accumulator

systemic resistance rather than modulated systolic pressure. However, the actual adaptation of the physiological system to this simple proposed scheme must be directly observed to evaluate in detail its effectiveness.

The boiler-pressure control referred to earlier could be set to make the effective resistance negative. The control would then simulate some of the lost function of the physiological heart, with the potential of nearly ideal performances under all loading conditions. All of the complex loss functions are not simulated, however, and the overall system might prove to be unstable. Steady-state data of itself cannot preclude this possibility. Dynamic data is necessary; in vivo experiments with this more complex system are most probably the best approach.

2. Dynamic Characteristics

The dynamic characteristics are determined largely by the inertia I and compliance C of the second-order coupling, and to a lesser extent by the resistance R .

The inertia I must be large enough to return the steam piston for the next power stroke. The kinetic energy stored in the flywheel must overcome the resistance, requiring about 20% of the total energy of the power stroke.

To interpret this requirement, the following symbols are defined:

Q_o	Nominal or mean value of Q (at nominal heart rate), in. ³ /sec.
r_o	Nominal heart rate, beats/sec.

V_o	Nominal volume flow for nominal heart beat, Q_o/r_o , in. ³ ,
p_o	Nominal value of pressure p , lb/in. ² ,
$p_{\max}$, $p_{\min}$	Maximum and minimum values* of p , lb/in. ² ,
a	Ratio $p_{\min}/p_{\max}$ ($a < 1$),
T	Mechanical advantage of mechanical transformer associated with heart pump ($T < 1$),
f_o	Nominal frequency of piston, cycles/sec,
V	Useful reservoir volume, in. ³ ,
n	V/V_o ,
C	Compliance of reservoir, in. ³ /psi,
I	Inertance of flywheel (as seen by fluid), psi in. ³ /sec ³ .

The statement above regarding minimum inertance becomes:

mean energy in $I \geq 0.2 \times$ energy in one stroke of piston

or,

$$\frac{1}{2} I Q_o^2 \geq 0.2 \frac{r_o}{f_o} p_o V_o$$

$$I \geq \frac{0.4 r_o p_o V_o}{f_o Q_o^2} = \frac{0.4 p_o}{f_o r_o V_o} \quad (1)$$

* This is for the linear range, i. e. , presuming the spring is not resting against either mechanical stop. At one limit cavitation occurs, giving nearly zero pressure; at the other limit a relief valve is provided.

The maximum steady flow occurs at minimum reservoir pressure. This pressure should not be too low, however, to keep the deviation from normal physiological characteristics and/or the energy dissipation within reasonable bounds. A reasonable limit probably is approximately

$$a \equiv \frac{p_{\min}}{p_{\max}} \geq 0.5 \quad (2)$$

(Note that the engine cannot stall so long as $p_{\max}/T$ exceeds the systolic pressure, and it should be by a considerable margin. The system is most efficient at high cardiac outputs, which is desirable to minimize size and weight. Too small a value of "a" necessitates dissipating excessive energy under rest conditions.)

Approximately two-thirds of the flow of each heartbeat is temporarily extracted from the compliance reservoir and the remainder comes directly from the engine. Further, the reservoir normally is only partially full. At a rather high flow rate, when the reservoir is only one-quarter full, we would still like to be able to extract this quantity of water, which under this condition would be approximately $1.0 V_o$ (or three-halves nominal water extraction). Thus the following approximate requirement applies:

$$n \geq 4 \quad (3)$$

Should the reservoir fill completely, the relief valve simply vents the excess flow to prevent the engine from stalling. One must also consider the opposite problem of an empty reservoir, which greatly slows the heart pump temporarily (while the steam engine accelerates to full speed). This should not happen for moderately

heavy and steady cardiac outputs if the foregoing decisions are implemented. The entire system will have some limit to the steady cardiac output it can maintain, however. An attempt to exceed this limit results in highly transitory emptying of the reservoir during each beat. With proper design no asynchronies occur, and the host merely is forced to use less energy in a manner similar to that of a normal person. (Officials of the Olympic Games might bar owners of artificial hearts from endurance events, deeming their competition unfair.)

The behavior is more complicated during unsteady demands. The critical case is a sudden increase in demanded cardiac output. The engine responds in the manner of a simple second-order dynamic system, as shown in Figure V-11. The cross-hatched area represents the maximum temporary net decrease in the volume of water contained in the reservoir, which for reasonable amplitudes of the step should not exceed the critical volume. The plot does not show individual heartbeats, several of which are presumed to occur during the recovery time, but rather a somewhat filtered time average. Unless the system is very highly damped the frequency of oscillation is nearly the same as the undamped natural frequency.

$$\omega_n = \sqrt{\frac{1}{I C}} \quad (4)$$

The recovery time is approximately one-third the period, or about $2/\omega_n$ seconds, and the cross-hatched area is about $1.2 \Delta Q/\omega_n$ (the value for zero damping is $1.0 \Delta Q/\omega_n$, in which ΔQ is the magnitude of the step of flow).

It is preferable to have ω_n as large as possible, or the inertance I and compliance C as small as possible. The minimum value of I is

given by Eq. (1). The compliance of the reservoir is the ratio of the volume change to consequent pressure change, or

$$C = \frac{nV_o}{p_{\max} - p_{\min}} = \frac{1+a}{2(1-a)} \frac{nV_o}{p_o} \quad (5)$$

The compliance thus is minimized when a and n assume their minimum values as given by Eqs. (2) and (3), respectively. Thus,

$$\omega_{n \max} = \sqrt{\frac{f_o r_o}{24}} \quad (6)$$

The ratio of minimum recovery time to nominal beat rate is approximately

$$\frac{2r_o}{\omega_{n \max}} \approx 10 \sqrt{\frac{r_o}{f_o}} \quad (7)$$

A typical value of the ratio f_o/r_o is 8, which would give a minimum recovery time of about 3.5 nominal heartbeats (or about 2.3 seconds for $r_o = 90$ beats per second). This seems entirely satisfactory, but one would not wish to deviate very far from the limiting situation.

Also of interest is the amplitude of the flow step which, when superimposed on a steady-state condition of a half-full reservoir, comes close to emptying the reservoir. This condition implies

$$1.2 \frac{\Delta Q_{\max}}{\omega_n} \approx \frac{1}{2} n V_o$$

or

$$\frac{\Delta Q_{\max}}{Q_o} \approx \frac{n V_o \omega_n}{2.4 Q_o} = \frac{n \omega_n}{2.4 r_o} \approx \frac{1}{3\sqrt{r_o/f_o}} \quad (8)$$

which, for f_o/r_o equals 8, permits a maximum flow step of nearly the same amplitude as the initial flow, Q_o . This also seems quite satisfactory, though it also may not contain a large enough margin to permit significant changes in the parameters I , a , n , etc.

The dynamic analysis above suggests that the parameters

$$\begin{aligned} I &= \frac{0.4 p_o}{f_o r_o V_o} \\ a &= 0.5 \\ n &= 4 \end{aligned} \tag{9}$$

give satisfactory performance, and are close to optimum. Minor changes are inevitable in the face of better physiological data. However, the simple technique discussed in this section is more important than the specific numerical results.

G. PARAMETERS FOR POWER COUPLING

Expressions for the desired inertance of the flywheel and compliance of the fluid accumulator (reservoir) were given in the previous section. These results specify the characteristics of the flywheel and reservoir but not their physical structure. These parameters were referenced to the flow and pressure of water in the high pressure portion of the system. Here these parameters are converted to actual dimensions and stiffnesses, etc.

The volume of water pumped in one cycle of the piston-cylinder water pump is $S_H A_H$, where S_H is the stroke and A_H the piston area. Thus, the angular velocity of the flywheel shaft, presuming it makes one revolution per cycle of the piston, is

$$\omega = \left(\frac{2\pi}{S_H A_H} \right) Q \quad (10)$$

where Q is the average flow rate of the water.

We identify the inertia I as before, and introduce a mechanical inertia I' referenced to the rotation of the shaft:

$$\text{Kinetic Energy} = \frac{1}{2} I Q^2 = \frac{1}{2} I' \omega^2 \quad (11)$$

Thus,

$$I' = \left(\frac{S_H A_H}{4\pi} \right)^2 \left(\frac{4 p_o}{f_o r_o V_o} \right) = \frac{S_H^2 A_H^2 p_o}{\pi^2 f_o^2 r_o^2 V_o} \quad (12)$$

If the flywheel is geared to run faster than the pump by the ratio R , then the inertia (and weight and size) of the flywheel can be reduced:

$$I' = \frac{S_H^2 A_H^2 p_o}{\pi^2 f_o^2 r_o^2 V_o R^2} \quad (13)$$

If the flywheel has a mass m and a characteristic radius b , then

$$I' = \frac{m b^2}{2} = \frac{S_H^2 A_H^2 p_o}{\pi^2 R^2 f_o^2 r_o^2 V_o} = \frac{p_o r_o V_o}{\pi^2 R^2 f_o^3} \quad (14)$$

The maximum radius of the flywheel of course could be somewhat smaller than b , since much of its mass can be concentrated near the rim.

For the typical values

$$\begin{aligned} p_o &= 75 \text{ lb/in.}^2 \\ r_o &= 90 \text{ beats/min.} \end{aligned} \quad (15)$$

$$f_o = 8 r_o \quad (15 \text{ cont.})$$

$$R = 1$$

$$V_o = 0.20 \text{ in.}^3$$

there results

$$\frac{m b^2}{2} = 0.132 \times 10^{-2} \text{ lb sec}^2 \text{ in.} \quad (16)$$

This is typified by a steel disk 0.4 inch thick and 2.6 inches in diameter. Use of gear couplings ($R \geq 1$) reduces this size sharply.

The fluid accumulator (reservoir) is shown in Figure V-12. The bellows assure total sealing of the reservoir; almost all of the force is provided by the helical spring (or springs).

The energy stored in the spring(s) is

$$\frac{1}{2} \left(\int Q dt \right)^2 = \frac{k}{2} \left(\int V' dt \right)^2 \quad (17)$$

where V' is the velocity of the piston face. This integration yields

$$k = \frac{A_p^2}{C} = \frac{2(1-a)}{1+a} \times \frac{A_p^2 p_o}{nV_o} \quad (18)$$

Applying the design suggestions, $a = 0.5$ and $n = 4$, gives

$$k = \frac{A_p^2 p_o}{6V_o} \quad (19)$$

plus the requirement

$$\frac{1}{2} \times \max A_p = 4 V_o \quad (20)$$

1253-D

Figure V-11. Flows into and out of Reservoir (Accumulator)
Following Step Increase in Demanded Cardiac Output
(Individual Beats Not Shown)

1420-D

Figure V-12. Fluid Accumulator (semi-schematic)

in which $x_{\max}$ is the maximum compression of the spring from its free length. Reasonable proportions of the reservoir could be a maximum piston travel equal to the radius of the bellows, or

$$A_p = \pi \left(\frac{x_{\max}}{2} \right)^2 \quad (21)$$

Equations (19) through (21) now can be combined to give

$$x_{\max} = \left(\frac{32}{\pi} V_o \right)^{1/3} \quad (22)$$

$$k = \frac{\pi}{3} x_{\max} p_o \quad (23)$$

for the typical values

$$\begin{aligned} V_o &= 3.4 \text{ cc} = 0.20 \text{ in.}^3 \\ p_o &= 75 \text{ psi} \end{aligned} \quad (24)$$

this gives

$$\begin{aligned} x_{\max} &= 1.3 \text{ inches} \\ k &= 100 \text{ lb/in.} \end{aligned} \quad (25)$$

The spring would weight about 0.1 lb for a working stress of 100,000 lb/in.². Fortunately, these parameters are acceptable with respect to size, weight and gyroscopic effect.

H. ANALOGUE COMPUTER RESULTS

The dynamics of the individual heartbeats plus transient changes in demand were programmed on the analogue computer of the Mechanical

Engineering Department of the Massachusetts Institute of Technology. The pressures and flows of interest were monitored in real time for flow demand functions which were properly clipped sine waves. A wide variety of disturbances and parameters were tried. The procedure gave no new hard answers, but demonstrated the general correctness of the analytical results and that the choice of parameters is important.

Typical analog output traces for constant cardiac output are sketched in Figure V-13. Over a wide range of flywheel inertias, engine RPM remains constant. With an aluminum flywheel, there is only a 10 percent ripple indicated by the dotted lines in Figure V-13. The change in hydraulic pressure ΔP , is dependent on flywheel size. For a high-pressure reservoir at a mean pressure P_M of 80 psia, the maximum ΔP is 7 psia and the minimum is 2 psia. The maximum occurs for the smallest flywheel inertia and the minimum occurs for the largest flywheel inertia. It is interesting to note the phase difference between $Q_{out}(t)$ and $P_H(t)$. The peak of $P_H(t)$ lags behind the peak of $Q_{out}(t)$. The amount of lag is proportional to the flywheel inertia. The lightest flywheel produces the least lag and the heaviest flywheel produces the most lag. The effect is produced because of the energy stored in the flywheel.

Perhaps the more interesting case is the transient. When the cardiac output increases or decreases from an equilibrium condition, the engine RPM must readjust to meet the new demand. Some typical analogue output traces for a step increase in cardiac output are sketched in Figure V-14. Before a new equilibrium condition is reached both the engine RPM and the reservoir pressure, $P_H(t)$, go

through several damped oscillations. The amount of overshoot is determined by the degree of damping or resistance. The resistance term of the engine depends on its mechanical efficiency. If the mechanical efficiency is low, the damping will be high. Hence the overshoot, ΔRPM , will tend to be small and the number of overshoots will be small. However, if the mechanical efficiency is high, the magnitude of ΔRPM will be large and the number of overshoots will increase. If the mechanical efficiency is too high, the engine will "run away" on the high end of the overshoot and stall out on the low end of the undershoot. Of course, a pressure relief valve on the hydraulic reservoir will prevent stalling. The RPM of the engine increases when the cardiac output increases because the fluid in the hydraulic reservoir has been temporarily drained. The draining of the reservoir means that the load on the engine has changed. There is a time lag, due mainly to the flywheel inertia, before the engine responds. However, the energy stored in the flywheel will cause the RPM to overshoot its equilibrium value. When this happens, the high pressure reservoir is in danger of filling to capacity and "topping out." To avoid this, a relief valve can be employed that will shunt fluid through the reservoir in a dummy circuit.

"Bottoming out" could occur during a severe undershoot. Because of the system lag, the engine will undershoot its equilibrium RPM after the first overshoot in RPM and hydraulic pressure. If the undershoot is severe, the Q_{out} demanded may be higher than the Q_{in} and after a while the hydraulic reservoir piston will bottom. The blood pump will stop and the hydraulic pump may cavitate causing

Figure V-13. Steady-State System Response

Figure V-14. System Transient Response

the next overshoot to be higher than the first instead of damped. The system will then become unstable. One of the ways to avoid this situation is to adjust the engine mechanical efficiency. When the cardiac output demand decreases suddenly, the engine speed and hydraulic pressure will oscillate in a manner similar to the case described above but in the opposite direction.

The analog results of the effect of mechanical efficiency on engine damping point to the desirability of a hydraulic reservoir relief valve. The relief valve would decrease mechanical efficiency during transients and clamp out oscillations. This would allow the steady-state mechanical efficiency to be made as high as possible.

Consider next the effect of flywheel inertia on engine transient response. Referring to the sketch of transient response, Figure V-14, the quantity τ_n is the duration of one complete oscillation. The flywheel inertia partially determines this duration. A large flywheel means a large natural frequency period, τ_n , and a small flywheel means a small τ_n . However, the inertia affects neither the number of oscillations nor the degree of oscillation (i. e., ΔRPM).

In order to keep the steady-state speed relatively constant, the flywheel inertia should be high. However, the inertia of the flywheel should be low enough that new steady-state conditions can be reached quickly. The original assumption in deriving the system differential equation now becomes important. The work of the engine and thus the engine RPM was averaged over one cycle. Therefore, the actual change in engine RPM may be higher than that indicated by the analogue computer output. This is especially true if the flywheel

inertia is small. The best approach is to adjust the flywheel inertia so that the natural frequency of the engine is higher than the maximum heart rate. This would allow the oscillations of the engine to be almost damped out before the next heart beat. That is, if τ_n were smaller, more damping will occur before the next heart beat.

The natural frequency of the engine is indicated by the system differential equation. The square root of " ω " coefficient is the natural frequency ω_n .

$$\omega_n = \sqrt{\left(\frac{HR \cdot S_p \cdot \bar{N}_H}{480 \pi^2 N \bar{N}_R \overline{RPM}^2 U} \right) \left(\frac{P_{HI}^2 - P_{LOW}^2}{IP_D^2} \right)} \quad (26)$$

The quantity in parentheses is a constant, $M^2 =$

$$\frac{HR \cdot S_p \cdot \bar{N}_H}{480 \pi^2 N \bar{N}_R \overline{RPM}^2 U} \quad (27)$$

$$\omega_n = M \sqrt{\frac{P_{HI}^2 - P_{LOW}^2}{IP_D^2}} = \frac{M}{\sqrt{IP_D}} \sqrt{(P_{HI}^2 - P_{LOW}^2)} = \frac{M}{\sqrt{IC}} \quad (28)$$

$$C = \frac{P_D^2}{P_{HI}^2 - P_{LOW}^2} \equiv \text{the system compliance}$$

$I \equiv$ flywheel inertia or system inertance

Thus, it is obvious from the equation for the natural frequency that the flywheel inertia and system compliance can be adjusted to achieve the desired speed of response together with a relatively constant speed.

The natural frequency of the system ($\omega_n = \frac{M}{IC}$) affects hydraulic reservoir "droop." This term refers to the fact that the mean hydraulic pressure is inversely proportional to the cardiac output within a range determined by the natural frequency. A high natural frequency will result in a small "droop," range, whereas a low natural frequency yields a large "droop" range. A large "droop" range increases the danger of the hydraulic reservoir "bottoming" or "topping out." It also affects the operation of the blood pump because of the wide mean hydraulic pressure variations.

The "droop" effect is a direct result of the system lag and speed of response, both of which are determined by the natural frequency. When cardiac output drops suddenly from a steady-state condition, a surplus of hydraulic fluid is built up because of the system lag and response time. As the new equilibrium point is reached, no compensation is made for this surplus. The mean hydraulic pressure is raised. When cardiac output increases suddenly from a steady-state condition, a fluid deficiency is created and the mean hydraulic pressure of the new steady-state is lower. Thus, both the "droop" problem and the speed of response problem point to achieving as high a natural frequency without sacrificing near constant engine speed during steady-state conditions.

CHAPTER VI.

IDENTIFICATION OF MAJOR PROBLEMS

A. INTRODUCTION

Most of the major problems associated with nuclear-fueled implantable circulatory systems fall within two groups--engineering and medical. The engineering problems can be further delineated into the subgroups of efficiency and reliability. The medical problems can be categorized into those of radiobiological effects, blood-materials interface, additional endogenous heat, blood pump control, and surgical implantation.

1. Efficiency

The feasibility of an implantable power source for a circulatory support system hinges on a relatively high system efficiency in order to minimize: the heat to be dissipated in the body, the radiation dose rate, and the cost. Therefore, those factors which most affect efficiency must be considered problem areas until demonstrated otherwise. These include the usual sources of inefficiencies one normally expects: such as losses from the heat source, minimum conduction losses in engine structural components, high actual cycle efficiency, and efficient auxiliaries and controls. In addition to these areas where one would normally anticipate inefficiency, there is another less obvious source of losses. This is the coupling and control system of the power source with the blood pump(s). If one assumes relatively fixed conditions of cardiac output and peak systemic pressure, it is relatively easy to design an efficient coupling and control system. However, if one anticipates the normal variations that are likely to occur, then it is quite difficult to design a coupling and control system that is, first,

adequate to provide the necessary control function, and secondly, that is efficient most of the time. This is a crucial area as far as maximizing overall system efficiency is concerned. It is also an area where good input data or specifications do not exist. However, it has too great an effect on the choice of the overall system approach to be ignored. The development of high-performance heat engines, even more with dynamic-energy converters than with static converters such as thermionic and thermoelectric generators, is a matter of refined mechanical design rather than analytical sophistication. Thermo Electron's experience in steam engine power supplies has shown that choice of gross design parameters is rather straightforward. The iterative cycles of test and refined design are most expensive in time and money. However, the analytical promise can only be realized through painstaking tests, critical interpretation of data and design excellence.

2. Reliability

Since the utility of the implantable power source hinges on its reliability, this also must be regarded as a problem. Specifically, this requires the development and demonstration of the valves, the bearings, and the flow controls and regulators. Reliability is difficult to discuss at the conceptual stage. The power source is an intricate mechanism and there is little room for redundancy. Therefore, individual component failure rates must be extremely low.

Although it is acknowledged to be an important factor in complex systems, reliability is difficult to quantify at the conceptual design stage. Only after the design is stabilized and component failure rates have been measured, can the system reliability be estimated with

confidence. Until that time, technological history is probably a better harbinger of reliability than detailed reliability analyses without meaningful input data. Elimination of unnecessary components is a characteristic of good design. However, addition of components to decrease failure rates of all components may enhance reliability. Thus, the elimination of the lubrication system on high performance machinery can hardly be expected to improve its life. Thermo Electron's approach to long life operation is to design each essential component in a highly conservative manner, lubricate it properly and run it at the lowest temperature consistent with system efficiency. Achievement of the longevity goals of the artificial heart program will require extensive testing.

That mechanical system such as automatic transmissions and jet engines do operate for long periods without maintenance is evidence that a reliable and efficient circulatory support system is possible. However, the present high performance of jet engines and automatic transmissions is a result of investing tens of years and tens of millions of dollars in their development. The realization of a clinically suitable radioisotope fueled circulatory support system may well require a comparable commitment.

If one assumes that the most critical components are in the engine power train, then, based on Thermo Electron's experience to date, one can expect state-of-the-art steam engines to run at least 3600 hours. Thus a five year life for the circulatory support system appears to be a reasonable goal if adequate funds and time are allocated to its development.

B. RADIOBIOLOGICAL EFFECTS

The potential radiation risks inherent in the wide-spread application of radioisotope fueled implantable power sources can be put into perspective by a review of the pertinent radiobiological data. The radiation effects to the host and the population at large under absolute fuel containment will be considered first. Next, the crucial problem of fuel release under highly unlikely, but still credible conditions will be briefly discussed. Although it is beyond the scope of the present study to consider either problem in depth, it is important that the problem areas be identified and characterized to the extent that available data permit.

Radiation exposure guidelines necessarily involve as much philosophy as physics and biology. The philosophy of radiation exposure applied to the artificial heart program requires fundamentally different considerations from those relevant to occupational radiation protection standards. Occupational standards are based on the upper exposure limit being as low as economics and practical application will allow while still deriving the benefit of the activities in question. However, a radioisotope power source would provide its host with a vista of personal freedom that can never be achieved with electrical rechargeable units. Just as society presently allows its members the option of statistically shortening their life spans by smoking, it is only reasonable that society should also allow individuals implanted with circulatory assist devices the option of leading a statistically shorter but more meaningful life.

Fortunately, the Federal Radiation Council (Federal Radiation Council, Background Material for the Development of Radiation

Protection Standards, Report No. 1 - May 13 1960; inserted in Federal Register, Document No. 60-4539, pp. 4402-4403 - May 17, 1960) has officially introduced a much more flexible approach to the problems of radiation protection than is applied to occupation exposure. This approach is based on the principle that criteria for radiation protection depend on a balance between the risk of radiation injury and reasons for accepting the exposure. To quote the Council, ". . . there can be no single permissible or acceptable level of exposure, without regard to the reasons for permitting the exposure. . . and . . . there can, of course, be quite different numerical values for the RPG (Radiation Protection Guide) depending on the circumstances. . . the Radiation Protection Guides provide a general framework for radiation protection requirements. It is expected that each Federal agency, by virtue of its immediate knowledge of its operating problems, would use these guides as a basis upon which to develop detailed standards tailored to meet its particular requirements." Thus, the artificial heart program is not bound by the Federal Radiation Council to a single set of "permissible dose" values. Rather, the AEC and/or the NIH are obligated to balance the risk of radiation exposure against the benefits to the individual and society. Too restrictive protection criteria may vitiate the advantages of the radioisotope powered circulatory support device.

The radiation environments around the various candidate radioisotope heat sources are given in Chapter III. For purposes of the present discussion, assume that "medical grade" Pu-238 will fuel the implantable power source. Assuming minimal shielding and central location of the fuel within the power source, the maximum surface dose rate should be less than 100 mrem/hr. At a distance of 5 cm

from the power source surface, the dose rate should not exceed 20 mrem/hr.

It is of prime importance that the radiation exposure from the radioisotope does not result in uniform whole-body dose. Animal experiments have shown that nonuniform exposure is less effective than uniform exposure in producing both early and late radiation manifestations such as early lethality, decrement in hematopoietic function, actuarial life-shortening and incidence of neoplastic disease. The radiation response of the organism will depend on the spatial distribution of the dose in relation to the location of the principal critical organs and their relative radiosensitivity. Organ redundancy may well be an important factor in mitigating the biological system response to radiation.

Another favorable factor in reducing the body insult from radiation emanating from an implanted power source is the protraction of dose. It is well established that radiobiological changes associated with a given radiation dose are less pronounced when the dose is protracted or fractionated than when it is administered in a short-term, high-intensity incident. This is due to the repair of sublethal injury in the surviving cells and the proliferation of the surviving cells between fractions. Except for the dose at 10 rads/day, Brecher et al. (G. Brecher, E.E. Brambel and F.D. Brambel, Ann. N.Y. Acad. Sci., 114, 549-556, 1964). could find few pathological findings which could be related to radiation exposures in dogs, in spite of large accumulated doses. Doses of 10 rads/day (whole body) did show radiation-related hemotological effects such as hypoplasia of the bone marrow and lymph nodes and manifestations of hemorrhage. Pontifex and

Lamberton (A.H. Pontifex and L.F. Lamberton, Brit. J. Radiol., 33, 736-747, 1960) that 50 rads/day (whole body) continuous X irradiation over a period of 150 to 200 days was the highest dose rate at which steady state peripheral blood counts could be maintained in rats. The Mexican accident (Martinez et al, Rev. Med. Inst. Med., Seguro Social, 3, 14-68, 1964) is an example of response to protracted co-60 gamma irradiation. The one survivor (an adult male) had been exposed continuously to an estimated daily dose of 9.3 rads (maximum 16.2 rads) for 106 days, giving a minimal total dose of 985 rads (maximum 1640 rads). Note that all the foregoing examples involve whole body dose rates far in excess of those that would be delivered to an individual with an implanted power source fueled with Pu-238. Not only would the dose rates be much lower to the recipient of a radioisotope powered circulatory support device, but the significant irradiation would be to a small fraction of the total body.

Dr. Leonard Sagan of the U.S.A.E.C., Division of Biology and Medicine discussed the radiobiological data from clinical therapeutic irradiations and occupational exposures of radiologists and uranium miners at a joint AEC-NHI review meeting in Washington, D.C., on December 14, 1967. Dr. Sagan was of the opinion that the radiation hazard to the patient from "medical grade" plutonium is not likely to be a deterrent in applying radioisotope power sources for circulatory support devices.

Perhaps the most pertinent data comes from the joint NHI-AEC program for studying endogenous heat. A 16 watt, Pu-238 heat source fabricated by Mound Laboratory placed in a heat exchanger was implanted in the thoracic aorta of a female mongrel dog named "Plutina." The

purpose of this implant is to study the combined effects of radiation and additional endogenous heat. This dog made a rapid post operative recovery, and is in excellent condition after six months of implantation. A battery of blood factors have been checked periodically, and except for somewhat low albumin, all appear normal. This experiment is especially significant since the dose rates on the surface of the heat exchanger are essentially unattenuated by either geometry or shielding. The measured maximum dose rate on the heat exchanger surface is approximately 400 mrem/hr, about equally divided between neutron and gamma components. This dose rate is at least 5 times the maximum expected on the surface of the radioisotope fueled power source. In addition, the dog is smaller (approximately 90 pounds) than the usual human candidate for such a device. Promising as these limited results are, extensive chronic irradiation studies should be immediately initiated.

Based on the preponderance of the foregoing evidence, Thermo Electron concludes that the potential benefits of a radioisotope fueled circulatory support device outweigh the radiation hazard to the patient.

However, Dr. Sagan pointed out an additional radiation factor that must be evaluated. This factor is the radiation on the population (sometimes termed the "crowded elevator syndrome"). A typical background radiation level is 0.02 mr/hr. The Federal Radiation Council has recommended a maximum permissible dose rate to the general population of 0.06 mr/hr. The maximum surface dose rate on the surface of the dog implanted with the 16 watt, Pu-238 source is roughly 25 mrem/hr. Based on inverse square attenuation, the dose rate at a distance of one foot from the "hottest" surface of the dog should be less than 1 mrem/hr. This dose rate is far lower than the dose rate of

the surface of many luminous dial watch faces! Assuming that the host interacts no more than one hour a day with this intimacy to a single individual, the mean dose rate over a day would meet the recommendations of the Federal Radiation Council. Of course, in the recipient's family life, elemental precautions such as twin beds would be required. It could be safely assumed that an individual with an radioisotope fueled power source would be highly motivated to minimize radiation exposure to his family. Perhaps some occupational restrictions might be desirable, but these should not constitute a major inconvenience. Therefore, radiation exposure to the general population from implanted radioisotope power sources does not appear to be a deterrent to their application.

Another facet to the problem of population exposure is that of genetic risk to offspring. The individual with an implanted nuclear source will be subject to gonad or ovary doses well above those established for routine occupational hazards. The dose rate to the reproductive organs might be as high as 25 mrem/hr (corresponding to 220 rem/yr). Assuming that this dose rate were taken for 5 years, the total dose to the reproductive organs would be 1100 rem. The probability of mutation in offspring can be estimated from animal data. From Langham (W.H. Langham, Editor, Radiobiological Factors in Manned Space Flight, Publication 1487, National Academy of Sciences, Washington, D. C., 1967). --

$$\text{Mutation rate per rem/gene} \doteq 5 \times 10^{-8}$$

$$\text{Number of genes per haploid set} \doteq 10^4$$

$$\text{Probability of new mutation in first generation heterozygote} \doteq 5 \times 10^{-2}$$

Assuming a dose of 1100 rem, the probability of a new mutation in the immediate offspring would be approximately,

$$(5 \times 10^{-8}) (10^4) (5 \times 10^{-2}) (1.1 \times 10^3) = 2.8 \times 10^{-2}$$

Although this estimate should be taken with considerable reserve, it indicates that the genetic risk may be high enough that it would not be socially responsible for an individual using a nuclear assist device to become a parent. Sperm banks and/or adoption may constitute a partial solution to this problem. Of course, a large fraction of the population group who would be candidates for circulatory assist devices would be over the normal age for parenthood so that genetic risk becomes an academic consideration.

The primary radiation hazard problem has yet to be discussed. Historically, the primary hazard barrier to the large scale application of the radioisotope power sources (i. e., odd numbered SNAP units) has been the problem of source integrity under all combinations of highly unlikely, but still credible conditions. The simple fact that the radioisotope source can be thermally useful implies that even a small release of the fuel material is potentially quite hazardous.

Consider the following improbable, but possible, chain of events. Suppose the implantable power source malfunctions and soars the fuel temperature immediately before the airliner crashes on the edge of a major water supply. In addition, suppose the fuel is then subjected to a high temperature aircraft fire for half an hour before the wreckage explodes with overpressure and shoots shrapnel directly at the fuel capsule. Finally, suppose that the fuel capsule then drops into the

water reservoir. How can the fuel capsule be designed such that the probability of polluting the reservoir is vanishingly small?

Thermo Electron concludes that the primary radiation obstacle to the large-scale deployment of radioisotope powered circulatory support devices is that of ensuring fuel containment under all credible conditions.

C. BLOOD -MATERIALS INTERFACE

The blood-materials interactions which give rise to clotting, hemolysis and protein denaturization are not well understood. These problems are so complex that advances in basic blood-materials mechanisms are unlikely to yield any practical solution within the next few years. For example, over a dozen biochemicals have been identified in the scheme of protein interactions leading to clot formation. The surface potential of prosthetic materials has been shown to be an important factor in clotting. In addition, hydrodynamic flow patterns probably are important. Up to now, the chemical, electrical and fluid flow considerations have been studied separately. Therefore, a fundamental grasp of the clotting problem appears unlikely in the near future.

Fortunately, a basic understanding of the blood-materials interaction may not be necessary. As valuable and desirable as fundamental knowledge is, technological history is replete with examples of practical developments which took place with only a limited fund of basic information. For example, steam engines powered the industrial revolution long before their fundamental thermodynamics were understood and galena crystals detected radio waves decades before their rectification mechanisms could be explained.

A salient fact is that normal intima does an excellent job of repelling thrombus. Therefore the most promising avenue for non-thermogenic prosthetic surfaces is to grow a thin layer tissue on the blood pump/heat exchanger surfaces. Progress in this area has been made by DeBakey, Monsanto and the TECO-CHMC team. Some of this work is described in Chapter IV.

There is a wide spread of opinion on the problem of hemolysis. Hemolysis can apparently be due to both mechanical stress and surface properties of the prosthetic wall. With design refinement, an artificial sac pump should not give appreciably higher mechanical stress to the blood than the natural heart. As with the clotting problem, a pseudo-endothelium surface may well be compatible with respect to hemolysis. Preliminary data with autologous linings on flocked Dacron surfaces of the TECO-CHMC assist pumps indicate that the hemolysis level drops to near normal after the pseudo-endothelium layer is formed.

D. ADDITIONAL ENDOGENOUS HEAT

Since no artificial assist device is 100 percent efficient, additional endogenous heat loads will have to be dissipated by the body. Radioisotope-fueled circulatory support systems add an additional heat load to the body due to the reject heat of the thermodynamic converter. Several ways of dissipating this additional heat load have been proposed:

Air cooling in the lungs and larynx.

Liquid cooling of the bladder.

Conduction through the skin and subcutaneous layer.

Blood cooling in a major artery with distribution and dissipation throughout the body.

Of these alternatives, only heat exchange with the blood appears to be feasible. These alternatives are discussed in more detail in Thermo Electron Report No. TE 36-66, "Characterization of Biomedical and Technological Problems," Volume 2 (1965).

The two primary problems with blood heat exchange are reducing the total heat load and transfer of the heat across a blood-compatible interface without undue damage to the blood. The joint NHI/AEC sponsored studies of additional endogenous heat at TECO-CHMC suggest that an adult human could be expected to dissipate up to 50 watts without distress. Although a maximum of 50 watts heat dissipation implies an efficient conversion from thermal to mechanical energy, it should be within the development capability of modern technology.

The rate of heat production by the cells of the body increases by approximately 70% for every five or six degrees Centigrade increase in body temperature. For example, when the body temperature is raised from its normal value of 37°C to 42.5°C , the basal metabolic heat production increases from a normal value of 40 calories/hr/meter² to a value of approximately 68 calories/hr/meter.² The rate of heat production, therefore, rises approximately 13%/°C. At such high levels of body temperature, it becomes progressively more difficult for the normal body mechanisms to dissipate heat at the rate at which it is being produced. As a result, high body temperatures can become unstable producing still higher body temperatures. This instability occurs at about 42°C .

The various methods by which heat is lost from the body include radiation, convection and evaporation. These physiologic mechanisms are effective at normal or slightly increased body temperatures, varying

their individual contributions according to environmental conditions. However, these mechanisms become less effective as body temperature approaches 42°C. Although there is evidence that temperature regulation is seriously impaired in the range beyond 41°C (Battelle Northwest has found that minipigs show thermal instability at heat inputs of 80 watts), the normal physiologic mechanisms (based upon hypothalamic activity and the less effective peripheral, skin-control mechanisms) are capable of dissipating heat from the body and controlling core temperatures in the range between 38°C and 41°C. This is accomplished by both a change in the rate of heat production and a change in the rate of heat loss, depending upon environmental conditions.

The problem of transferring heat across a blood compatible interface without undue damage should be possible. A heat transfer analysis of this problem is given in Chapter IV. Experiments with mongrel dogs (TECO-CHMC) and pigs (BNW) have indicated that the primary problems are related to the patency of the intracorporeal heat exchanger prosthesis rather than the effects of the heat itself. Problems of clotting, surgical trauma and percutaneous lead infection have masked much of the data.

Of course, use of radioisotope heat eliminates the problem of lead infection. It is also likely that refined surgical technique will reduce operative mortality level to that of other major cardiovascular surgery. The problems of clotting and blood damage have been discussed previously. If one assumes that the problem of blood materials compatibility is resolved for the blood pump, then it will also be resolved for the heat exchanger. The remaining problem is one of temperature-rise in the boundary layer of the blood. In vitro experiments using

prosthetic heat exchangers with 16 and 24 watt Plutonium-238 heat sources are analyzed in Appendix D. These data demonstrate that a heat flux of one watt per square centimeter can be attained at flow rates typical of large arteries without exceeding the guideline value of 4°C for the maximum rise in blood temperature in the boundary layer.

It is pertinent that a large mongrel dog, Plutina, has survived for six months in excellent condition with a heat exchanger (16 watt Plutonium-238 heat source) sutured into her thoracic aorta. Another mongrel dog, Discard, lived with an electrical heat exchanger for nine months before dying of infection.

All the dogs with radioisotope heat exchangers exhibited the same type of post operative rectal temperature history. Typically, rectal temperature climbs to about 2°C above normal within 12 hours after the implant. It then drops back to normal (within normal temperature fluctuations) within a day where it remains unless other complications (such as infections) take place. Although cooling blankets have been used on occasion, usually no special measures are taken to modify the dog's temperature response.

The foregoing discussion gives a basis of optimism for heat rejection to the blood, providing the blood-materials interface problem is solved. However, a separate heat exchanger requires a separate major operation to implant it. A more optimal solution would be to use the blood pump(s) artificial ventricle surface area as the heat exchanger surface. This has the advantage of requiring no additional prosthesis, or additional prosthetic interface with the blood. It also simplifies the surgical implantation and does not require the size, shape, location and length of the arterial and venous connections to

deviate from those of the natural heart. Furthermore, if the blood pumps are operated or actuated hydraulically, then the hydraulic fluid can be used as a heat transfer fluid between the power source and blood pumps. In addition the bulk blood temperature are lower because the heat exchanger (i. e., blood pump) is placed in the highest possible blood flow. The pumping action can also be expected to promote better blood mixing, thus reducing the peak temperature rise of the blood. Obviously, the weight and volume of a separate heat exchanger are eliminated.

Although the present data on additional endogenous heat are limited the results have been generally encouraging. A complete understanding of this problem requires investigation of:

- (1) the maximum heat load into the body
- (2) the maximum heat flux density into tissue and blood
- (3) the maximum bulk temperature rise of blood
- (4) the maximum rise of the blood temperature in the boundary layer
- (5) the effects of normal environmental variations on chronically induced hyperthermia.

Certainly additional work will be desirable to better characterize the problems of additional endogenous heat.

E. BLOOD PUMP CONTROL

The primary requisite of the circulatory support system is that it maintain an adequate cardiac output (typically, 5 liter/minute). If the power source drives an assist pump, this blood pump should respond to changes in stroke volume and heart rate. In addition, the circulatory

support system should operate over a range of systemic pressures. A total heart replacement must sense body circulatory requirements (probably venous return pressure) and balance pulmonary and systemic flows at different pressure levels.

Although fundamental physiological studies and analytic models of the cardiovascular system have been of value, present medical knowledge cannot identify the necessary and sufficient conditions for controlling an implantable circulatory support device. It is unlikely that basic biological studies can provide the required control information for early circulatory support systems. Rather, the necessary and sufficient control factors will more likely be empirically established from in vivo experimentation with blood pumps. Recent experiments by Thermo Electron - Children's Hospital Medical Center team with a left ventricle to aorta assist pump have indicated that a relatively simple counterpulse control system which can follow the stroke volume and beat rate of the natural heart may be adequate. Admittedly, the experimental animals have been healthy so that conclusions as to the behavior of animals with diseased hearts must be tentative. However, the pumping action actuated by the relatively simple counterpulse control system (pneumatic drive ECG trigger) has been well tolerated by the calves.

The recent human heart transplants also give a basis of optimism on total replacement hearts. The fact that the transplanted heart pumps adequately without neural communication with the brain is quite interesting. One suspects that the blood pump control problems may be less stringent than originally expected. The cardiovascular system may accommodate itself to the artificial pump over a wide range of conditions.

Controls efficiency (both in engine and blood pump) is a factor that has been largely ignored to date. Most studies of blood pump controls have been restricted to their physiological function. However, a large fraction of the power source output could easily be consumed in dissipative processes such as throttling, leakage and bias. The extent of the blood pump controls loss depends largely on the sophistication of the control. Since the blood pump control functions are not yet fully established, the control efficiency is difficult to estimate. Assuming that (for an assist pump) the circulatory support system need only respond to changes in stroke volume and heart rate, a blood pump control system operating off a hydraulic reservoir (see Appendix G) should be capable of an efficiency of 67 percent.

The engine control and blood pump controls problems are interdependent. The engine control problem is more difficult if no short time (order of seconds) intermediate energy storage (e.g., hydraulic accumulator) is used.

Although not ideal, early implantable circulatory support systems may be designed to meet systemic pressures only slightly higher than normal. Compression of the systemic pressure range permits a simple throttle control to be used without an excessive efficiency loss in the control unit. Normally, such a circulatory support system would stall under some emotional and activity stresses. Of course, such stalling would distress the recipient of such a device, but the stall would lower the systemic pressure and permit the artificial pump action to begin again. Such would be the penalty of simple efficient control technique unless the body would learn to respond with only small increases in systemic pressure.

Information on physiological control requirements, particularly for pathologic states, is still inadequate. Immediate effort is required on establishing minimum control requirements so that the implantable power source can be optimized as a system.

F. SURGICAL IMPLANTATION

In order to promote rehabilitation of the patient, the power supply for the circulatory support system is to be implanted completely within the body. Ideally, it is desirable to have the power supply as an integral part of the blood pump(s) or at least in close proximity to them. However, the blood pumps must be in juxtaposition to the heart and major vessels, which are in the center of the thoracic cavity behind the sternum. Therefore, one would like to locate the blood pumps and power supply in the thoracic cavity. Unfortunately, the remainder of the thoracic cavity is occupied by the lobes of the left and right lungs. Dr. DeBakey's group in Houston has encountered extreme difficulty in implanting their assist device (without a power supply) in the thoracic cavity of both animals and human patients. The problem was so difficult that they have now exteriorized their pump.

Over the past year, the CHMC-TECO group has also devoted a major effort to the implantation of assist devices in the thoracic cavity of animals. If the pump design is optimized with regard to the shape and location of the pump and its connections, it would just fit in the thoracic cavity. This pump is described in Chapter IV. Chronic animal tests are now in progress with human-size pumps implanted in the thoracic cavity of calves. In the case of the total heart replacement, the blood pumps would occupy the space previously occupied by the natural heart.

While it is possible to locate the blood pumps in the thoracic cavity, it appears to be extremely difficult, if not impossible, to locate the power supply in the thoracic cavity. It has been suggested by others that one of the five lobes of the lung be excised to make room for the blood pump or power supply. CHMC-TECO have rejected this approach since:

- 1) Many patients treated with circulatory support systems will also have pathologic pulmonary systems and need all the vital capacity possible.
- 2) The removal of a lobe of the lung is a major surgical procedure in itself.
- 3) If the unit requires replacement, one should avoid a second thoracic surgical procedure.
- 4) There are better alternatives such as locating the power source within the abdominal cavity.

A possible location for the power supply is under the left hemi diaphragm, as shown in Figure VI-1. The unit would lie in a horizontal position adjacent to the stomach and would be supported by a sling attached to the diaphragm. A relatively short interconnecting hydraulic cable results, thus minimizing unnecessary fluid flow losses. At the time of implantation, an incision would be made in the diaphragm and the unit lowered through the diaphragm into position from the thoracic cavity. The incision in the diaphragm would be closed by sutures, ensuring a good seal around the hydraulic cable. When the patient's condition warranted, the thoracic cavity would be closed in the normal manner.

Figure VI-1. Location of Components within the Body

The development of medical grade plutonium fuels, now being supported by the AEC, has resulted in lower dose rates from impurities than previously anticipated. Further reduction of the Pu-236 content to reduce the latent shielding problem from the build-up of Tl-208 would be very desirable.

The development of the thermal energy storage (TES) devices now supported by NHI is supplying guidelines for the design of TES subsystems. This work should continue. The development of multi-layer thermal radiation shields to reduce thermal losses from radio-isotope heat sources, presently being supported by the AEC, is quite necessary for the development of an implantable power source. This work should also be continued.

In summary, there are problem areas requiring further effort. Most of these areas are presently being investigated and these investigations should continue. The entire area of the thermodynamic converter is only now beginning to be investigated, and this area must be expanded. The coupling and control system between the blood pump and power source needs further investigation. Finally, all of the results from the diverse medical programs must be integrated into an optimized system design.

CHAPTER VII

COMPARATIVE ANALYSIS

In an age of nuclear reactors and supersonic aircraft, the choice of the steam engine as the converter from thermal to mechanical energy deserves some explanation. Unfortunately, the comparative analysis given in this chapter is not fully objective. This lack of objectivity is not the result of base motives, but a lack of the same familiarity with gas cycle engines as the evaluators possessed with steam engines.

It would not be fair to judge the merits of the Stirling engine on the basis of a Thermo Electron design since the design ingenuity of Stirling engine advocates may have resolved problems to which Thermo Electron sees no practical solution. However, the basic cycle thermodynamics are discussed with particular emphasis on how practical losses degrade their behavior.

Historically, the steam engine was the prime mover for the industrial revolution and has powered pumps, factories, ships, cars, trains and electric generators. Admittedly, the relationship between steam cars and heart pumps is somewhat circuitous. However, the fact that although hot air engines have been available for 150 years (the Stirling engine, complete with regenerator, was patented in 1817), but have yet to seriously compete with steam engines as prime movers is quite relevant to the choice of the thermodynamic cycle to be used in the power source for an implantable circulatory support system. The fundamental reason for the practical success of the steam engine is its insensitivity to frictional, thermal and

leakage losses. These losses become relatively more important as the engine size decreases.

A. GENERAL

The efficiency of the thermal to mechanical conversion system is primarily important and should be as high as possible in order to minimize the heat rejection within the body, to minimize the weight and size of the power unit, and to reduce the fuel inventory (hence, fuel cost and radiation).

Figure VII-1 shows block diagrams for the two basic approaches for utilizing the power produced by a radioisotope source to drive an artificial heart. In the first approach, the thermal power of the isotope is converted into electrical power using a thermoelectric or a thermionic generator. The electrical power is then converted into mechanical power by an electric motor, solenoid or piezoelectric device. This mechanical power in turn is coupled to the blood pumps. The overall system efficiency is then the product of the efficiency of each of these intermediate stages or

$$\eta_A = \eta_{\text{Radioisotope (R)}} \eta_{\text{Thermal to Elec. Conv. (E-C)}} \eta_{\text{Elec. to Mech. Conv. (E-M)}} \eta_{\text{Coupling (c)}} \eta_{\text{Blood Pumps (B. P.)}}$$

In the second of the two approaches, the thermal power of the radioisotope is converted directly into mechanical power using a reciprocating engine which is coupled to the blood pumps. Again, the overall efficiency is the product of the efficiency of each of these intermediate stages, or

Figure VII-1. Block Diagram showing Major Components of Radioisotope Powered Artificial Hearts

$$\eta_B = \eta_{\text{Radioisotope (R)}} \eta_{\text{Thermal to Mechanical (T-M)}} \eta_{\text{Coupling (C)}} \eta_{\text{Blood Pumps (B. P.)}}$$

In comparing the overall efficiency of these two approaches, one notes that they both have several components in common. Assuming the efficiency of these components which are common to both approaches is equivalent, one notes that in order for the overall efficiency of the two approaches to be equal, the following equality would apply:

$$\eta_{E. C.} \eta_{E. M.} = \eta_{T. M.}$$

That is, the product of the efficiency of the thermal-to-electrical converter and that of the electrical-to-mechanical converter must equal the efficiency of the thermal-to-mechanical converter. If the latter efficiency is greater than the former, then approach B will have a greater efficiency than approach A, and vice versa.

B. SYSTEM WITH ELECTRICAL CONVERSION DEVICE

At the present time, the most practical conversion mechanism is an electric motor, hydraulic pump and control system. This approach probably results in a maximum efficiency of 50%. Therefore, in order to deliver an average power of 3 watts (chosen as a basis of comparison only) to the blood, 6 watts of electrical power is required into the electric motor. The electric motor, hydraulic pump, and controls would weigh approximately 1-1/2 to 2 lbs.

There are basically two methods of converting thermal energy directly to electrical energy: thermoelectricity and thermionics.

Although both methods are capable of high reliability, their efficiencies are marginal for the present application.

Although known in principle for over a century, thermoelectricity became a practical conversion scheme in the latter half of the 1950's, along with many other semi-conductor and solid-state developments. The first isotope-thermoelectric generator was demonstrated in January 1959. This generator had a power output of 3 watts, weighed approximately 6 lb, and was 3% efficient with air cooling and 5% efficient with a water-cooled cold junction. This unit had a specific power of 0.5 watt per lb. By 1963, the specific power had been increased to one watt per lb and the efficiency to approximately 5% at the 25-watt level. Today, using a radioisotope-powered thermoelectric generator, one should be able to achieve an overall efficiency of 10% and a specific power of 1-1/2 watts per lb at a power level of 6 watts. This is exclusive of shielding and is based upon blood cooling of the cold junction. When one adds in the power-flattening and voltage-regulation equipment, the efficiency then becomes 9% at a specific power of 1 watt per lb. The system would then require 67 watts thermal power input at the end of the fueling life, and somewhat more at the time of implantation. It would weigh approximately six pounds including the fuel, generator, and power-conditioning equipment, but not including the radiation shielding, which would depend on the radioisotope selected.

Thermionics as a practical means of generating electric power for thermal sources was introduced in the late 1950 s. The potential efficiency of thermionics exceeds that of thermoelectrics in large units, but the high temperature problems were more difficult.

From mid-1961 to 1964, significant progress was made on the development of practical hardware generators, and the performance capabilities of thermionics advanced rapidly. Even though improvements have been made, still further major improvements are possible, based upon the results of current research.

Unfortunately, even though rapid progress can be expected in thermionics, it is not likely to occur in the areas which would make it more attractive as a power source for circulatory support systems. Thermionics is a high temperature phenomenon, which limits the choice of heat sources, placing a premium on high power density. This is not likely to change significantly, since it is a characteristic of the thermionic process. Also, most of the advances being made in thermionics can best be realized by retaining the same size of unit and increasing the power level. The 6- to 10-watt level is already on the low end of the spectrum for thermionics. Thermionics cannot take advantage of the low heat-sink temperatures possible with blood cooling in the form of increased efficiency. This again is a basic characteristic of thermionics and is not likely to change. One of the principal advantages of radioisotope thermionics for most applications (e. g. , space power) is the high rejection capabilities. This is no advantage for the circulatory support system application. Thus the direct conversion from thermal energy would be better accomplished through the use of a thermoelectric generator. This electrical energy must then be converted to mechanical power.

Therefore, the produce of $\eta_{E. C.}$ and $\eta_{E. M.}$ for this approach is:

$$\eta_{E. C.} \eta_{E. M.} = (9\%)(50\%) = 4\frac{1}{2}\%$$

The total weight breakdown of this approach, less shielding is given in Table VII-1.

TABLE VII-1
COMPONENT WEIGHTS OF ELECTRICAL CONVERSION SYSTEM

Thermoelectric Generator and Power Conditioning Equipment	6 lbs
Electric Motor, Hydraulic Pump and Controls	1-1/2 lbs
Blood Pumps	<u>2 lbs</u>
Total*	<u>9-1/2 lbs</u>
Minimum heat dissipation to the blood	64 watts
Maximum heat dissipation (for one-half life) to blood	131 watts

* Does not include a battery which would be required for peak power.

Thermoelectric generators would be quite interesting if the conversion efficiency were doubled. Such a technology advance appears unlikely. Therefore thermoelectric generators would appear to be too heavy and too inefficient for application to circulatory support systems.

C. SYSTEMS WITH DIRECT MECHANICAL CONVERSION DEVICE

There are several types of heat engines capable of converting the thermal power of the radioisotope into mechanical power to drive the blood pumps. In order to be capable of being implanted in the

body, the system must be closed-cycle. The closed cycles which can be considered for this application fall into two categories: gas cycles and liquid-vapor cycles.

For an ideal reversible heat engine operating between two temperature limits, the maximum attainable thermodynamic efficiency is that of the Carnot cycle. As shown in Figure VII-2, the Carnot cycle consists of two constant-temperature processes connected by two isentropic processes. Heat is exchanged with the surroundings only during the constant-temperature processes 1-2 and 3-4. The heat input from 3-4 is given by:

$$Q_{34} = T_H (S_4 - S_3)$$

The heat rejected to the low-temperature sink during process 1-2 is given by:

$$Q_{12} = T_c (S_1 - S_2) = T_c (S_4 - S_3)$$

The net work can be obtained from the first law of thermodynamics:

$$W_{\text{net}} = Q_{34} - Q_{12} = (T_H - T_c) (S_4 - S_3)$$

The cycle efficiency is:

$$\eta_{\text{Carnot}} = \frac{T_H - T_c}{T_H}$$

A Carnot cycle using water is analyzed in Chapter III. It is not practical because of the difficulty in approaching the reversible adiabatic compression of a two-phase substance.

Figure VII-2. Carnot Cycle

1. Gas Cycles for Direct Mechanical Conversion

The ideal gas cycles which can be considered for this application are the Brayton, Ericsson, and Stirling cycles. These cycles are shown in Figure VII-3.

The ideal Brayton cycle consists of isentropic compression and expansion processes connected by two constant-pressure heating and cooling processes. In the ideal Ericsson cycle, the isentropic processes of the Brayton cycle are replaced by constant-temperature processes. In the ideal Stirling cycle, two constant-volume processes connect constant-temperature expansion and compression processes.

A regenerator is normally used in all these cycles to increase the cycle efficiency by exchanging heat between the constant-pressure processes in the Brayton and Ericsson cycles and between the constant-volume processes in the Stirling cycle. The regenerator reduces the required heat input and heat rejection for the cycle without decreasing the net work output. In both the ideal Ericsson and ideal Stirling cycles, a perfect regenerator allows all the heat input to take place at a constant temperature, T_H , and all the heat rejection to take place at a constant temperature, T_C . The ideal Ericsson and Stirling cycles are therefore reversible heat engines operating between two temperature limits. The efficiency of these ideal cycles is then the same as that of a Carnot cycle operating between the same temperature limits.

$$\eta_{\text{ideal Stirling}} = \eta_{\text{ideal Ericsson}} = \frac{T_H - T_C}{T_H}$$

Figure VII-3. Gas Cycles

In an ideal Brayton cycle, the heat input and heat rejection both occur during constant-pressure processes. Since the heat input does not take place at constant temperature, this cycle is not reversible when operating between two temperature limits. The efficiency of the ideal Brayton cycle is therefore less than that of either the ideal Ericsson or the ideal Stirling cycle. However, the efficiency of the ideal Brayton cycle approaches the Carnot cycle efficiency as the cycle pressure ratio decreases. This occurs because the heat input and rejection processes approach constant-temperature processes at low pressure ratios. The net work output per pound of working fluid is lower, however, at low pressure ratio.

In all these gas cycles, the compression work is a large fraction of the expansion work in the cycle. For example, when these ideal cycles operate between 1250°F and 100°F, the ratio of the compression work to expansion work is approximately 50% for the Brayton cycle and 30% for the Ericsson and Stirling cycles. Thus, a small inefficiency in the compression process can affect the power output and cycle efficiency tremendously. This is a distinct disadvantage at lower power levels, where a small change in the frictional loss of a bearing or seal in the compressor or a small change in the leakage, due to wear, could alter the cycle output drastically.

In the Brayton cycle, the isentropic expansion work can be expressed as:

$$W_{\text{expansion isentropic}} = \frac{C_p T_H (P_R^{\frac{K-1}{K}} - 1)}{P_R^{\frac{K-1}{K}}}$$

and the isentropic compression work as

$$W_{\text{compression}} = C_p T_c \left(P_R^{\frac{K-1}{K}} - 1 \right)$$

isentropic

The efficiency of the expansion, η_E , and the efficiency of the compression, η_C , can be introduced and combined with the above relationships to give:

$$W_{\text{net}} = \left[\frac{T_H \eta_E}{P_R^{\frac{K-1}{K}}} - \frac{T_c}{\eta_C} \right] C_p \left(P_R^{\frac{K-1}{K}} - 1 \right)$$

The ideal required heat input is equal to the ideal isentropic expansion work.

$$Q_{\text{in}} = \frac{C_p T_H \left(P_R^{\frac{K-1}{K}} - 1 \right)}{P_R^{\frac{K-1}{K}}}$$

This expression is general in that it is correct if the inefficiencies are due to mechanical losses, but not exact for thermal or fluid losses. Therefore, the cycle efficiency is

$$\eta_{\text{T.M.}} = \eta_{\text{Brayton cycle}} = \eta_E - \frac{T_c P_R^{\frac{K-1}{K}}}{T_H \eta_C}$$

In the Ericsson and Stirling cycles the work of expansion is:

$$W_{\text{expansion}} = \eta_E T_H \Delta S$$

and the work of compression is:

$$W_{\text{compression}} = \frac{T_c \Delta S}{\eta_C}$$

and the heat input is:

$$Q_{in} = T_H \Delta S$$

Therefore the efficiency of the cycle becomes

$$\eta_{T.M. \text{ cycle}} = \eta_{\text{Ericsson cycle}} = \eta_{\text{Stirling cycle}} = \eta_E - \frac{T_c}{T_H \eta_C}$$

For the Brayton cycle, a reasonable value for the parameter $P_R^{\frac{K-1}{K}}$ would be approximately 1.5 for typical pressure ratios and specific heat ratios of appropriate gases. Using this value for comparison purposes, the effect of the expansion efficiency, η_E , and compression efficiency, η_C , are presented in Table VII-2 for these cycles. For simplification η_E will be equated to η_C . The temperatures chosen are $T_H = 1250^\circ\text{F}$, $T_C = 100^\circ\text{F}$.

TABLE VII-2
REGENERATIVE GAS CYCLE EFFICIENCIES
FOR VARIOUS EXPANSION AND COMPRESSION EFFICIENCIES

Cycle Efficiency (η_{TM})	$\eta_C = 1.00$	$\eta_C = \eta_E$		$\eta_C = 0.57$
		$\eta_C = 0.80$	$\eta_C = 0.70$	
Brayton efficiency % Carnot	0.51 0.76	0.18 0.27	0 0	No net output
Ericsson and Stirling efficiency % Carnot	0.67 1.00	0.39 0.58	0.23 0.34	0 0

It can readily be seen that relatively high expansion and compression efficiencies are required for these cycles to produce any net work. At low power levels reciprocating engines and compressors have much higher efficiency than turbomachinery. It is extremely unlikely that turbomachinery could obtain the required efficiency, even at very high speeds. Since the Brayton cycle would require that these efficiencies exceed 80% in order to work, the Brayton cycle can be eliminated from further consideration, even with reciprocating components. The other inefficiencies in a Brayton cycle would make it extremely difficult to produce an acceptable cycle efficiency at these low power levels.

At these power levels, the requirement of efficiencies of expansion and compression in excess of 70% for the Ericsson and Stirling cycles dictates the use of reciprocating components. The Stirling cycle has a higher work output per displacement volume of the working fluid than the Ericsson cycle. In addition, the Stirling cycle has been more highly developed as a practical cycle. Therefore, it can be concluded that the Stirling cycle is the most competitive for this application.

a. Stirling Cycle Fundamentals

The Stirling cycle is based on the compression of a given quantity of gas at low temperature and its expansion at high temperature. The heat is supplied to the gas from the outside through a wall. Because of the high heat capacity of the wall, it would be grossly inefficient to heat and cool the gas simply by rapid heating and cooling of the wall. However, Stirling realized that the gas

temperature could be made to change periodically by using a "displacer" or "transfer" piston to transfer the gas back and forth between two spaces, one at a fixed high temperature and the other at a fixed low temperature (see Figure VII-4). If the displacer piston in Figure VII-4 is raised, the gas will flow from the hot space via the heater and cooler ducts into the cold space. If now the displacer piston is moved downwards the gas will return to the hot space along the same path. During the transfer stroke the gas gives up a large quantity of heat and an equal quantity of heat is taken up during the second stroke. The regenerator shown in Figure VII-4 is inserted between the heater duct and cooler duct in order to prevent unnecessary wastage of this heat. The regenerator is filled with porous material to which the hot gas yields heat before entering the cooler. When the gas streams back, the regenerator takes up the stored heat again prior to its entry into the heater.

The displacer system works in combination with a power piston which compresses the gas while it is in the cold space and allows it to expand while in the hot space. Since compression takes place at a lower temperature than expansion, a net surplus of work results.

The four phases through which the system passes during the ideal cycle are shown in Figure VII-5. Note that the idealized cycle corresponds to a discontinuous movement of the power and displacer piston. In all practical versions of the Stirling engine thus far built the movements of the power and displacer pistons have been continuous, not discontinuous. The continuous movements may be obtained with the aid of a crank and connecting rod mechanism. Although sharp transitions between the four phases can no longer be distinguished, this

Figure VII-4. Displacer-Piston System (Movement of piston up and down causes the gas to be transferred back and forth between the hot and cold spaces, via heater, regenerator and cooler,)

Figure VII-5. Diagrams to Illustrate the Stirling Cycle.

- I All the gas is in the cold space.
- II The power piston has compressed the gas while it is at low temperature.
- III The displacer piston has pushed the gas through the cooler, regenerator and heater into the hot space.
- IV The hot gas has expanded and the power and displacer pistons have together returned to the lowest position. While the power piston remains there, the displacer piston will now push the gas through the heater, regenerator and cooler into the cold space, thus returning the system back to configuration I.

will neither alter the principle of the cycle nor detract from its efficiency. The essential condition for obtaining a surplus of work is that the volume variation of the hot space should have a phase lead with respect to that of the cold space.

Under the ideal conditions of isothermal behavior in cold and hot spaces and a 100% efficient regenerator, the Stirling engine becomes a reversible cyclic process in which heat is supplied at a single temperature and rejected at a single temperature. Under these conditions, a corollary to the second law of thermodynamics states that the efficiency is that of the Carnot cycle.

b Stirling Engine Configurations

All existing designs of Stirling engines may be broadly classified into two groups: (1) two loaded piston machines, and (2) displacement piston machines. In the latter group a further subdivision can be made between machines in which the piston and displacer operate in a single cylinder and those in which separate cylinders are provided for the displacer and the piston. In each case these machines consist of two variable volume working spaces at different temperatures, connected by an open duct which contains the regenerator and the heat exchangers. An example of each of these three arrangements is shown in Figure VII-6. The designation into alpha, beta and gamma configurations correspond to original designs by Rider, Stirling and Robertson, respectively. All these arrangements are thermodynamically equivalent.

Kirkly¹ has compared the different configurations using a dimensionless quantity referred to as the power parameter which takes into

account the total swept volume of the machine and the maximum cycle pressure. He found that for any temperature ratio the beta configuration has the highest value of the power parameter and the gamma engine has the lowest. This is true for all values of dead space ratios. The superiority of the beta engine is due to the economy with which the total swept volume is used. This is the result of arranging the strokes of the power and displacement pistons so that they overlap. The inferiority of the gamma engine is due to splitting the cold space between two cylinders. Since the minimum volume of the cold space is greater than zero the pressure variation during the cycle is less than in alpha and beta engines. A reduction in power is the result.

In spite of these considerations, it is not clear which design will ultimately be used most widely. At the present time, Philips and General Motors have developed the beta configuration (in conjunction with the rhombic drive and roll sock seal) to the highest degree.

Both McDonnell-Douglas and Aerojet-General have devised modified versions of the gamma (i. e. , Robertson) configuration in which the displacement piston motion is coupled to either a mechanical or gas spring. These designs have the advantage that the number of engine parts is reduced.

The regenerator of the McDonnell-Douglas engine is incorporated into the displacement piston. A plug on the displacement piston shaft also factors into this design.

ALPHA (RIDER) CONFIGURATION

BETA (STIRLING) CONFIGURATION

GAMMA (ROBERTSON) CONFIGURATION

Figure VII-6. Stirling Engine Configurations.
H (heater), R (regenerator), C (cooler)
D (displacement piston), P (power piston)

c. Finkelstein Analysis

The classical isothermal theory of Schmidt² is simple, elegant and over-optimistic since it does not account for any losses and assumes that compression and expansion occur at constant temperature only. Although the Schmidt analysis results in explicit algebraic equations, the assumptions imply infinitely large heat transfer through the cylinder walls. The Schmidt analysis always gives results much greater than actual measurement. Other investigators such as Kirkley¹, Walker³ and Karavansky⁴ have treated the problem with more refinement, but never realistically.

Even though it treats the regenerator pretty much as a "lump," the analysis by Finkelstein⁵ on which the program described herein is based, is the only published treatment which would appear to be general and reasonably realistic. Finkelstein removes the major assumption of isothermal processes and, in addition, allows for imperfect regeneration, aerodynamic flow losses, unsteady state speed and unsteady temperature conditions. Heat transfer in the heat exchangers, as well as in the main working spaces is limited by arbitrary heat transfer coefficients. Clearance at the piston positions corresponding to top dead center in the two main working spaces is considered. An important feature of the Finkelstein development is that the volume variations of the expansion and the compression space need not be sinusoidal but can follow any arbitrary law with respect to crank angle. He describes the cycle processes in a system by a set of eleven simultaneous equations, of which six are nonlinear interdependent differential equations.

Obviously, Finkelstein's analysis requires a computer. Huffman and Bienert⁶ have solved Finkelstein's set of eleven equations using an extension of the fourth-order Runge-Kutta integration on a recursive basis. The integration steps are carried out over small increments of crank angle. After the normalized temperatures and pressures are calculated, the crank angle is increased and the output values of the previous integration step are used as initial values for the next integration step. The crank angle is stepped across 360 degrees in this fashion.

The computer results indicate that the Stirling cycle efficiency is sensitive to the heat transfer coefficients in the expansion space, compression space, heater and cooler. Since the uncertainties in heat transfer coefficients and aerodynamic friction factors span a wide performance range, the program has not proven of practical value.

The Stirling engine also has a problem of long term gas containment. Hydrogen and helium would be the best gases to use because of their good heat transfer properties. Both gases readily diffuse through most materials. Metal bellows could contain helium, but Thermo Electron's experience with metal bellows indicates that their reliability is highly questionable.

2. Liquid-Vapor Cycles for Direct Mechanical Conversion

The Rankine cycle is employed when the thermodynamic advantages of using a fluid in both its liquid and vapor phases are desired. An important advantage of the Rankine cycle for this application is the greatly reduced ratio of compression work to expansion work that

results from compressing a liquid rather than a gas. In an ideal Rankine cycle, the ratio of compression work to expansion work can be less than 1.0% when water is the working fluid. This compares with the 30% ratio in a Stirling cycle. Thus a Rankine cycle can be essentially independent of the efficiency of the compression process. Also, in the case of the Rankine cycle, the condenser and boiler both involve either a condensing or boiling fluid, thus increasing the overall heat transfer coefficient. In the case of the Stirling or Brayton cycles, the two heat exchangers are gas heat exchangers, with a much lower heat transfer coefficient, thus requiring much greater heat transfer area. This, in turn, means increased weight and size.

The ideal Rankine cycle is illustrated in Figure VII-7. This cycle consists of isentropic expansion and compression processes connected by constant-pressure heating and cooling processes. The compression work in this cycle is greatly reduced because the vapor is condensed and pumped as a liquid.

The reason that the compression work is so much lower in a Rankine cycle than in a gas cycle can be seen by looking at the equation for the compression work.

$$W_{\text{compression Rankine}} = v(\Delta P)$$

Since the specific volume (v) of the liquid is so much lower than that of the vapor, the ratio of compression work to expansion work will be low. The selection of the working fluid for the Rankine cycle is discussed in Chapter III. Water, organic fluids and liquid metals are evaluated for the implantable power source application.

Figure VII-7. Ideal Rankine Cycle

For an ideal steam engine operating between 1250°F and 100°F with a high pressure of 1200 psia on a Rankine cycle, as shown in Figure VII-7, the isentropic work output is:

$$\begin{aligned} W_{\text{expansion isentropic}} &= h_1 - h_2 \\ &= 1641 - 957 = 684 \text{ Btu/lb} \end{aligned}$$

The pump work is:

$$\begin{aligned} W_{\text{pump isentropic}} &= h_4 - h_3 = v_3 (P_4 - P_3) = 0.185 (0.0161)(1200-1) \\ &= 3.6 \text{ Btu/lb} \end{aligned}$$

The heat input can be calculated from:

$$Q_{\text{in}} = h_1 - h_4 = 1641 - 71.6 = 1569 \text{ Btu/lb}$$

Therefore the ideal steam cycle efficiency is:

$$\begin{aligned} \eta_{\text{ideal steam cycle}} &= \frac{W_{\text{expansion isentropic}} - W_{\text{pump isentropic}}}{Q_{\text{in}}} \\ &= \frac{684 - 3.6}{1569} = 0.434 \end{aligned}$$

The effect of inefficiencies in the expansion process and pumping process can be introduced by defining the following efficiencies:

$$\begin{aligned} \eta_E &= \frac{W_{\text{expansion}}}{W_{\text{expansion isentropic}}} \\ \eta_p &= \frac{W_{\text{pump isentropic}}}{W_{\text{pump}}} \end{aligned}$$

These relationships can be introduced into the cycle efficiency to give:

$$\eta_{\text{cycle T.M.}} = \frac{\eta_E \frac{W_{\text{expansion isentropic}}}{Q_{\text{in}}} - \frac{W_{\text{pump isentropic}}}{Q_{\text{in}}}}{1}$$

For the above cycle conditions, Table VII-3 shows the effect on steam cycle efficiency when η_E is set equal to η_p .

TABLE VII-3
STEAM CYCLE EFFICIENCY (η_{TM})

$\eta_E = \eta_p$	$\eta_p = 1.0$	$\eta_p = .8$	$\eta_p = .6$	$\eta_p = .4$	$\eta_p = .2$
Cycle Efficiency (η_{TM})	0.43	0.35	0.26	0.17	0.076
% Carnot	0.64	0.51	0.38	0.25	0.11

Comparison of Tables VII-2 and VII-3 quickly illustrate the basic difference between a Rankine cycle and a gas cycle. It shows that in a Rankine cycle, net work can be obtained with much lower efficiency components and that the effect of these inefficiencies is much more gradual than in a gas cycle.

In order to have as high an efficiency as possible, the component efficiencies should be as high as possible. This dictates the use of reciprocating components. Turbomachinery efficiencies would be too

low at these power levels. In addition, the high speed that would be required for turbomachinery would necessitate the use of gears in order to drive the blood pumps at the desired speed. The difficulty that would be encountered with turbines can be illustrated by reference to the TRW mercury turbine. This turbine, which is in a high state of development, has an engine efficiency of approximately 45% at the design point of 24,000 rpm and 5.3 horsepower (4 kilowatts) of shaft power. The difficulty of obtaining any reasonable efficiency at 0.001 times this power level with turbomachinery can easily be appreciated.

Thermo Electron's experience with small reciprocating steam engines presents an interesting comparison. An engine which was built two years ago has an isentropic efficiency of 54% at 1800 rpm and a shaft power of 190 watts. This engine operates at 300 psia and 800°F. This corresponds to a cycle efficiency of 13%, which is 28% of the Carnot efficiency for an 800°F inlet and a 212°F condenser. An engine which was built last year has an engine efficiency of 61% at 3600 rpm and 900 watts of shaft power. This engine operated at 900°F and 600 psia. This is a cycle efficiency of 18%, which is 36% of the Carnot efficiency for a condenser temperature of 212°F. It should be noted that these efficiencies have been obtained at 1/20 to 1/4 the power level of the mercury turbine and at speeds approximately 1/10 that used with the turbine.

D. CONCLUSIONS

Two basic approaches for utilizing a radioisotope source to power an artificial heart have been considered. In the first approach, the radioisotope thermal power would be converted into electrical

power with a thermoelectric or thermionic device. The electrical power would then be converted into mechanical power. It was shown that for this approach, the overall system efficiency was too low and the weight too high for this application. In the second approach a closed-cycle heat engine would convert the radioisotope thermal power directly into mechanical power. Gas cycles and liquid-vapor Rankine cycles were considered. The Stirling cycle appears to be the most promising gas cycle. For the immediate future, water appears to be the best working fluid for the Rankine cycle. However, organic working fluids hold a great deal of promise for longer range development.

The overall thermal-to-mechanical efficiency (cycle efficiency) for the Stirling cycle and Rankine steam cycle are compared in Figure VII-8. The only inefficiencies considered in this analysis are those associated with the expansion and compression processes. Although these are the major sources of inefficiencies, they are not the only ones. The ideal Stirling cycle efficiency will drop because of imperfect regeneration, aerodynamic flow losses, finite heat transfer coefficients and thermal conduction. The effect of these inefficiencies will be to make this break-even point even higher than 80%. It is not easy to achieve expansion and compression efficiencies that are this high in large machines. At the small size required for this very-low-power application, it would be extremely difficult.

At the same power levels, Stirling engines have been considerably heavier than Rankine engines operating on water. The prototype Stirling engine built for the U. S. Army Engineer Research and Development Laboratories (ERDL) by General Motors weighs 400 lbs and

produces 3 kilowatts of electrical power. This compares with the Thermo Electron 1.5-kilowatt prototype package built for ERDL which weights 120 lbs. The specific weight of the steam system at 0.080 lb per watt is half the specific weight of the Stirling engine at 0.133 lb per watt.

In conclusion, a Rankine engine at the low power output of this application should be both more efficient and lighter in weight than a Stirling engine, which is the second most attractive approach. Therefore, a Rankine engine, initially using a water working fluid, is the recommended system for converting the thermal power produced by the radioisotope source to mechanical power.

Figure VII-8. Comparison of Stirling and Rankine Steam Cycle

REFERENCES FOR CHAPTER VII

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CHAPTER VIII.

ESTIMATE OF RESOURCES

The objective of Task E., Estimate of Resources, is to prepare plans for the development of a fueled prototype of the implantable power source for a circulatory support system. More specifically, the statement of work requires:

"E. Estimate of Resources

Estimate the gross resources in manpower and materials, and the time required to carry the conceptual design through a preliminary detailed design, and to achievement of a workable prototype model."

A. APPROACH

In order to estimate the resources required to accomplish an objective it is necessary first to develop a plan which delineates the steps required to reach this objective. Furthermore the preparation of such a plan requires some assumptions concerning the relationship of the work in question to other concurrent programs relevant to the objective being sought. This is particularly true in this case. The program plan is affected by the AEC's parallel programs in developing the various radioisotope sources, safety studies of the use of radioisotope sources in wide scale treatment of heart disease, programs to determine allowable radiation dose rates in heart disease patients; NHI's programs for developing materials compatible with blood, blood pump development, physiologic control requirements, and the additional endogenous heat program.

If the object of the program were the clinical application of the implantable power sources, then, accurate information would be required concerning each of the above programs in order to develop the plan for the implantable power source. However, the purpose of the program to be planned is the "----- achievement of a workable prototype," which requires some knowledge of these other programs, but is not completely dependent on the timing and outcome of each program. Fortuitously, Thermo Electron is directly involved in most of the above programs, and is in a position to factor in the required information.

The following is a list of assumptions on which the planning of the development program for the implantable power source is based:

1. Although the "workable prototype" is not necessarily required to be implantable in a living organism, it should be as close as possible to fulfilling this requirement. The "workable prototype" should be suitable for animal experimentation, but not clinical application.
2. Since the total system must incorporate a blood pump as an integral subsystem, the program will include the effort necessary to include an optimized blood pump subsystem, based upon the then existing state-of-the-art in blood pump design, materials, and controls.
3. Even though it is unlikely that the allowable dose rates for heart disease patients will be firmly established in time to be used in the design of the workable prototype, specific dose rates will be assumed at the appropriate time, based upon the then existing best estimates.

4. Similar working assumptions will be made for the physiologic control system and the internal heat dissipation.
5. Emphasis shall be placed upon the use of Pu-238 as a heat source.
6. A continuous integrated program will be assumed, with no allowances for time lost due to renewals of contracts, lack of funds, etc.
7. Although long term operation without maintenance is a primary design and development goal, the plan does not include an extensive qualification program.
8. The primary plan is based upon two contractors continuing through Phase II, with one contractor completing the program.
9. The cost estimate does not include the cost of the required parallel efforts on the part of the AEC or NHI.

B. OVERALL PROGRAM

The program required to realize the clinical application of implantable power sources consists of four phases. Phase I is the Conceptual Design Study, under which this report is being prepared. The end results of the Conceptual Design Phase will be four reports, each describing a somewhat different implantable power source design. Two of these designs will be radioisotope powered steam engines, and the other two will be radioisotope powered Stirling engines. Based upon the results of the study, it should be possible to select the two most attractive systems. These reports are expected to be sufficiently complete to permit the initiation of a detailed preliminary design as

the next step, as well as the initiation of tests on vital components and subsystems

Phase II, Engineering Tests and Model Evaluation, spans an eighteen month period following the completion of the Conceptual Design Phase. The objective of this phase shall be to develop the major components and subsystems of the implantable power source and evaluate them experimentally. Specific tasks will deal with all of the component and subsystems identified during the Conceptual Design Phase as being critical and which require either testing to generate the required data, or sequential development to realize the required performance. Alternate materials and subassemblies will be evaluated.

Phase III, Prototype Development and Testing, is scheduled for thirty months. The purpose of Phase III is to integrate all of the components of the previous phase into a system design which is then fabricated, tested, and refined until it is a fully functional system. Restricted qualification tests of endurance and function will be performed. Fueled prototypes suitable for implantations in animals are the end objects of this phase.

The final phase, Phase IV, is the clinical application of the implantable power source in the treatment of heart disease, which represents the final goal of the overall program. Although the radio-isotope circulatory support system will be used for patient treatment, additional development and refinement of the controls efficiency, and reliability will be pursued.

There are additional work items not an integral part of the program, but which are very pertinent to achieving an optimized

effective clinical device. This work has been grouped into two categories:

1. Radioisotope Fuel Source Development
2. Required Parallel Efforts

The work under the radioisotope fuel source development includes 1) study of alternate fuels 2) choice of fuel, 3) refinement of fuel, 4) selection of fuel form(s), 5) compatibility studies and 6) tests of capsule design tests. This work is not included in the cost estimate later in the chapter since the AEC, itself is in the best position to forecast these expenses.

Cooperative, parallel efforts are required on the part of the AEC and NHI. The National Heart Institute would be expected to continue programs to develop 1) compatible blood-materials interface, 2) assist blood pumps, 3) total replacement blood pumps, 3) physiological control requirements and 4) additional endogenous heat constraints. Of course, other energy systems for circulatory support systems using biological fuel cells and rechargeable battery should continue to be developed.

In addition to the radioisotope fuel source development, the AEC would be expected to 1) prepare detailed plans for quantity medical grade radioisotope production 2) sponsor chronic, low level, radiobiological experiments with implanted sources 3) perform shield optimization studies and 4) conduct an evaluation of the hazards of general population use of radioisotope heat sources. This in-depth analysis would include:

1. A review of the radiation exposure of other people in society from such patients, especially those of the immediate family.

2. An assessment of the safety problems arising from automobile, plane, and other transportation related accidents.
3. An evaluation of the safety problems involved if a patient is accidentally incinerated either from a fire in his home or car, or from industrial accidents.
4. Recommendations will be formulated for the control and regulation (if required) of the patients in regard to airline travel, travel abroad, marking and identification of patients employment restrictions, methods of reclaiming fuel from deceased patients, etc.

To refer to the "blood pumps for the implantable power supply" is a misnomer. The blood pumps are an integral part of the total system, and must, of necessity, be developed as a subsystem. Features such as the control system, the heat rejection system, the low pressure reservoir, and the coupling mechanism between the hydraulic fluid and the blood are all essential considerations in the overall system design and analysis; yet, all are physically a part of the blood pumps. It would be impossible to build a meaningful implantable power supply without the blood pump subsystem.

On the other hand, there are separate programs in which research and development on blood pumps are being conducted. It is not the purpose of this project to compete with these other programs, but rather to incorporate the best that these programs can offer into the design of the blood pump subsystem. For instance, these other blood pump development programs are concentrating on problems such as the type

of valves the bladder configuration, fabrication techniques with new materials and blood flow characteristics through the pump. All of these items will be useful inputs to the development of the blood pump subsystem for the implantable power supply. Other aspects that must also be integrated into the design are: heat rejection to the blood flowing through the pump the coupling and hydraulic control components, and the low pressure reservoir. All of these factors must be incorporated into a functional design in such a way that the new configuration is capable of implantation in the body. Even after the complete prototype development has been initiated, much can be gained by the continual refinement and testing of critical components and subsystems. This work should be closely coordinated with the prototype development.

The prototype should be evolved sufficiently by Phase III to permit a thorough reliability analysis. The purpose of such an analysis is to identify specific areas which are critical to achieving the life and reliability goals of the program. In addition to identifying these areas, it should be possible to establish the degree of improvement required and the number and type of tests necessary to prove the attainment of these objectives. The results of the reliability analysis will be followed up by limited endurance testing.

C. TENTATIVE PROGRAM PLAN

The tentative program plan suggested by Thermo Electron has the following features:

AEC liaison will be close. The Project Office will aid in program formulation and assist in arranging for utilization of relevant technology developed under other AEC-sponsored programs (such as fuels technology).

Immediate fabrication and test of subsystems. This will quickly characterize the system problem areas and permit an early attack on potential bottlenecks.

Subsystems are decoupled, insofar as possible. Precise definition of subsystem functions permits early identification of development problems, and specialty groups can develop the individual subsystems without becoming bogged down in questions of subsystem interaction. By judicious interface specification, the subsystems identified in Table VIII-1 can be highly decoupled, thereby easing the problem of system integration. System development is divided into three (3) assembly steps.

Assembly I - Breadboard Assembly

The objective of this step is to evaluate the design of each subsystem and identify problem areas as soon as possible. Although some components may be oversized and inefficient, they should function individually and collectively.

Assembly II - Miniaturized Assembly

The objective of this step is to demonstrate high performance of all components in their compact size. However, the components will still be arranged in a breadboard fashion to allow easy component modification and system optimization. Improved efficiency, controllability, and reliability will be primary goals.

Assembly III - Implantable Assembly

The objective of this step is to package compactly and encapsulate the miniaturized assembly developed during Assembly II and to demonstrate its performance.

Definitive Program Milestones

Decoupling of subsystems and the three-phase assembly development sequence give many milestones by which progress can be measured throughout the program.

An abbreviated program plan for the radioisotope-fueled Rankine-cycle circulatory support system is given in Figure VIII-1, and a description of each task and sub-task is provided below:

TASK I: ADMINISTRATION

Subtask 1.1 - Program Direction

Administrative excellence is essential for technical success with minimum expenditure of time and money. Program direction will include periodic reviews and evaluations of project activities.

Subtask 1.2 - AEC Liaison

Close liaison with the AEC is imperative. In addition to the formal reporting channels informal channels will be used to ensure effective communication. The Program Office will aid in the initial project planning provide liaison with other contractors, and participate in the review and evaluation of the project's progress.

Subtask 1.3 - Reporting

Reporting will include monthly progress letters six-month technical progress report and final report.

Figure VIII-1. Program Plan for Radioisotope-Fueled, Rankine-Cycle Circulatory Support System

TASK II: SYSTEM INTEGRATION

In order to achieve the advantages of decoupled subsystems, an additional burden is placed on the systems integration tasks. Careful definition of subsystem specification and interfaces is mandatory for the subsystems to mesh into a compact and efficient circulatory support system.

Subtask 2.1 - Analysis

Thermodynamic, mechanical, hydraulic, and nuclear analysis of components, subsystems and system will be required throughout the life of the radioisotope fueled circulatory support system development. The design of definitive tests and interpretation of test data are important for system optimization. The analysis of test data is important for system optimization. The analysis of system response is especially important in coupling the engine with the blood pump and its controls.

Subtask 2.2 - Subsystem Specifications

To ensure that a given subsystem is neither under-designed nor grossly over-designed, judicious specifications must be made for all subsystems. Optimum performance will require judicious parameter choices which will require continual revision as test results accrue and system analysis is refined. Although carefully drawn subsystem specifications should not be rigid, they should not be changed without good reason. Otherwise, orderly subsystem development will be greatly perturbed, and the advantages of decoupled subsystems will be

vitiated. A formal review procedure, analogous to the "configuration change board" used in the aerospace industry, will be used to process major design changes so that program control can be maintained.

Subtask 2.3 - Subsystem Interfaces

This subtask is closely allied with Subtask 2.2. This effort coordinates the subsystem design efforts so that they will properly mesh together. In addition, the subsystem interfaces will be chosen for optimum system performance. The definition of each subsystem interface will require periodic review, but drastic modifications should not be necessary.

Subtask 2.4 - System Test

This subtask is limited to complete system tests (breadboard, miniaturized and implantable). These tests emphasize subsystem rather than component performance. Overall system efficiency and dynamic response will receive primary attention.

TASK III: POWER SOURCE DEVELOPMENT

The subsystem identification in Table VIII-1 shows that the power source is comprised of three subsystems: heat source, Rankine engine and hydraulic. The primary components of each subsystem are also identified in Table VIII-1. Every component will require design fabrication and test. Typically, this path will be retraced several times for each component. Sub-assemblies will be individually tested, then tested together as a system.

TABLE VIII-1

SUBSYSTEM IDENTIFICATION

Heat Source Subsystem

Radioisotope Fuel Capsule
 Shielding
 TES material
 TES container
 Vacuum foil insulation
 Boiler — — — — —

Rankine Engine Subsystem

Boiler — — — — —
 Working fluid
 Valves
 Cylinder
 Piston
 Flywheel
 Condenser
 Feedwater pump
 Hydraulic pump — — — — —

Hydraulic Subsystem

Hydraulic pump — — — — —
 Sump pump
 High-pressure accumulator (reservoir)
 Low-pressure accumulator (reservoir)
 Hydraulic line

Blood Pump Controls

Hydraulic line
 Pressure sensor
 Volume sensor
 Hydraulic valve
 Pilot valve
 Pressure transducer — — — — —

Blood Pump

Pressure transducer — — — — —
 Fluid coupling
 Pump bladder
 Check valves
 Pump housing
 Autologous interface with blood

Subtask 3.1 - Radioisotope Fuel Source

This subtask is primarily one of design integration of the fuel capsule into the implantable power source. Choice of fuel, fuel form, fuel purity, encapsulation material and capsule design will be in conjunction with the AEC and its other contractors. Fuels technology status, hazards analyses and radiobiological data will be factored into these decisions. The source fabrication and qualification will be accomplished at other facilities.

Subtask 3.2 - Thermal Energy Storage

The thermal energy storage (TES) subassembly will be designed along the lines of the TES prototype. Fabrication problems will be solved, life tests (container and static vacuum) performed and subassembly performance demonstrated.

Subtask 3.3 - Vacuum Foil Insulation

Thermo Electron experience gained under AEC-sponsored program, "Vacuum Foil-Type Thermal Insulation for Radioisotope Systems," will be factored into the insulation package development. Foil and spacer materials will be chosen from the results of this program. However, specific edge and penetration problems peculiar to the configurations of the implantable power source hardware will require special attention.

Subtask 3.4 - Engine Development

The work under this subtask includes all design, fabrication and test activities required to develop a high-performance Rankine-cycle engine. The detailed activities will be based on Thermo

Electron's experience in developing 100-, 300-, 600-, and 1500-watt steam engines. This experience has emphasized the importance of early hardware test. Extensive tests of working fluid-lubricated bearings will be made to confirm the soundness of the design used. Reduction of thermal losses will receive primary attention. Parallel organic and water working fluid programs will be pursued. Choice of working fluid will be on basis of hardware tests.

Subtask 3.5 - Hydraulic Development

The primary problems associated with the hydraulic subsystem involve developing pumps (feedwater, hydraulic and sump) and reservoirs with low leakage and low pressure drops. In addition, this subtask includes the development of miniature pressure relief valves.

Subtask 3.6 - Prototype Assembly and Test

The assembled power source provides the best test bed for testing the thermal storage engine and hydraulic subassemblies. Parallel approaches will necessitate the fabrication of several prototype assemblies. Breadboard, miniaturized and implantable power source assemblies will be fabricated and tested.

TASK IV: BLOOD PUMP AND BLOOD PUMP CONTROLS

Conceptually, the reference control scheme in the circulatory support system is identical to that of the present console used to drive the TECO-CHMC assist pump. The object of this task is to evolve control hardware which performs the same function but is hydraulically

actuated. The blood pump will be modified to accept hydraulic drive and perform certain additional functions.

Subtask 4.1 - Assist Pump Modification

The blood pump will be modified to function not only as a pump, but also as a heat exchanger. Fortunately, the modifications to the CHMC-TECO pump are relatively minor. This involves primarily modification in the size and location of the drive tube.

Subtask 4.2 - Blood Pump Controls

Both counterpulse and "follower" blood pump control will be designed and fabricated. The choice of control system will be on the basis of hardware tests and in vivo data.

Subtask 4.3 - Prototype Assembly and Test

The blood pump and control assemblies will be tested using a hydraulic accumulator power source and a mock circulation loop. Component substitution and adjustment will be made to give the most reliable and physiologically desirable pumping mode.

Tasks 3 and 4 require several in vivo tests to support the engineering activities. The problems that require animal experimentation are: (1) diaphragm penetration, (2) component placement, (3) power source suspension, (4) heat transfer through the blood-pump bladder, and (5) control concept demonstration.

D. ESTIMATE OF RESOURCES

The resources required to accomplish the objectives of the program consist of qualified manpower facilities, and money.

1. Manpower

The program plan has been prepared assuming that the development program will be undertaken by a fully trained biomedical engineering group familiar with all aspects of the program, working in conjunction with a medical group consisting of both cardiologists and cardiovascular surgeons who are involved in artificial heart research and development. Manpower requirements, by classification, for the period of the program are shown in Table VIII-2. The total program will require approximately 560 man-months of effort, spread over a forty-eight-month period. The program plan is such that a relatively small trained staff would remain with the program over the entire four-year period. The typical level of effort is around 16 or 17 men per month, with a peak of only 19 men. This stability of level of effort is a necessity for programs requiring a staff of highly trained specialists.

2. Facilities

The facility requirements are relatively modest. Most of the components are small precision parts, and the power rating is such that extremely accurate but small test equipment is required. Specialized facilities such as mock circulatory loop and dynamometer will be required. A laboratory with approximately 10,000 ft² of floor space should be adequate. Other special facilities, such as a precision machine shop, component and prototype test stations, and mock circulation test facilities will be required to support the program.

Table VIII-2. Manpower Requirements (Man-Months)

FISCAL YEAR		FY 1968			FY 1969						FY 1970						FY 1971						TOTAL (Man-Months)
	Man-Months	2	4	6	8	10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40	42	
BIO-MED. ENGINEERING																							
1.	Program Manager	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2				36
2.	Systems Engineer		1	2	2	2	2	2	2	2													15
3.	Senior Bio-Med Engr.		1	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2				33
4.	Senior Mech. Engr.	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2							30
5.	Senior Controls Engr.		1	2	2	2	2	2	2	2	2	2	2	2	2	2							21
6.	Reliability Spec.													2	2	2							6
7.	Mech. Engrs. & Design	4	6	6	4	4	4	4	4	6	4	4	4	4	4	4	4	3	2				75
8.	Mech. Tech.		3	6	8	8	8	8	8	8	8	8	8	8	10	8	8	4	4	2			119
9.	Medical Tech.		2	2	2	2	2	2	2	2	2	2	2	2	2	6	6	6	6				49
10.	Other Support		10	10	10	10	10	10	10	10	10	10	10	10	8	6	6	6	6				152
	Personnel (Machinists)																						
	Processing Tech. Data																						
	Processing personnel																						
	Technical publications																						
	Personnel, draftsmen etc.)																						
	Sub-total	8	27	34	34	34	34	34	34	36	32	32	34	32	32	24	23	23	20	Sub-total			536
MEDICAL SUPPORT																							
1.	Cardiologist		1/4		1/4		1/4		1/4		1/4		1/4		1/4	1/2	1/2	1/2	1/2				3 3/4
2.	Surgeon		1/4				1/4				1/4				1/4	1/2	1/2	1/2	1/2				3
3.	Medical Support Personnel															4	4	4	4				16
	Sub-total	-	1/2	-	1/4	-	1/2	-	1/4	-	1/2	-	1/4	-	1/2	5	5	5	5	Sub-total			22 3/4
	TOTAL	8	27 1/2	34	34 1/4	34	34 1/2	34	34 1/4	36	32 1/2	32	32 1/4	34	32 1/2	37	29	28	25	TOTAL			558 3/4

3. Estimated Cost

The estimated cost of Phase II and III to achieve a workable, Rankine-cycle, prototype is \$1,730,000 for a single contractor. However, it would be wise to carry at least two contractors through Phase II. A cost breakdown is given in Table VIII-3.

This cost estimate does not include fuels technology effort, fuel capsule development, hazard analysis, radiobiological experiments, production planning expense, or program administration by the AEC. Also it does not include the cost of parallel efforts on blood compatible materials, blood pumps, physiological control requirements, and additional endogenous heat.

TABLE VIII-3
ESTIMATED COST OF PROGRAM
(x \$1, 000)

	TOTAL <u>FY 68, 69, 70, 71</u>
<u>LABOR</u>	
1. Tech. Management and Senior Engineer	\$ 197,000
2. Designers and Technicians	245,000
3. Machinists and Technical Support Personnel	<u>115,000</u>
Labor Sub-Total	\$ 557,000
<u>OVERHEAD</u> (144%)	803,000
<u>MATERIAL and ODC</u>	150,000
<u>SUBCONTRACT</u> (Medical Support)	<u>14,000</u>
Sub-Total	\$1,524,000
G and A (at 4.6%)	<u>70,000</u>
Sub-Total	\$1,594,000
Fee (at 8.5%)	<u>136,000</u>
TOTAL ESTIMATED COST	\$1,730,000

APPENDIX A

CIRCULATORY SUPPORT SYSTEM REQUIREMENTS

To provide a basis for the selection of a design approach to the development of an implantable power source for a circulatory support system, the following discussion will review briefly the major requirements for such a system and evaluate alternative choices for the radio-isotope fuel and the thermodynamic converter.

A. REQUIREMENTS

1. Patient Needs

Efforts to estimate the need for various types of artificial heart devices have met with little success because available data are not sufficient to define the requirements in a detailed and specific manner. While it is clear that many types of patients may benefit from temporary ventricular assistance, it is equally clear that others will require permanent circulatory support in the form of either a left ventricular assist device or a total heart replacement. Not only is it difficult to estimate the number of patients in each category at the present time, it is virtually impossible to predict whether temporary ventricular assistance will be effective in restoring cardiac function in a particular case or whether prolonged circulatory support will be required.

Despite the present uncertainties of diagnosis, it is apparent that at least the following types of patients will require a total heart replacement:

- Combined right and left ventricular failure secondary to a massive myocardial infarction.
- Extensive myocardial fibrosis with intractable angina.
- Rheumatic heart disease patients with significant myocardial damage secondary to years of chronic valvular disease.
- Multi-valve replacement patients who fail to respond post-operatively.
- Congenital heart disease patients.

In addition, many of the patients initially treated with temporary assist devices will suffer progressive myocardial damage because of coronary artery insufficiency, develop aneurysms or myocardial rupture, and also require a total heart replacement. Finally, it is certain that some percentage of the patients receiving temporary ventricular assistance will not recover sufficiently to leave the hospital without some form of permanent circulatory support.

In all instances in which a prosthetic blood pump must be used for more than a few weeks, either as a total heart replacement or a left ventricular assist device, an implantable power source is necessary to avoid bacterial contamination and infection. In addition, the use of an implantable system eliminates the possibility of accidental damage to external components and promotes vocational rehabilitation.

Until a detailed prospective study of heart disease patients has been completed, it will be difficult to estimate with accuracy the

number of patients whose lives can be extended by the use of totally implanted circulatory support system. It is generally agreed that such patients number in the tens of thousands each year, and some estimates are as high as several hundred thousand per year.

In dealing with a patient population of this size, it must be recognized that no single device can perfectly match the requirements of all individuals. Variations in age, body surface area, activity level, occupation, and degree of circulatory support required will almost inevitably require the use of a family of devices designed to cover a wide range of specifications. The extent of this range cannot be predicted on the basis of available data; therefore, the following discussion is based on typical characteristics of the "average" heart disease patient.

2. Blood Pumping Power

The average power index ($\text{watts}/\text{meter}^2$) of the biologic heart has been calculated as a function of cardiac index ($\text{liters}/\text{min}/\text{meters}^2$) for 10, 20, 50 and 80-year old patients at heart rates of 70 to 200 per minute. The results are recorded in Figure A-1. Assuming an average body surface of 1.73 m^2 , a minimum power of 1.04 watts is developed in the aged patient at rest, and a maximum power of 6.2 watts is developed in the 20-year-old man during periods of exercise. For a more complete and detailed treatment of the pumping power of the normal heart, see Appendix B.

A typical daily power profile for a sedentary worker is presented in Figure A-2, which shows peak power levels 2.5 times the basal values. Note that the power also varies over one cardiac cycle, as shown in Figure A-3. The peak instantaneous power during ventricular systole is approximately five times the average pumping power.

Since the duration of the maximum instantaneous power peaks is quite short, the energy required for these periods is small.

For purposes of this discussion, the various terms used to describe the power characteristics of the heart are defined as follows:

Mean Power - 24-hour average power consumption (1.75 watts)

Average Power - Power averaged over each cycle

Peak Power - Power averaged over each cycle during periods of high activity (approximately 4 watts)

From the equation in Figure A-1, the pumping power of the left ventricle can be calculated by eliminating the term for peak right ventricular pressure, P_R . The ratio of the pumping power of the left ventricle to that of the heart becomes

$$\frac{Q_L}{Q_H} = \frac{P_L - 4}{P_L + P_R - 4}$$

The peak right ventricular pressure (P_R) is very nearly 20% of the peak left ventricular pressure (P_L) for various age groups and body surface areas, or

$$P_R = 0.2 P_L$$

Therefore,

$$\frac{Q_L}{Q_H} = \frac{P_L - 4}{1.2 P_L - 4} = \frac{P_L}{1.2 P_L}, \text{ with less than 3\% error,}$$

and

$$Q_L = 83\% Q_H$$

Figure A-1. Calculated Power Index versus Cardiac Index for Various Ages

CARDIAC OUTPUT AND AVERAGE POWER VS. TIME FOR A TYPICAL DAY OF OFFICE WORK

Figure A -2. Typical Daily Power Profile

POWER VARIATION DURING SYSTOLE

Figure A-3 Ratio of Instantaneous Power to Average Power over one Cardiac Cycle

Thus, one can determine the pumping power of the left ventricle from Figures A-1 and A-2 by realizing that 83% of the pumping work of the heart is performed by the left ventricle.

The function of a ventricular assist device is to supplement the work of the left ventricle. It may be required to handle as little as 20% to 30% or as much as 100% of the work of the normal left ventricle. A circulatory support system operating as an assist device may be required to produce as much as 83% of the work of the normal biological heart, which for a typical (statistical mean) office worker with a body surface area of 1.73m^2 , would be a mean power of 1.45 watts, with peak power ranging up to 3.3 watts.

A total heart replacement must pump the blood of both the left and right ventricles. However, there will be no coronary flow, which reduces the power by approximately 5%. For the same typical office worker, the mean power of the total heart replacement is then 1.66 watts, with peak power ranging up to 3.8 watts.

3. Energy Storage Requirements

From the foregoing discussion, it is apparent that the power requirements of the heart are anything but constant. The power demand may increase by a factor of $2\frac{1}{2}$ for periods as long as 20 minutes. During such a period the additional energy consumed is considerable. To satisfy these requirements the system must be capable of storing thermal, mechanical, or electrical energy, or it must contain enough radioisotope fuel to generate the amount of heat required at peak power conditions. Assuming a 15% system efficiency, 1.75 watts of mean blood pumping power and 4 watts of peak power, the thermal power

input with no energy storage is 26.6 watts, while with energy storage it is only 11.65 watts. Therefore, more than twice as much fuel is required if energy storage is not employed. More than half the heat released is wasted when energy storage is not used, and of course, this additional heat load must be dissipated within the body. When one considers the difference in heat rejected within the body, the radiation dose rates from the isotope, and the cost and supply of isotopes, it is apparent that provision for energy storage must be included in the design of an optimum system.

The time integral of the power profile of Figure A-2 is shown in Figure A-4 and yields the energy consumption vs time. The resulting curve is very nearly a straight line, indicating that even though the power peaks are high, their effect is diminished by their short duration. The slope of a straight line through the origin and tangent to the curve in Figure A-4 represents the average power, which is 1.75 watts. (The line corresponding to a system without energy storage is shown for comparison.) The maximum deviation of the energy-vs-time curve from the line of average power (1.75 watts) defines the energy storage requirement, which under the assumed conditions is 3.5 watt-hours.

This energy storage can be either electrical or thermal. The energy that must be stored is:

$$E = \frac{\Delta E_{\max}}{\eta_c}, \quad \text{where } \eta_c \text{ is the efficiency of the conversion system, either electrical or thermal.}$$

It is apparent that high conversion efficiency is desirable not only to minimize fuel inventory and reject heat, but also to reduce the weight of a system including energy storage.

4. Physiologic Requirements

The operation of a circulatory support system must be controlled in such a way as to interact closely with normal physiologic regulatory mechanisms. The primary function of a total heart and an assist device is to maintain cardiac output, according to the demands of the body. A ventricular assist device must replace that aspect of cardiac function which is lost in the damaged heart, must reduce ventricular pressure, and must maintain or increase coronary blood flow. Whichever system is used, a detailed definition of the control system function and the selection of the physiologic parameters to be sensed and controlled must be based on an understanding of the normal and abnormal characteristics of the biological control system, which consists of a set of intrinsic mechanisms augmented by a set of extrinsic mechanisms.

a. Intrinsic Mechanisms

The intrinsic control mechanisms are responsible for cardiac autoregulation, independent of central and peripheral controls. This independence has been amply demonstrated by experiments in which total cardiac denervation (cutting of all sympathetic and parasympathetic nerve connections) has led to maintenance of such autoregulation at a reduced level.

Heterometric autoregulation is by far the more important of the two intrinsic mechanisms and may be defined simply as the ability of the heart to eject whatever volume of blood it receives from the vena cava. This is commonly known as the Frank-Starling mechanism and is initiated by an increase in venous return to the right atrium, which raises right atrial pressure (and, hence, end-diastolic pressure) and increases blood volume in the right ventricle. Thus, the stroke volume and output of the right ventricle are increased.

Figure A-4. Energy Consumption versus Time for a Typical Day

The resulting increase in pulmonary flow causes a corresponding increase in left atrial pressure, within a few beats. Again, the Frank-Starling mechanism causes a proportional increase in the output of the left heart, tending to stabilize left atrial pressure over a narrow range. This mechanism is vital since it simultaneously controls cardiac output and maintains a balanced flow between the left and right heart.

The homeometric autoregulation mechanism is responsible for smaller changes in cardiac performance than heterometric autoregulation, acting over many beats of the heart. It may be simply defined as the ability of the heart to perform more work when faced with stress, but at pre-stress levels of filling pressure and volume. Thus, it acts to increase efficiency and to preserve the heterometric mechanism for further stress adaptation. Two principal phenomena are capable of eliciting this change:

1) Increased Aortic Pressure (Resistance)

First described by Anrep in 1912, and later detailed by Sarnoff, this mechanism enables the heart to generate more pressure when peripheral resistance increases.

2) Increased Heart Rate

First described by Bowditch in 1871, and later detailed by Sarnoff and by Mitchell in denervated hearts, this mechanism enables the heart to decrease its systolic duration at a given filling volume and pressure when heart rate increases.

When the left ventricle is damaged, it often loses a significant portion of its heterometric and homeometric reserve, while the equivalent bi-atrial and right ventricular mechanisms remain relatively unaltered. Thus, a state of left ventricular failure ensues, since the ventricle can no longer adequately adapt to changes in pumping requirements: Cardiac output falls and left atrial pressure rises. Any assist pump, and hence its control system, must be able to make up for such a deficit in cardiac output, which may result in shock, and reduce the potentially dangerous elevation in left atrial pressure, which may result in pulmonary edema. Thus, by artificially restoring the intrinsic regulation of atrial pressure to normal performance levels, normal output is restored, and extrinsic mechanisms continue to control blood pressure and venous return normally.

The rate of contraction of atria and ventricles is dependent on the intrinsic generation of an electrical signal by the sino-atrial node. The rate at which this node fires may, in turn, be altered by extrinsic influences. Of itself, however, it is a prime intrinsic regulator of cardiac output, since doubling of heart rate, at constant stroke volume, doubles output.

b. Extrinsic Mechanisms

As long as the nervous system is intact and total body blood volume is relatively normal, this group of biological controls will continue to function normally. There are two main types of extrinsic controls, functioning simultaneously by way of the sympathetic (adrenalin) and parasympathetic (acetylcholine) nervous systems:

1) Peripheral Vascular Regulation

The most important extrinsic regulation mechanisms are the responses of the peripheral vascular network, which compensate for changes in state (e. g. , rest to exercise). Signals from the carotid body are used to control systemic blood pressure by regulating peripheral vascular resistance. By a similar pathway, peripheral chemical sensors (oxygen, acidity) can also regulate vascular resistance and thus aid in maintaining appropriate tissue perfusion pressure at any given cardiac output. During demand for increased cardiac output, signals to and from the central nervous system cause increased venous return, thus increasing atrial volume and pressure, and eliciting an immediate Frank-Starling response manifested as increased cardiac output.

2) Cardiac Regulation

The central nervous system and peripheral controls can increase or decrease heart rate and affect atrial or ventricular myocardial contractility through the homeometric mechanism.

c. Abnormalities in Rate and Rhythm

Three abnormalities in the cardiac rate and rhythm are of considerable importance in the design of a control system for a ventricular assist device: First, when no normal electrical signal is transmitted from the node (atrio-ventricular block), the ventricle contracts at too slow a rate (30 - 50 beats per minute) to maintain normal cardiac output. Second, when an abnormal focus of electrical signal generation is present in acutely (myocardial infarction) or chronically (fibrosis, ischemia) damaged hearts, an irregular and/or rapid rhythm will result. Third,

when all electrical signals cease completely, or when ventricular fibrillation occurs (random electrical activity without effective muscular contraction), irreversible damage (shock) or death will quickly result

Such abnormalities in rate and rhythm must be corrected to acceptable levels before a patient would be considered for a permanent assist device. Those patients experiencing continuing abnormalities will be candidates for a total heart replacement.

The foregoing discussion indicates the nature of the interface between the circulatory support system, or subsystem, and the complex biologic system of which it must be an integral part. This interface has not, unfortunately, been defined in detail as yet, nor does it lend itself to the conventional type of engineering description. It is apparent, however, that a thorough knowledge of the normal and abnormal physiologic environment is a critical input to the design of an effective circulatory support system.

5. Control System Requirements

The essential function of the control system is to maintain proper cardiac output under varying conditions. During periods of stress or moderate exercise, cardiac output is increased physiologically by increasing stroke volume and heart rate. Thus, in accepting a given fraction of the work of a damaged ventricle, an assist device should be able to increase cardiac output (up to threefold) by increasing stroke volume and beat rate. The control system must apportion the increase in output between these two parameters. Further, since the demands

for changing the output are frequently transient, rapid and sensitive responses are mandatory.

A further requirement is that the control system must maintain proper balance between the output of the biologic left ventricle and assist pump, on the one hand, and the right ventricle, on the other. Since atrial pressure is proportional to blood volume accumulation, balance is achieved by sensing and controlling this variable.

As discussed above, central nervous system regulation of heart rate, by its influence upon the sino-atrial node, continues even in the presence of significant left ventricular damage. Current assist devices are synchronized and counterpulsed to eject blood during early ventricular diastolic filling.

Since the assist pump must be counterpulsed with the ventricle, the beat rate can only be adjusted in discrete steps (B , $B/2$ or $B/3$). In the moderately ambulatory patient, frequent changes in left atrial pressure would require frequent changes in beat rate. Since atrial pressure changes in a gradual, continuous manner (although the rate of change may be quite rapid), it would be desirable to have a control mechanism which can respond to relatively small changes in left atrial pressure without changing beat rate.

This can be achieved by controlling stroke volume with the assist device rate equal or proportional to heart rate, in order to maintain the normal heart rate response mechanism. The assist device must be regulated to discharge a volume of blood equal to its intake. The intake can be adjusted by varying the filling pressure and the filling duration, as in the Frank-Starling mechanism of the heart. This results in a control system which approximates the physiologic control

mechanism, providing a continuous response to changes in left atrial pressure.

In the series-assist device, filling pressure is the primary mode of control to reduce ventricular pressure. In a parallel-assist device, the gradient between left atrial pressure and assist pump filling pressure determines the volume of blood flowing into the assist pump and, therefore, the stroke volume. In the case of a parallel assist device, also, control of filling pressure permits regulated increases in ventricular pressure, and, consequently, in ventricular work, as the ventricle heals.

The control system for the total heart replacement must regulate the heart rate and stroke volume to ensure an adequate flow of blood during the varying load conditions faced by the body. In the natural heart, both central nervous control and venous return are active in regulating cardiac output to match the physiologic requirements. In the present stage of development, only the venous return appears accessible as a controlling variable for the replacement. The control system must thus respond to changes in that variable to produce the required threefold increase in cardiac output without exceeding the normal physiologic limits for atrial pressures. All vascular system pressures must be maintained within the normal range. Left and right heart flow will be balanced by the atrial pressure response of the ventricles. Normal pressure-time waveforms must be produced. Transient response of the total heart replacement to changing physiologic demand must be as rapid as that of the normal heart.

The major difference in the control for the assist device compared with that of the total replacement is the requirement for synchronizing the assist with the natural heart. It is undesirable to use electrical inputs for this purpose because of lead breakage and amplification requirements. In the case of assist devices requiring counter-pulsation, a pressure-peak fluid sensor can be used to provide a triggering signal. Because of the overall compactness of an implanted system, the delays involved should not be excessive. The trigger pulse will be generated near the start of natural systole, and the assist contraction can be delayed until shortly after the end of systole.

APPENDIX B

DETERMINATION OF BLOOD-PUMPING POWER REQUIREMENTS FOR AN ARTIFICIAL HEART

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APPENDIX B

DETERMINATION OF BLOOD-PUMPING POWER REQUIREMENTS FOR AN ARTIFICIAL HEART

INTRODUCTION

A basic consideration in the design of an artificial heart is the pumping power which is required to circulate blood throughout the body. This is especially important for the permanent left ventricle assist device and the total heart replacement, where the patient may be equipped with a portable power supply, perhaps implanted within the body. The size, weight, and power requirements of this power supply depend directly on the blood-pumping power required. In order to design optimized artificial heart devices, it is necessary to know how pumping power varies as a function of physical activity, emotional state, and age.

The purpose of the work reported here was to determine the blood-pumping power requirements from the available but incomplete hemodynamic data. The importance of instantaneous, average, and daily mean pumping power is discussed.

HEART POWER CALCULATION

The work done by the normal heart can be calculated from the pressure-versus-time and volume-versus-time diagrams for the left and right ventricles. It does not include the energy changes of the arterial or venous walls, which are passive, compliant elements of the cardiovascular system. Then the work calculated in this way is the blood-pumping work. This energy is dissipated as heat by friction or viscosity effects in the small arteries, arterioles, capillaries, and venules.

Work per stroke is calculated, as shown in Figure B-1 by integrating pressure times incremental volume change over systole, the period of ventricular blood ejection. Pressure (P) and volume (V) are functions of time, and the integration between the beginning and end of systole is straightforward.

In order to calculate ventricular work per stroke, graphical or numerical integrations can be made. Numerical integration was used because it more readily permits a variety of numerical analyses and calculations. The accuracy of the numerical calculations depends on how well the empirical or mathematical curve fits the data. Average power is work per stroke times heart rate, usually in beats per minute. Instantaneous power can also be calculated to determine power and energy changes within each beat. It is equal to instantaneous pressure multiplied by the instantaneous flow, $\frac{dV}{dt}$.

Figure B-2 shows data compiled from typical, normal left ventricular pressure-versus-time traces. The empirical curve is a fourth-order polynomial equation in time for the period of systole. The fit of this empirical curve, the dashed line shown here, is seen to be fairly good. The polynomial equation of the empirical curve is shown in the form used to adjust it to fit the data.

Figure B-3 similarly shows data for the left ventricle volume versus time, and an empirical curve, which is again a polynomial equation.

The fit of both of these empirical curves was considered satisfactory because the error in fit results in less error in the average power calculation than that resulting from the inaccuracies of the data and the variation in curve shape among individuals.

BLOOD PUMPING POWER CALCULATIONS

$$\text{Average Work Per Stroke} = \int_{P_1}^{P_2} P dV = \int_{t_1}^{t_2} P \frac{dV}{dt} \cdot dt = W_s$$

where: $P = P(t)$

$V = V(t)$

P_1, t_1 = pressure and time at beginning
of systole

P_2, t_2 = pressure and time at end of systole

$$\text{Average Power} = W_s B$$

where: B = heart rate

$$\text{Instantaneous Power} = \frac{dW}{dt} = P \frac{dV}{dt}$$

Figure B-1. Blood Pumping Calculations

VENTRICULAR PRESSURE - TIME CURVE FITTING

Figure B-2. Ventricular Pressure - Time Curve Fitting

VENTRICULAR VOLUME-TIME CURVE FITTING

Figure B-3. Ventricular Volume - Time Curve Fitting

Figure B-4 shows various approximations for the empirical curve fitting. Some references in the literature suggest the use of a constant peak ventricular pressure approximation, which gives a rather high value of work output. The sinusoidal approximation is a somewhat better fit. The more accurate polynomial approximation indicates the work is $0.68 P_s V_s$, with a possible error of about $\pm 10\%$ for normal individuals. It is useful to derive curves of power output versus age and activity for normal humans. Such curves can be determined from the hemodynamic data. The following figures indicate the extent of the available data.

Figure B-5 shows basal systemic blood pressures in mm Hg and its standard deviation for almost 80,000 adults in the age groups from 20 to 106 years. Systolic pressure is seen to increase by about 20% over the age span, with the deviation increasing from about 10% to 20%.

Figure B-6 gives cardiac output and its scatter for individuals from 19 to 86 years of age. This wide scatter is not completely attributable to differences in body surface area or in weight. Considerable variation in cardiac output is found among people of the same body surface area or weight. The last two figures indicate that considerable data exists for adults at rest.

Figure B-7 gives an attempt to correlate mean systemic blood pressure with cardiac index, based on data in the literature. Here the effects of age and activity are presented. A similar graph can be drawn for the pulmonary system. Moreover, peak ventricular pressures can be similarly correlated with the variables, leading to the desired blood-pumping power correlation.

VENTRICULAR SYSTOLIC PRESSURE AND VOLUME APPROXIMATIONS

Figure B-4. Ventricular Systolic Pressure and Volume Approximations

BLOOD PRESSURES AND STANDARD DEVIATION IN APPARENTLY
HEALTHY PERSONS, 20 TO 106 YEARS OF AGE

AGE GROUP	MALES		FEMALES	
	SYSTOLIC	DIASTOLIC	SYSTOLIC	DIASTOLIC
20 - 24	123 ± 13.7	76 ± 9.9	116 ± 11.8	72 ± 9.7
25 - 29	125 ± 12.6	78 ± 9.0	117 ± 11.4	74 ± 9.1
30 - 34	126 ± 13.6	79 ± 9.7	120 ± 14.0	75 ± 10.8
35 - 39	127 ± 14.2	80 ± 10.4	124 ± 13.9	78 ± 10.0
40 - 44	129 ± 15.1	81 ± 9.5	127 ± 17.1	80 ± 10.6
45 - 49	130 ± 16.9	82 ± 10.8	131 ± 19.5	82 ± 11.6
50 - 54	135 ± 19.2	83 ± 11.3	137 ± 21.3	84 ± 12.4
55 - 59	138 ± 18.8	84 ± 11.4	139 ± 21.4	84 ± 11.8
60 - 64	142 ± 21.1	85 ± 12.4	144 ± 22.3	85 ± 13.0
65 - 69	143 ± 26.0	83 ± 9.9	154 ± 29.0	85 ± 13.8
70 - 74	145 ± 26.3	82 ± 15.3	150 ± 25.8	85 ± 15.3
75 - 79	146 ± 21.6	81 ± 12.9	158 ± 26.3	84 ± 13.1
80 - 84	145 ± 25.6	82 ± 9.9	157 ± 28.0	83 ± 13.1
85 - 89	145 ± 24.2	79 ± 14.9	154 ± 27.9	82 ± 17.3
90 - 94	145 ± 23.4	78 ± 12.1	150 ± 23.6	79 ± 12.1
95 - 106	146 ± 27.5	78 ± 12.7	149 ± 23.5	81 ± 12.5

Figure B-5. Blood Pressures and Standard Deviation in Apparently Healthy Persons, 20 to 106 Years of Age.

AGE CHANGE IN BASAL CARDIAC OUTPUT IN NORMAL MALES

Figure B-6. Age Change in Basal Cardiac Output in Normal Males

CARDIOVASCULAR CHARACTERISTICS OF THE HUMAN MALE

Figure B-7. Cardiovascular Characteristics of the Human Male

Figure B-8 shows a typical variation of left and right mean atrial pressures with cardiac index. For the purposes of power output calculations, the venous or atrial pressure at the beginning of systole was assumed to be zero, and the left atrial pressure was assumed to be 4 mm Hg. These are the basal mean atrial pressures. This assumption results in a maximum error of the power output calculation of about 6% , on the high side. This is consistent with the accuracy of the data and the pumping-power calculation. The inaccuracy of correlation between body surface area and cardiac function data necessitates the measurement of blood pressures and cardiac output for each individual to determine his blood-pumping power. However, the correlation is valuable for showing trends and for preliminary design and development of artificial heart devices.

Based on data shown in the preceding figures and other data, peak ventricle pressure versus age, activity, and cardiac index can be determined. Less data are available for high activity and for the right ventricle and the atria; it is hoped that more of this data will become available in the near future.

Figure B-9 shows left ventricular function versus age and activity. The decrease in cardiac index with age is shown, as is the increased resistance to flow (the increasing pressure with age at constant cardiac index) and the decreased reliance on stroke volume for increasing cardiac output.

A similar plot of right ventricle function is shown in Figure B-10. Peak right ventricle pressures are seen to be about one-fifth the corresponding pressures of the left ventricle. An increasing pulmonary resistance to flow with age is indicated.

The previous two graphs of left and right ventricular function permit calculation of power output of the heart; the results are given in Figure B-11. The equation for power output gives the previously determined constant based on the shapes of ventricular pressure and volume curves, the cardiac output or index (since stroke volume times heart rate equals cardiac output), the sum of left and right peak ventricular pressures (given on the two previous figures) minus the mean atrial pressure, which here is assumed constant, and a conversion factor to obtain power in watts or watts per meter.²

For adults, using the typical body surface area of 1.73 m^2 , a minimum power of 1.04 watts is developed in the 80-year-old at rest, and a maximum power of 6.2 watts is developed in the 20-year-old during periods of high activity.

Power output can, in certain individuals, be considerably greater or less than the extremes indicated here. It has been determined from the available data that some individuals have power outputs as great as 12 watts during periods of high activity.

The assumption that the polynomial which fits the left ventricle pressure-volume curve also fits the right ventricle curve is fairly accurate. Errors in right ventricle data or curve-fitting are somewhat less important because the power produced by the right ventricle is about one-fifth the power of the left ventricle.

LEFT AND RIGHT ATRIAL PRESSURE AS A FUNCTION OF CARDIAC INDEX

Figure B-8. Left and Right Atrial Pressure as a Function of Cardiac Index.

PEAK LEFT VENTRICLE PRESSURE vs CARDIAC INDEX FOR VARIOUS AGES

Figure B-9. Peak Left Ventricle Pressure versus Cardiac Index for Various Ages

PEAK RIGHT VENTRICLE PRESSURE vs CARDIAC INDEX FOR VARIOUS AGES

Figure B-10. Peak Right Ventricle Pressure versus Cardiac Index for Various Ages

Figure B-11. Calculated Power Index versus Cardiac Index for Various Ages

PRESSURE MEASUREMENT FOR THE HEART POWER CALCULATION

Euler's equation for an ideal (frictionless) fluid with constant density is written:

$$\int \frac{\partial V}{\partial t} ds + \frac{V^2}{2} + \frac{P}{\rho} + gh = f(t)$$

where:

V = velocity

t = time

ds = element of axial distance

P = static pressure

ρ = mass density of blood = $1.057 \frac{\text{gm}}{\text{ml}^3}$

g = acceleration of gravity

h = height above an arbitrary reference surface

$f(t)$ = a function of time only

Multiplying through by ρ ,

$$\rho \int \frac{\partial V}{\partial t} ds + \rho \frac{V^2}{2} + P + \rho gh = \rho f(t)$$

The terms of the equation now express pressure. The term $\rho \frac{V^2}{2}$ is the velocity head. The term $\rho \int \frac{\partial V}{\partial t} ds$ is the acceleration head.

The term ρgh is the gravitational head, which varies with the vertical height h of the cross section of blood flow of interest above or below an arbitrary horizontal reference surface.

It is assumed that measurements of blood pressure are made with reference to atmospheric pressure at a reference level where the vasculature attaches to the heart. Some differences in pressure will exist if measurements are compared for horizontal and vertical patients. For instance, a vertical change in the reference or measuring plane of 1 cm will change pressure readings by 0.74 mm Hg.

The blood pressure which must be used in computing ventricular work is the stagnation pressure, not the static pressure. Stagnation pressure is defined as:

$$P_s = P + \frac{\rho V^2}{2} + \int \frac{\partial V}{\partial t} ds$$

Ideally, this pressure should be measured within the ventricle with a Pitot-tube type of pressure probe. Fortunately, the catheters commonly used for pressure measurement are tubes with ends cut perpendicular to the tube axis, and frequently are supplied with a rounded leading edge of the tube wall, they are quite similar to a Pitot tube. Measurements made by these catheters facing upstream will closely approach stagnation pressure.

However, this is true only if the catheter axis is approximately parallel to the blood flow. A catheter at a 90° angle to the flow will more nearly measure static pressure. Arterial catheterization does locate the catheters very nearly parallel to the blood flow, so that the error in stagnation pressure measurement is negligibly small. This may not be true within the ventricle, but the lower velocities and accelerations make static and stagnation pressures nearly identical.

The velocity head and acceleration head have been calculated from the empirical volume-time equation, in the region of peak systolic pressure, which is the pressure required for ventricular work calculation. The velocity head can be from about 4 mm Hg to a maximum of about 12 mm Hg at the highest levels of activity and exercise. Acceleration head depends on the distance of the measurement away from the ventricle. The acceleration head results in a maximum error (decrease) of about 1 mm Hg to 4 mm Hg pressure for each centimeter away from the ventricle, at the highest levels of activity. This estimate ignores the compliance of the artery and the resultant changing pressure-flow profiles, so is meaningful only within a few centimeters of the ventricle.

Experiments at Children's Hospital Medical Center show that catheter measurements of pressure (made with rigid-wall pressure catheters and accurate, high-frequency-response transducers and recorders) anywhere within the ventricle and in the artery proximal to the valve all agree within 5 mm Hg maximum error. This tends to confirm that peak systolic pressures reported in the literature and taken with catheters are stagnation pressure measurements. This is because the stagnation pressure should be nearly the same in these regions (assuming negligible valve pressure drop), while static pressure would differ noticeably due to the difference in velocity head in various parts of the ventricle and the artery.

It can be concluded that catheter measurements are adequate for obtaining the desired ventricle stagnation pressure, that arterial pressure measurements can be used if the valve pressure drop is negligibly small and the catheter is located proximal to the valve, and that the use of peak systolic pressure data reported in the literature and used for heart power calculations in this paper is permissible.

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APPENDIX C

LITHIUM HYDRIDE CONTAINMENT

Lithium hydride is distinguished from other light metal hydrides by its relative stability. The hydrides of sodium, potassium and calcium all decompose more readily, as do the complex boro hydrides of the type $M(BH_4)$. (Beryllium and magnesium do not form well defined hydrides.)¹ Lithium hydride possesses a cubic structure of the NaCl type, containing Li^+ and H^- ions, and is therefore capable of carrying an electric current when molten. Its resistance is about 30Ω cm at the melting point.² On contact with water, it reacts vigorously to produce hydrogen and lithium hydroxide. In moist air it forms a coating of lithium hydroxide and lithium carbonate. The solid material is shipped and handled in metal containers under nitrogen.²

As lithium hydride is heated towards its boiling point, it begins to dissociate, producing lithium metal (melting point $180^\circ C$) and hydrogen. The pressure of hydrogen above lithium hydride at its melting point is about 30 torr, and the heat of vaporization for the compound is 44 ± 2 kcal/mole.³ At room temperature lithium hydride is moderately reactive with atmospheric moisture and slightly reactive with air and oxygen.

In principle, the dissociation could be impeded by providing an atmosphere of hydrogen above the lithium hydride. In practice, this is not straightforward because of the difficulty in containing the hydrogen.

Hydrogen diffuses through metals not as the diatomic molecule⁴ but as single atoms, which are relatively small and highly mobile. If a simple iron envelope were used to contain the lithium hydride-hydrogen

mixture hydrogen would leak out at an unacceptably high rate and the lithium hydride would be converted to lithium metal. This problem has been faced before in particular in the design of Solar boilers for a Brayton-cycle engine for a space power supply named "Sunflower." Much effort was expended in that program on developing materials with low hydrogen permeability.⁵

STATE OF THE-ART CONTAINMENT

Lithium handling and containment at elevated temperatures has been the subject of intensive study for nuclear reactor cooling purposes. Impurities in the lithium appear to be the primary cause of corrosive damage and can be reduced or eliminated by careful purification before use.⁶ A recent survey⁷ reports that tungsten and the tungsten-base alloys W-0.9% Cb, W-15% Mo, W-10% Re, and W-25% Re are not attacked by lithium after 100 and 1000 hours' exposure at 2500°F (1370°C). In higher temperature tests, the W-15% Mo alloy proved most resistant. The Mo alloys TZM (Mo-0.5% Ti-0.1% Zr-0.02%C) and Mo-50% Re were resistant to attack at 2500, 2800, and 3000°F. Cb-1% Zr alloy contained purified lithium without adverse corrosion effects up to 2200°F (1215°C). Another source⁸ indicates that columbium, tantalum and molybdenum have good resistance to lithium corrosion up to at least 1650°F.

A study of lithium hydride containment⁹ reported that at 1500 to 1600°F (820-870°C) Haynes Alloy No. 25 (cobalt base) was attacked in the same extent by lithium and lithium-lithium hydride mixtures, during tests for 2265 hours (~3 months). Similarity of attack was also reported for 316 stainless steel in 858 hours at the same temperature.

The mode of attack found was intergranular or crevice attack to the extent of 0.0005 to 0.0011 inch after 2265 hours at 1600°F (870°C). In these tests the lithium hydride used had a purity of 98.5%.

DISSOCIATION OF LITHIUM HYDRIDE

The equilibrium pressure of hydrogen over lithium hydride is a function of temperature and composition:⁶ over pure lithium hydride it is very high, but 1 to 1.5% of metallic lithium drops the pressure to about 30 torr at 700°C. This pressure is invariant with composition over the range of 26 to 98.5% lithium hydride. It is therefore desirable to assume that the charge will consist of a single-phase solution of 1.5% lithium in 98.5% lithium hydride. It is probable that most thermal property measurements have, in fact, been made on such mixtures.

DESIGN CONSIDERATIONS

A power source for an implantable circulatory support system must contain lithium hydride with an acceptably small loss in thermal storage capacity over an extended period of time without degrading the vacuum foil thermal insulation. Suppose the radioisotope thermal input is 18 watts, so that an acceptable insulation loss is around 1.5 to 2 watts. Since the outer surface will have an area of 500 cm², this implies a flux through the container interface of $2-3 \times 10^{-3}$ W/cm². The presence of hydrogen (and all other gases) in the vacuum foil insulation must be kept below 10⁻⁶ torr in order to preserve its excellent insulation properties. Two questions therefore arise: 1. Is it possible to achieve permeation rate through the lithium hydride container low enough to ensure that a useful amount of lithium hydride for thermal energy storage is available at the end of, say, three years' operation? 2. How can the hydrogen pressure in the insulation space be minimized?

Assume that the degradation of 1 mole or 1.6% of the initial lithium hydride change over three years is acceptable. Such degradation would occur only by the loss of hydrogen through the lithium hydride container wall. Next, the corresponding permeation rate will be calculated and compared with measured rates.

The degradation of 1 mole of lithium hydride releases 1/2 mole of hydrogen, or 11.2 liters of H_2 at NTP. Suppose this release occurs over a period of about three years, or 10^3 days, or 2.4×10^4 hours. Then the corresponding permeation rate is $11,200 / 24,000$ ml (NTP)/hr = 0.47 ml/hr over 500 cm^2 , or about 10^{-3} ml/cm² hr.

The pressure gradient driving this diffusion will be only about 30 torr, while measurements are usually made for a gradient of 750 torr. Since the rate varies as the square root of pressure, for the purposes of comparison with published data this rate can be multiplied by $\sqrt{750/30} \approx 5$, giving 5×10^{-3} ml cm⁻² hr⁻¹. Published data for permeation rates are given in Table 15.1.

TABLE C-1
PERMEATION RATES AT 723°C (1330°F)

<u>Material</u>	<u>Rate</u>	<u>Reference</u>
	cm ³ /cm ² hr ⁻¹	
Tungsten	2.4×10^{-4}	2
304 SS + AO Glass	2.4×10^{-3}	2
Haynes #25 + W-Si	5×10^{-3}	2
Cr-Ni Steel	3×10^{-1}	4
Fe + 4.25% Cr	1.5×10^{-1}	4

Rates are standardized for a 1-mm (0.040-inch) thick membrane with a pressure difference of 1 atmosphere. For purposes of thermal energy storage, it can be concluded that materials with an acceptably low permeation rate have been tested in the laboratory. Two recent developments in refractory metal fabrication will facilitate the construction of practical containers. One of these is the advent of vapor-deposition on a commercial basis, particularly for tungsten. This allows layers of any thickness from 0.001 to 0.25 inch to be deposited on a variety of substrate materials. The other is the electro-deposition process developed by Union Carbide Corporation for coating conducting materials with Nb, Ta, Mo, or W from a molten salt bath.

Each of these processes can be used to fabricate parts of complex shapes without the problems associated with machining brittle materials. Both processes are carried out in the temperature range of 550° to 750°C, which includes the operating range for a lithium hydride TES device, and so problems of thermal expansion mismatch between deposit and substrate will be small.

CONTAINER MATERIAL STUDIES

On the basis of the extensive hydrogen permeability studies reported by Steigerwald,⁵ there are three leading candidates for container materials.

- (a) Tungsten. Uncoated tungsten was reported to have the lowest permeability of any material tested, by a factor of ten. It also has the highest density, by a factor of two. The low permeability is presumably due to the very low solubility of hydrogen (and other

interstitial elements) in tungsten (and Mo and Cr, the other group VI A metals). For reasons associated with this low solubility, tungsten is liable to embrittlement in weld areas at room temperature (though not, under any circumstances, at 700°C), and so a tungsten container closed by a weld may be inconveniently brittle, during installation and handling.

- (b) Haynes #25 Alloy Coated with Tungsten Silicide by a Vapor Pack Process. The W-Si intermetallic layer reduces H_2 permeability by a factor of 50 or so. The base alloy is readily available, reasonably ductile, strong and easily welded.
- (c) AO Smith #3308 Glass. This was tested on a 304 SS substrate, to which the glass was applied by the manufacturer, using a proprietary process. About a thirty-fold reduction in permeability was achieved. The substrate material is probably not critical to the permeation rate, and so the coating could be applied to any substrate with an appropriate expansion coefficient. The stainless steel chosen, #304, is particularly readily available and easily fabricated, though not so strong as the other materials considered.

Because the attractive high temperature properties of tungsten are compromised by its low-temperature brittleness, much work has been devoted to improving this property. It has been found that rhenium in concentrations up to 25% enhances the ductility of tungsten in both

wrought and recrystallized conditions, and that the alloys can be welded much more successfully than can pure W. This effect is achieved without altering the crystal structure of the material, and W and Re are adjacent in the periodic table, so it is probable that the hydrogen permeability of the alloy will be about the same as that of pure tungsten. Although fabrication experience with this alloy is limited, it deserves investigation because it offers the prospect of making a container which is at once thin, strong, and highly impermeable.

The permeation rates reported earlier were for hydrogen provided in gaseous form, without lithium hydride. It is important to establish that metal container surfaces do not themselves enhance the dissociation of lithium hydride, by carrying out permeation experiments using lithium hydride instead of hydrogen. If the metal surface does not alter the dissociation behavior of lithium hydride, these experiments should allow confirmation of the rates tabulated above. Then it is likely that net permeation fluxes will be lower than anticipated because the calculations were based on a membrane area of 500 cm^2 . However, in a real container only a fraction of the area will see hydrogen gas, and the flux may be smaller by a factor of three or four than that calculated. This may allow the use of a more easily fabricated alloy for the container.

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APPENDIX D

IN VITRO HEAT TRANSFER STUDY

This appendix discusses the in vitro experiment equipment, data and interpretation. This experiment used the heat exchangers and Pu-238 sources that are implanted in vivo. The experiment was set up and run in the Sears Surgical Laboratory at Boston City Hospital as part of the joint NHI/AEC Additional Endogenous Heat Studies.

A. NOMENCLATURE

T	temperature	k	thermal conductivity
V	av. velocity	c	specific heat
r	radial coordinate	μ	viscosity
x	axial coordinate	ρ	density
R	tube radius	q	heat flux
D	tube diameter	Q	flow rate
δ	boundary layer thickness	f	friction factor
h	convection coefficient $= q/A (T_w - T_a)$	<u>SUBSCRIPTS</u>	
<u>Re</u>	Reynolds No. $= \rho VD/\mu$	o	entrance
		e	exit
<u>Pr</u>	Prandtl No. $= c\mu/k$	a	mean tube condition at center of tube
<u>Nu</u>	Nusselt No. $= hD/k$	b	bulk properties
z	$= \frac{Re}{Pr} D/L$	m	axial condition
L	tube length	w	wall
A	tube surface area	v	velocity profile
$\dot{m}$	mass flow rate	t	temperature profile

B. ANALYSIS

Although the heat transfer problem for steady flow is well developed, no analysis is available for pulsatile (or transient) conditions. All solutions to the steady-flow heat transfer problem are for the general energy equation with suitable boundary conditions. One boundary condition is constant radial heat flux (q) at the wall. All theories assume velocities in the x -direction only. Turbulent-flow analyses are based primarily on empirical results.

The general energy equation is usually solved by making the following assumptions:

1. The parameters ρ , c and k are constant in the fluid. Generally this is not true, and in fact a correction for variation with temperature has been derived by Sieder and Tate. Since temperature variations are small in the in vitro heat transfer experiment, this assumption is valid for present purposes.
2. Neglect axial conduction. Correct for experiment under consideration in relation to radial heat transfer. The equation can be solved by assuming $\partial T / \partial x$ a constant, as has been done by Holman.
3. Velocity profile is independent of the temperature profile. This is true for high Prandtl numbers. For saline ($Pr = 4.55$) this is questionable, especially for pulsatile flow.

The objective in the analysis of blood flow through a tube with heat transfer is to find the convective heat transfer coefficient, h . This usually means finding Nu. In pulsatile flow this is especially

difficult, since free-convection effects are mixed in with forced convection. For steady flow, $\underline{Nu}$'s have been found for tube flow, based on flow rate and the nature of the flow: whether fully developed or developing, and whether laminar or turbulent. Some of these solutions are:

1. Steady, fully developed laminar, Newtonian:

$$\underline{Nu}_a = 1.86 (z)_L^{1/3} \left(\frac{\mu_a}{\mu_w} \right)^{0.14} \quad (1)$$

assumed = 1 for this study

2. Steady, fully developed turbulent, Newtonian:

$$\underline{Nu}_a = 0.023 (\underline{Re})_a^{0.8} (\underline{Pr})_a^{1/3} \left(\frac{\mu_a}{\mu_w} \right)^{0.14} \quad (2)$$

3. Steady, fully developed turbulent, Newtonian:

$$\underline{Re} \leq 2 \times 10^5$$

$$\underline{Nu} = 0.0395 \underline{Re}^{3/4} \quad (3)$$

4. Steady, fully developed turbulent, Newtonian:

$$\underline{Nu} = 0.023 \underline{Re}^{0.8} \underline{Pr}^{0.4} \quad (4)$$

5. Steady, non-developed, Newtonian, $10 < \frac{L}{D} < 400$:

$$\underline{Nu} = 0.036 \underline{Re}^{0.8} \underline{Pr}^{1/3} \left(\frac{D}{L} \right)^{0.055} \quad (5)$$

6. Steady, non-developed, non-Newtonian:

$$\underline{Nu} = 0.65 x^{1/3} z^{1/3} \quad (6)$$

$$\text{where } x = \frac{6n + 2}{n}$$

m from $\tau = k \dot{\gamma}^n$ (stress - strain)

(n = 1 for Newtonian)

(n $\approx$ 0.8 for blood)

Yields

$$\underline{Nu} = 1.30 z^{1/3} \text{ Saline}$$

$$\underline{Nu} = 1.33 z^{1/3} \text{ Blood}$$

Two effects are of importance: (1) that the heat flux generated by the source is carried away by the fluid, manifested in a bulk temperature rise, and (2) that the $\underline{Nu}$ determines the highest blood temperature at the wall.

This is expressed as:

$$q = \dot{m} c_a (T_{eb} - T_{ob}) \quad (7)$$

$$q = h_a A (T_w - T_a) \quad (8)$$

Expression (7) becomes:

$$q = \rho_a Q C_a (T_{eb} - T_{ob})$$

q is known heat flux

Q is measured flow rate

ρ , c are known fluid properties

T_{ob} is bulk input temperature (Assume uniform input temperature T_o .
According to test conditions, this is probably not the case.)

Expression (8) becomes:

$$q = \frac{k \underline{Nu}}{D} \pi D L (T_w - T_a)$$

where

$$T_a = \frac{T_e + T_o}{2} \text{ measured at center of tube.}$$

or

$$q = k \pi L \underline{Nu} (T_w - T_a)$$

Note that $(T_w - T_a)$ represents maximum temperature rise of the fluid, ΔT_B .

C. IN VITRO EXPERIMENT

A sketch of the experimental set-up for the in vitro heat transfer measurements is shown in Figure D-1. The thermistor placement is illustrated in Figure D-2. A more detailed description of the heat exchanger is given elsewhere.

Assume $T_3 = T_w$, maximum fluid temperature at exit wall

$T_5 = T_o$, inlet temperature

$T_6 = T_e$, exit temperature

To find ΔT_B , find $T_3 - T_5$.

To find $T_{eb} - T_{ob}$, assume $T_{ob} = T_o = T_5$.

To find T_{eb} , knowing T_6 and T_w , assume parabolic profile in boundary layer, as shown in Figure D-3.

For order of magnitude on δ_T ,

$$\delta_T = \delta_v / \underline{Pr}^{1/3}$$

where

$$\frac{\delta_v}{D} = \frac{8}{\phi \underline{Re}} \quad \text{and} \quad \phi = \frac{0.3164}{\underline{Re}^{1/4}}$$

The numerical values of the fluid properties assumed in the reduction of the experimental data are listed in Table D-1.

TABLE D-1
FLUID PROPERTIES AT BODY TEMPERATURE

	Saline	Blood
Density, grams/cc	1.000	1.057
Viscosity, grams/cm-sec	$6.8 \times 10^{-3}^*$	3.4×10^{-2}
Specific Heat, watt-min/gram°C	0.070	0.064
Thermal Conduct., watt/cm°C	$6.27 \times 10^{-3}^*$	$6.27 \times 10^{-3}^*$
<u>Pr</u>	4.55	20.8
<u>Re</u> (Q in L/min)	3280 Q	694 Q
<u>z</u>	2200 Q	2130 Q

* Assumed equal to water

The length for heating, L, was approximately three inches, so that

$$\frac{L}{D} = 8$$

These data were interpreted by evaluating Nu using equations 1 to 6. The Reynolds number, Re, was evaluated for each flow situation to determine whether the flow was laminar, (Re < 2000), turbulent (Re > 3900) or transitional ($2000 \leq \text{Re} \leq 3900$). Equations 1, 5 and 6 were used for laminar flow, and equations 2, 3, 4, 5 and 6 for turbulent flow. All equations were used for the transitional case. After obtaining a number of values for Nu using these equations, a weighted mean value was used to predict the temperature difference, ΔT_B . The

Figure D-1. Experimental Arrangement of In-Vitro Heat Transfer Equipment

Figure D-2. Thermistor Locations for In-Vitro Experiments

VELOCITY PROFILE

TEMPERATURE DISTRIBUTION

Figure D-3. Flow Profile of Coolant through Heat Exchanger

Figure D-4. Exit Profiles from Heat Exchanger

values for Nu were weighed on the basis of the assumptions made for each case and the quality of the data taken by each investigator. The steady-flow characteristics for saline and blood are listed in Tables D-2 and D-3 respectively.

TABLE D-2
STEADY SALINE FLOW CHARACTERISTICS
FOR 24-WATT Pu-238 SOURCE

Flow rate (L/min)	0.6	0.9	1.2	1.5	1.8	2.25
Re	1967	2950	3938	4920	5905	7383
Flow Condition	LAM	TRANS	TURB	TURB	TURB	TURB
Nu by eq 1	20.4	23.4	n.a. *	n.a.	n.a.	n.a.
by eq 2	n.a.	22.7	28.7	34.1	39.8	47.1
by eq 3	n.a.	15.8	19.2	23.2	26.6	31.5
by eq 4	n.a.	25.2	31.8	37.8	44.1	52.2
by eq 5	23.0	31.7	40.1	47.6	55.6	65.8
by eq 6	14.3	16.3	18.0	19.4	20.6	22.1
Nu selected	20	25	30	35	43	51
Predicted ΔT_B ($^{\circ}C$)	8.0	6.4	5.3	4.6	3.7	3.1
Experimental ΔT_B ($^{\circ}C$)	4.4	3.4	2.8	2.5	2.3	2.2

* n.a. indicates not applicable

TABLE D-3

STEADY BLOOD FLOW CHARACTERISTICS
FOR 24-WATT Pu-238 SOURCE

Flow rate (L/min)	0.6	0.9	1.2	1.5	1.8	2.25
<u>Re</u>	416	625	833	1040	1248	1560
Flow Condition	LAM	LAM	LAM	LAM	LAM	LAM
<u>Nu</u> by eq 1	20.1	23.2	25.4	27.4	29.1	31.4
by eq 2	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.
by eq 3	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.
by eq 4	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.
by eq 5	11.0	15.3	19.2	22.7	26.5	31.6
by eq 6	14.4	16.5	18.2	19.6	20.8	22.4
<u>Nu</u> selected	16	19	22	24	27	31
Predicted ΔT_B ($^{\circ}\text{C}$)	10.0	8.4	7.3	6.7	5.9	5.2
Experimental ΔT_B ($^{\circ}\text{C}$)	9.4	8.0	6.9	5.7	4.6	3.6

In all cases, actual ΔT_B is less than predicted.

The estimated thermal boundary layer thicknesses for various fluid and flow-rate conditions are given in Table D-4.

TABLE D-4

THERMAL BOUNDARY LAYER THICKNESS

Flow rate	0.6	0.9	1.2	1.5	1.8	2.25
Saline (inch)	0.032	0.024	0.019	0.016	0.014	0.012
Blood (inch)	0.104	0.076	0.061	0.052	0.045	0.038

Thus for the same T_o , the exit profiles for the same flow rate would be expected to look as shown in Figure D-4.

The temperature profiles of saline relative to blood are not consistently borne out by the experimental data. The reason is probably that the temperature rise in fluid temperature, T_a , is small compared with the thermistor accuracy ($T_6 - T_5 \approx 0.1^\circ\text{C}$).

However, use of parabolic distribution for determining a bulk T_e is justified by the data. What is not borne out is the relative positions of saline and blood profiles.

To find bulk T_{eb} , integrate using the profile shown in Figure D-3. for the case of $T_6 = T_e$.

In the boundary layer,

$$T_{bl} = T_e + \frac{(T_w - T_e)}{\delta_T^2} \left[r - (R - \delta_T) \right]^2$$

$$T_{eb} = \frac{2\pi}{A_{\text{tube}}} \int_0^R T r dr = \frac{2}{R^2} \int_0^{R-\delta_T} T_e r dr + \frac{2}{R^2} \int_{R-\delta_T}^R T_{BL} r dr$$

or

$$T_{eb} = \frac{2}{R^2} \left[\frac{T_e R^2}{2} + \int_{R-\delta_T}^R \left\{ T_e + \frac{T_w - T_e}{\delta_T^2} \left[r - (R - \delta_T) \right]^2 \right\} r dr \right]$$

D. CONCLUSIONS

The maximum rise in fluid temperature through the heat exchanger, ΔT_B , is plotted as a function of flow rate in Figures D-5 and D-6. The conclusions drawn from the in vitro experiment are:

Figure D-5. Differential Coolant Temperature as a Function of Flow Rate

Figure D-6. Differential Coolant Temperature
as a Function of Flow Rate

- (1) Turbulent mixing and pulsatile flow each increase heat transfer to the fluid. The effect of turbulence is illustrated by the increased ΔT_B for blood over saline for the same flow. Since heat transfer depends on Nu, which is in turn a function of z , and z is not a function of μ (thus approximately the same for blood and saline), the difference must be accounted for by the considerably higher Re for saline.

Effect of pulsatile flow to increase heat transfer (and decrease ΔT_B) is obvious for the data.

- (2) Increased flow rates mean increased heat transfer and decreased ΔT_B . For blood flow in vivo at about 6 liters/min, estimated (by extrapolation) ΔT_B is less than 2°C , which is certainly acceptable.

- (3) Increase in ΔT_B due to various factors:

(a) Due to steady instead of pulsatile flow,

For blood,

Q (L/min)	0.6	0.9	1.2	1.5	1.8	2.25
increase ΔT_B (%)	56	70	77	63	44	24

For saline,

Q	0.6	0.9	1.2	1.5	1.8	2.25
+ ΔT_B (%)	38	42	33	39	44	47

(b) Due to blood instead of saline,

For steady flow,

Q	0.6	0.9	1.2	1.5	1.8	2.25
$+\Delta T_B$ (%)	115	135	146	128	100	69

For pulsatile flow,

Q	0.6	0.9	1.2	1.5	1.8	2.25
$+\Delta T_B$ (%)	91	96	86	94	100	100

E. PROBLEM AREAS

Reduction of the experimental data demonstrates a number of ways in which the in vitro measurements could be improved. These ways include:

- (1) Thermistors should be better stabilized before taking readings.
- (2) Heat exchanger tube length, L , should be better defined.
- (3) The theory does not predict the large difference in ΔT_B for a given Q between blood and saline that experiment shows.
- (4) Leakage heat flux has not been determined.
- (5) Data reliability should be improved by taking more measurements more carefully. Occasionally T_6 shows up less than T_5 !
- (6) Due to the pump used, truly steady flow could not be obtained. Reservoir in system would resolve this problem.

- (7) Inlet conditions are not well defined. Only the axial temperature is measured.
- (8) No attempt has been made to evaluate the temperature profile in the fluid, which, along with the velocity profile which is also unknown, is important in evaluating Nu.

APPENDIX E

A detailed design of an implantable steam engine was made in order to better appreciate the practical problems of machining and assembly. This appendix contains the engineering drawings for the miniature engine components.

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REV	DESCRIPTION	DATE	BY	APP

PART	SIZE	REV	REQ	SUB-ASSEMB
4085-1	O		1	HEART ENGINE HOUSING
4085-2	O		1	ENGINE PUMP PISTON R.
4085-3	B		1	BEARING REWORK
4085-4	B		1	CONNECTING ROD REWORK

36	ND	1	SST	SPRING #C1 010 B 2
37	ND	1	SST	SPRING #C1 008 B 2
35	ND	1		O RING SET 1.6 SP
34	ND	14	SST	#4 40 NC 2 NUT
33	ND	20	SST	#4 40 NC 2.5 SS SHCS
32	B	1	304	O RING SPACER
31	ND	2	SST	DOWEL PIN PIC #02 650
30	ND	1	SST	LEE SPRING #C1 010 B 4
29	ND	1	SST	#513 1/2 TRAVEL SNAPRING
28	ND	1	RUBBER	#2 24 O RING
27	B	1	RUBBER	#2 16 O RING
26	B	1	440C	INTAKE VALVE
25	B	1	440C	VALVE STOP
24	C	1	410	STEAM CHEST
23	B	2	304	END CAP
22	B	2	DELORM	VALVE
21	B	2	304	VALVE HOUSING
20	C	1	304	FLYWHEEL
19	B	1	440C	CRANKSHAFT
18	C	1	304	CONNECTING ROD
17	B	1	440C	WRIST PIN
16	B	1	GRAP	LINER
15	B	1	440C	VALVE
14	B	1	304	EXHAUST VALVE NUT
13	B	1	304	PISTON RING SPACER
12	B	2	GRAP	PISTON RING
11	C	1	304	ENGINE PISTON (UPPER)
10	C	1	304	ENGINE PISTON (LOWER)
9	C	1	304	PISTON CONNECTOR
8	C	1	304	HYDRAULIC PISTON
7	C	1	T1	UPPER HOUSING
6	B	2	GRAP	BEARING
5	C	1	T1	MAIN HOUSING END CAP
4	C	1	T1	MAIN HOUSING
3	B	1	T1	END CAP
2	B	1	GRAP	HYDRAULIC PUMP LINER
1	D	1	T1	HYDRAULIC PUMP HOUSING

PART SIZE REV REQ		MATERIAL		NOTES	
CHECKED: [Signature] DATE: 1/1/00					
DRAWN: [Signature] DATE: 1/1/00					
SCALE: 1/2" = 1"					
TITLE: HEART POWER SUPPLY					
4085					

NOTE
1. DIA A-B'D TO BE CONCENTRIC TO EACH
OTHER WITHIN .0005 TIR & PERPENDICULAR
TO SURFACE F WITHIN .0005 TIR
2. SURFACE 'C' TO BE PERPENDICULAR
TO DIA 'A' WITHIN .0005 TIR

[illegible]

- NOTES:
1. SLOT TO BE CENTRALLY LOCATED 180° FROM .125 WIDE SLOT AND PARALLEL TO AXIS OF .125 DIA
 2. BRAZE ASSEMBLY AS INDICATED
 3. CHROME PLATE .004 .005 THICK ON SURFACES INDICATED
 4. MACHINE AS INDICATED
 5. DIA B TO BE CONCENTRIC WITH DIA B' WITHIN .0003 TIR

		UNLESS OTHERWISE SPECIFIED		THERMO ELECTRON 1000 UNIVERSITY AVENUE PITTSBURGH, PA 15206	
		DECIMAL 1 4 0 8 . X 1 5 2 0 1 XXX 4 0 0 8 17 1/2 RADIAL AND 1/2 INCH THREADS CLASS 2 1/4 DIA GROUND OTHER TIR ALL OTHER ON CHAMFERED SURFACES TO BE REMOVED WITH SURFACES TO BE REMOVED WITH 1/2 DIA ANGULAR TO 0.0005 DIA ALL OTHER ANGULAR TO 0.0005 DIA REMOVE ALL BURRS AND BORE 3/4		DRAWING CHECKED ENGINEER 101 22 4 TITLE ENGINE & PUMP PISTON ASSEMBLY	
MAT'L		REQ	SCALE 5 X 1	FINISH	PLM
SUPPLIED		1	NEXT ASY	SIMILAR TO	
					SIZE QUANTITY NUMBER C/F D4 4085-2

2-7

NO.	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

C-4085-5

B-4085-6

.1266

.0004

.0000

Ø

UNLESS OTHERWISE SPECIFIED	
DECIMAL: X ± .03, XX ± .01, XXX ± .005	
FILLET RADIUS: .010 MAX	
THREADS: CLASS 2	
H DIA. CONC. WITHIN T.I.R.	
ALL OTHER DIA. CONC. WITHIN .005 T.I.R.	
SURFACES TO BE PER. WITHIN .005 T.I.R.	
SURFACES TO BE PAR WITHIN .005 T.I.R.	
HOLE ANGULAR TOL. 0-2.0 RAD. ± 0° 5'	
2.0-4.0 RAD. ± 0° 5'	
ALL OTHER ANGULAR TOL. TO BE ± 0° 30'	
REMOVE ALL BURRS AND EDGES	

MAT'L	REQ
SUP. L.E.	1

SCALE	FINISH	R.M.S.
5X1	✓	

SIZE	DRAWING NUMBER	SHEET	REV
B	4085-3		

THERMO ELECTRON			
140 THURGOOD STREET, WALTHAM, MASS. 02154			
DRAWN	CHECKED	ENGINEER	
W 11-24-67			

TITLE	
BEARING FOR R-	

NO.	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

DIA 'B'

+ .0004
- .0000

.1266

B 4085-6

C 4085-18

B-4085 16

DIA A

+ .0004
- .0000

.1266

DIA A TO BE PARALELL WITH DIA B WITHIN .0003 TIR.

UNLESS OTHERWISE SPECIFIED		THERMO ELECTRON 125 REED ST. CARPENTERS 88 FIRST AVENUE WALTHAM MASS 02154	
DECIMAL: X ± .03, XX ± .01, XXX ± .005		DRAWN	CHECKED
FILLET RADIUS .010 MAX		8/11/24-67	ENGINEER
THREADS CLASS 2		TITLE	
* DIA CONC WITHIN .005 TIR		CONNECTING ROD	
ALL OTHER DIA CONC WITHIN .005 TIR		REWORK	
SURFACES TO BE PERP WITHIN .005 TIR		SIZE	DRAWING NUMBER
SURFACES TO BE PAR WITHIN .005 TIR		B	4085-4
HOLE ANGULAR TOL 0-20 RAD ± 0° 3'		SHEET	REV
20-40 RAD ± 0° 3'			
ALL OTHER ANGULAR TOL TO BE ± 0° 30'			
REMOVE ALL BURRS AND EDGES			
SCALE: 5X1	FINISH: CA R.M.S.		
MAT'L	REQ		
SUPPLIED	1		
NEXT ASSY	SIMILAR TO

DESIGNED BY: J. L. D. / K. D. S. 008 CHECKED BY: J. L. D. / K. D. S. 008 DATE: 10/10/00 ALL OTHER DIMENSIONS TO BE 10% OF REMOVE ALL DIMENSIONS AND TOLERANCES		TITLE HYDRAULIC PISTON HOUSING	
MP/L T1 180AV	S/S 1	SCALE 5X1 IN. TO 1/25" DIA	DRAWN TO D 4085

E-10

NO.	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

0.15 X 45° CHAM

4550 $+0.0005$
 -0.0005

373 DIA $+0.000$
 -0.005

710

4/

UNLESS OTHERWISE SPECIFIED		THERMO ELECTRON 1001 21ST ST. LYNN, MASS 01901 85 FIRST AVENUE WALTHAM, MASS 02154	
DECIMAL .X ± .03, XX ± .01, XXX ± .005 FILLET RADIUS .010 MAX THREADS CLASS 2 # DIA CONC WITHIN TIR ALL OTHER DIA. CONC WITHIN .005 TIR SURFACES TO BE PAR WITHIN .005 TIR HOLE ANGULAR TOL 0-20 RAD ± 0° 30' 20-40 RAD ± 0° 30' ALL OTHER ANGULAR TOL TO BE ± 0° 30' REMOVE ALL BURRS AND EDGES		DRAWN <u>W511367</u> CHECKED _____ ENGINEER _____	
SCALE 5X1 FINISH: 32 RMS.		TITLE HYDRAULIC PUMP LINER	
MAT'L GRAPHITE	REQ 1	SIZE B	DRAWING NUMBER 4085
		SHEET 2	REV 1

E-11

NO.	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

1.100 DIA

O62

1/8 DR (.125 DIA) 6 HOLES
EQUI SP ON A .900 DBC

UNLESS OTHERWISE SPECIFIED		 THERMO ELECTRON <small>ENGINEER & CORPORATION</small> <small>85 FIRST AVENUE WALTHAM MASS 02154</small>																			
DECIMAL: .X ± .03, .XX ± .01, .XXX ± .005 FILLET RADIUS .010 MAX THREADS: CLASS 2 H/DIA CONC. WITHIN .T.I.R. ALL OTHER DIA CONC. WITHIN .005 SURFACES TO BE PERP WITHIN .005 SURFACES TO BE PAR WITHIN .005 HOLE ANGULAR TOL. 0-2.0 RAD ± 0° 5' 2.0-4.0 RAD ± 0° 5' ALL OTHER ANGULAR TOL TO BE ± 0° 30' REMOVE ALL BURRS AND EDGES		<table border="1" style="width: 100%;"> <tr> <td style="width: 50%;">DRAWN</td> <td style="width: 50%;">CHECKED</td> <td style="width: 50%;">ENGINEER</td> </tr> <tr> <td>1/3 11-13-67</td> <td> </td> <td> </td> </tr> </table>		DRAWN	CHECKED	ENGINEER	1/3 11-13-67														
DRAWN	CHECKED	ENGINEER																			
1/3 11-13-67																					
<table border="1" style="width: 100%;"> <tr> <td style="width: 50%;">SCALE: 2.5 X 1</td> <td style="width: 50%;">FINISH: 125 R.M.S.</td> </tr> <tr> <td> </td> <td> </td> </tr> <tr> <td>NEXT ASSY</td> <td>SIMILAR TO</td> </tr> </table>		SCALE: 2.5 X 1	FINISH: 125 R.M.S.			NEXT ASSY	SIMILAR TO	<table border="1" style="width: 100%;"> <tr> <td colspan="3" style="text-align: center;">TITLE</td> </tr> <tr> <td colspan="3" style="text-align: center;">END CAP</td> </tr> <tr> <td style="width: 33%;">SIZE</td> <td style="width: 33%;">DRAWING NUMBER</td> <td style="width: 33%;">SHEET REV</td> </tr> <tr> <td>B</td> <td>4085</td> <td>3</td> </tr> </table>		TITLE			END CAP			SIZE	DRAWING NUMBER	SHEET REV	B	4085	3
SCALE: 2.5 X 1	FINISH: 125 R.M.S.																				
NEXT ASSY	SIMILAR TO																				
TITLE																					
END CAP																					
SIZE	DRAWING NUMBER	SHEET REV																			
B	4085	3																			

MAT'L	REQ
T1 120AV	1

NO.	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

Technical drawing of a bearing. The side view shows a cylindrical part with a diameter of .250. The length is .212. The top surface has a thickness of .124 with a tolerance of +.000/-0.002. The bottom surface has a thickness of .16 with a tolerance of +.0000/-0.0005. A chamfer of .015 x 45° is indicated. The end view shows a circular feature with a crosshair.

UNLESS OTHERWISE SPECIFIED	
DECIMAL: .XX ± .03, .XX ± .01, .XXX ± .005	
FILLET RADIUS: .010 MAX	
THREADS: CLASS 2	
H DIA. CONC. WITHIN TIR	
ALL OTHER DIA. CONC. WITHIN .005 TIR	
SURFACES TO BE PAR. WITHIN .005 TIR	
SURFACES TO BE PAR. WITHIN .005 TIR	
HOLE ANGULAR TOL. 0-2.0 RAD. ± 0° 3'	
2.0-4.0 RAD. ± 0° 5'	
ALL OTHER ANGULAR TOL. TO BE ± 0° 30'	
REMOVE ALL BURRS AND EDGES	

THERMO ELECTRON	
300 FIRST AVENUE, WALTHAM, MASS. 02154	
DRAWN	CHECKED
11-14-67	
TITLE: BEARING	
SIZE	DRAWING NUMBER
B	4085
SHEET	REV
6	

MAT'L	REQ
GRAPHITE	2

SCALE	FINISH
5X1	R.M.S.
NEXT ASSY	SIMILAR TO

E-16

E-17

E-18

HOLE	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

UNLESS OTHERWISE SPECIFIED		THERMO ELECTRON ENGINEERING CORPORATION 80 FIRST AVENUE - WALTHAM, MASS. 02154	
DECIMAL: .X ± .03, .XX ± .01, .XXX ± .005 FILLET RADIUS: .010 MAX THREADS: CLASS 2 H/DIA. CONC. WITHIN T.I.R. ALL OTHER DIA. CONC. WITHIN .005 T.I.R. SURFACES TO BE PAR. WITHIN .005 T.I.R. SURFACES TO BE PAR. WITHIN .005 T.I.R. HOLE ANGULAR TOL. 0-2.0 RAD ± 0° 5' 2.0-4.0 RAD ± 0° 5' ALL OTHER ANGULAR TOL. TO BE ± 0° 30' REMOVE ALL BURRS AND EDGES		DRAWN: 11-22-67 CHECKED: [] ENGINEER: [] TITLE: PISTON RING	
SCALE: 5X1 FINISH: 125 R.M.S.		SIZE: 8 DRAWING NUMBER: 4085 SHEET: 12 REV: []	

MAT'L	REQ
GRAPHITE	2

SCALE	FINISH
5X1	125 R.M.S.

SIZE	DRAWING NUMBER	SHEET	REV
8	4085	12	[]

E-21

HOLE	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

UNLESS OTHERWISE SPECIFIED		 THERMO ELECTRON <small>140 STEELE ST. CROFTON, MD. 88 FIRST AVENUE WALTHAM, MASS 02154</small>	
DECIMAL: X ± .03, XX ± .01, XXX ± .005 FILLET RADIUS .010 MAX THREADS: CLASS 2 * DIA CONC WITHIN .005 TIR ALL OTHER DIA CONC WITHIN .005 TIR SURFACES TO BE PER WITHIN .005 TIR SURFACES TO BE PAR WITHIN .005 TIR HOLE ANGULAR TOL 0-2.0 RAD ± 0° 5' 2 0-4.0 RAD ± 0° 5' ALL OTHER ANGULAR TOL TO BE ± 0° 30' REMOVE ALL BURRS AND EDGES		DRAWN: 11 22 67 CHECKED: _____ ENGINEER: _____	
SCALE: 5X1 FINISH: 125 R.M.S. NEXT ASSY: _____ SIMILAR TO: _____		TITLE: PISTON RING SPACER SIZE: B DRAWING NUMBER: 4085 SHEET: 13 REV: _____	

MAT'L	REQ
304 ST ST	1

E-22

NO.	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

DR & TAP FOR 3-48 NC-3B

UNLESS OTHERWISE SPECIFIED		 THERMO ELECTRON <small>ENGINEERING CORPORATION</small> <small>25 FIRST AVENUE - WALTHAM, MASS. 02154</small>							
DECIMAL: .X ± .03, .XX ± .01, .XXX ± .005 FILLET RADIUS: .010 MAX THREADS: CLASS 2 H DIA. CONC. WITHIN T.I.R. ALL OTHER DIA. CONC. WITHIN .005 T.I.R. SURFACES TO BE PER. WITHIN .005 T.I.R. SURFACES TO BE PAR. WITHIN .005 T.I.R. HOLE ANGULAR TOL. 0-2.0 RAD ± 0° 5' 2.0-4.0 RAD ± 0° 5' ALL OTHER ANGULAR TOL. TO BE ± 0° 30' REMOVE ALL BURRS AND EDGES		<table border="1" style="width: 100%;"> <tr> <td>DRAWN</td> <td>CHECKED</td> <td>ENGINEER</td> </tr> <tr> <td>S 11-22-67</td> <td> </td> <td> </td> </tr> </table>		DRAWN	CHECKED	ENGINEER	S 11-22-67		
DRAWN	CHECKED	ENGINEER							
S 11-22-67									
SCALE: 5X1 FINISH: 125 R.M.S. NEXT ASSY: SIMILAR TO:		TITLE: <div style="text-align: center; font-weight: bold;">EXHAUST VALVE NUT</div>							
MAT'L	REQ	SIZE	DRAWING NUMBER SHEET REV						
304 ST ST	1	B	4085 14						

E-23

NO.	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

$\begin{matrix} +.0000 \\ -.0004 \\ .1260 \end{matrix}$
 $\begin{matrix} +.000 \\ -.005 \\ .350 \end{matrix}$

UNLESS OTHERWISE SPECIFIED		THERMO ELECTRON 145 HYPERION CORP. 200 RD. FOST AVENUE WALTHAM, MASS. 02154	
DECIMAL: .X ± .03, XX ± .01, XXX ± .005 FILLET RADIUS .010 MAX THREADS: CLASS 2 DIA CONC. WITHIN .005 TIR ALL OTHER DIA. CONC. WITHIN .005 TIR SURFACES TO BE PERP. WITHIN .005 TIR SURFACES TO BE PAR. WITHIN .005 TIR HOLE ANGULAR TOL. 0-2.0 RAD ± 0° 5' 2.0-4.0 RAD ± 0° 5' ALL OTHER ANGULAR TOL. TO BE ± 0° 30' REMOVE ALL BURRS AND EDGES		DRAWN: 11-22-67 CHECKED: _____ ENGINEER: _____ TITLE: WRIST PIN	
MAT'L	REQ	SCALE: 5X1	FINISH: R.M.S.
440 C ST ST	1	NEXT ASSY	SIMILAR TO
		SIZE	DRAWING NUMBER
		8	4085-
		SHEET	REV
		17

HOLE	DESCRIPTION	DBC	NO OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

UNLESS OTHERWISE SPECIFIED		THERMO ELECTRON ENGINEERING CORPORATION 88 FIRST AVENUE WALTHAM MASS 02154	
DECIMAL: .XX ± .03, .XX ± .01, .XXX ± .005 FILLET RADIUS .010 MAX THREADS: CLASS 2 *DIA. CONC. WITHIN TIR ALL OTHER DIA. CONC. WITHIN .005 TIR SURFACES TO BE PAR WITHIN .005 TIR SURFACES TO BE PAR WITHIN .005 TIR HOLE ANGULAR TOL. 0-2.0 RAD ± 0° 30' 2.0-4.0 RAD ± 0° 30' ALL OTHER ANGULAR TOL. TO BE ± 0° 30' REMOVE ALL BURRS AND EDGES		DRAWN 11-24-67 CHECKED ENGINEER	
SCALE: 5X1 FINISH: 125 R.M.S. NEXT ASSY SIMILAR TO		TITLE VALVE SIZE: 8 DRAWING NUMBER: 4085 SHEET: 22 REV	

MAT'L	REQ
DELFIN	2

E-31

HOLE	DESCRIPTION	DBC	NO OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

32 (#30 DRILL) .128

0.62

0.255

0.09

0.015

0.13

UNLESS OTHERWISE SPECIFIED

DECIMAL. X ± .03, XX ± .01, XXX ± .005

FILLET RADIUS .010 MAX

THREADS CLASS 2

* DIA CONC WITHIN TIR

ALL OTHER DIA CONC WITHIN .005 TIR

SURFACES TO BE PERP WITHIN .005 TIR

HOLE ANGULAR TOL 0-2.0 RAD ± 0° 3'

2 0-4.0 RAD ± 0° 3'

ALL OTHER ANGULAR TOL TO BE ± 0° 30'

REMOVE ALL BURRS AND EDGES

SCALE 5X1

FINISH 12S RMS

MAT'L REQ

304 ST ST 2

NEXT ASSY

SIMILAR TO

SIZE DRAWING NUMBER SHEET REV

B 4085 23

THERMO ELECTRON

85 FRB AVEN E WA HAM MASS 02 54

DRAWN CHECKED ENGINEER

5/12/67

TITLE

END CAP

E-33

[illegible]

E-34

NO.	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

UNLESS OTHERWISE SPECIFIED	
DECIMAL: .XX ± .03, .XXX ± .01, .XXXX ± .005 FILLET RADIUS: .010 MAX THREADS: CLASS 2 * DIA. CONC. WITHIN T.I.R. ALL OTHER DIA. CONC. WITHIN .005 T.I.R. SURFACES TO BE PAR. WITHIN .005 T.I.R. HOLE ANGULAR TOL. 0-2.0 RAD. ± 0° 5' 2.0-4.0 RAD. ± 0° 5' ALL OTHER ANGULAR TOL. TO BE ± 0° 30' REMOVE ALL BURRS AND EDGES	
SCALE: 5X1	FINISH: 125 R.M.S.
NEXT ASSY	SIMILAR TO

THERMO ELECTRON 120 TOTTENHAM ST. WALTHAM, MASS. 02154		
DRAWN	CHECKED	ENGINEER
11-21-67		
TITLE: VALVE STOP		
SIZE	DRAWING NUMBER	SHEET REV
8	4085	26

MAT'L	REQ
440 C ST ST	1

E-35

NO.	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

UNLESS OTHERWISE SPECIFIED	
DECIMAL: .X ± .03, .XX ± .01, .XXX ± .005	
FILLET RADIUS .010 MAX	
THREADS: CLASS 2	
H DIA CONC. WITHIN T.I.R.	
ALL OTHER DIA. CONC. WITHIN .005 T.I.R.	
SURFACES TO BE PERP. WITHIN .005 T.I.R.	
SURFACES TO BE PAR. WITHIN .005 T.I.R.	
HOLE ANGULAR TOL. 0-2.0 RAD ± 0° 3'	
2.0-4.0 RAD ± 0° 5'	
ALL OTHER ANGULAR TOL. TO BE ± 0° 30'	
REMOVE ALL BURRS AND EDGES	

THERMO ELECTRON	
140 BEECH ST. CORPORATION ST.	
85 FIRST AVENUE WALTHAM, MASS. 02154	
DRAWN	CHECKED
11 24 67	
TITLE	
INTAKE VALVE	
SIZE	DRAWING NUMBER
B	4085
SHEET	REV
27	

MAT'L	REQ
440 C ST	1

SCALE	FINISH
5X1	R.M.S.
NEXT ASSY	SIMILAR TO

NO.	DESCRIPTION	DBC	NO. OF HOLES	REV	DESCRIPTION	DATE	BY	APPR

1.10 DIA

$\begin{matrix} +.002 \\ -.000 \end{matrix}$
.020

$\begin{matrix} +.005 \\ -.000 \end{matrix}$
.694

1.25 DIA THRU 6 HOLES
EQUI SP ON A 900 DBC

UNLESS OTHERWISE SPECIFIED		 THERMO ELECTRON 125 REED ST. WALTHAM, MASS 01981	
DECIMAL: .X ± .03, .XX ± .01, .XXX ± .005 FILLET RADIUS .010 MAX THREADS: CLASS 2 H DIA CONC WITHIN .010 TIR ALL OTHER DIA CONC WITHIN .005 TIR SURFACES TO BE PER WITHIN .005 TIR SURFACES TO BE PER WITHIN .005 TIR HOLE ANGULAR TOL 0-2.0 RAD ± 10° S 2 0-4.0 RAD ± 10° S ALL OTHER ANGULAR TOL TO BE ± 0° 30' REMOVE ALL BURRS AND EDGES		DRAWN: 11/29/67 CHECKED: ENGINEER: TITLE: 'O'RING SPACER SIZE: B DRAWING NUMBER: 4085 SHEET: 33	
MAT'L	REQ	SCALE: 2.5 X 1	FINISH: 125 R.M.S.
304 ST ST	1	NEXT ASSY	SIMILAR TO

APPENDIX F

DIRECT-ACTING CONTROL SYSTEM

This concept for an implantable assisted circulation system consists of an engine directly connected to a hydraulic pump, as shown in the block diagram of Figure F-1. The inflow and outflow lines from the hydraulic pump are connected to the assist pump in such a way that both operate in direct synchrony. Therefore, each power stroke of the engine results in a systole in the pump. Means are provided in the pump to realize a mechanical advantage so as to minimize the mass of reciprocating fluid in the lines connecting the power source with the blood pumps. Means are also provided in the blood pump to permit variable stroke volume on a beat-by-beat basis, so as to exactly match the stroke volume of the left ventricle. In addition, adjustment can be made of the back pressure against which the left ventricle must pump.

The control system is best described by following through a complete cycle. Starting at top dead center, the intake valve opens and a pressure of 400 psia is applied to the engine piston which begins to accelerate downward. Since the hydraulic pump piston is directly coupled to the engine piston, it too accelerates downward, rapidly raising the pressure in the hydraulic system. As soon as the pressure in the hydraulic system exceeds 20 to 30 psia, the clutch (connecting the flywheel with the engine) locks. Since the engine is accelerating, the flywheel also accelerates. This loads the engine during its period of greatest output and, hence, greatest period of acceleration. As the piston travels downward, the intake valve soon closes, and near isentropic expansion begins, gradually decreasing the pressure in the cylinder and decreasing the accelerating

Figure F-1. Diagram of Direct-Acting Control System

force acting on the piston. Meanwhile, the piston in the blood pump has begun its displacement, which, in turn, actuates the blood pump and ejects blood. The aortic pressure quickly reaches its peak value of 120 to 150 mm of Hg. Likewise, the hydraulic fluid has reached its peak value. As the engine piston advances, the pressure in the cylinder finally decreases to the point where the force of the steam on the piston is less than the restraining force due to the hydraulic pressure, which, in turn, is determined by the blood pressure. The piston would begin to decelerate and would be in danger of stalling (depending on the aortic pressure) if it were not for the inertia of the flywheel. Therefore, the flywheel carries the piston through bottom dead center regardless of aortic pressure, making the blood pump a positive-displacement pump. This is important if the physiologic automatic controls are relied upon to regulate cardiac output.

Once the piston passes through bottom dead center, the hydraulic pressure immediately falls and unlocks the clutch of the flywheel. The flywheel, therefore, continues to rotate and drive the feedwater pump. The engine piston, hydraulic piston, and blood pump piston all remain stationary at bottom dead center until the left ventricle ejects blood into the pump. The blood at the pressure level achieved by the left ventricle's systole returns the three pistons toward top dead center. The return flow of the hydraulic fluid passes through a branch line containing a Venturi tube. A pressure differential develops, depending on the flow velocity, which is determined by the pressure achieved by the left ventricle. A higher ventricular pressure causes a higher flow velocity which, in turn, causes a greater differential pressure. A greater pressure differential causes a higher pressure setting for the boiler

pressure regulator. Also, the higher ventricular pressure returns the piston faster to accommodate a higher beat rate. An increased stroke volume in the left ventricle generally causes a greater strength of contraction, and hence a higher peak pressure. This will signal the boiler to increase the pressure so that the next stroke will be faster and can add more energy to the flywheel to accommodate higher peak pressure.

The feedwater pump will be driven by the flywheel at a rate approximately ten times the engine speed. Therefore, following an increased pressure setting on the boiler pressure regulator, the feedwater pump will be able to complete three or four full strokes to raise the boiler pressure before the next power stroke of the engine.

The concept for this controller is shown schematically in Figure F-2. The pressure differential developed by the return flow acts across a diaphragm, increasing deflection with increasing return flow. This deflection causes a higher pressure threshold on the boiler regulator before fluid is bypassed around the feedwater pump instead of being pumped into the boiler.

As previously discussed, the blood pump has a variable stroke volume so as to match the left ventricle on a beat-by-beat basis. This is accomplished by means of a pressurizer on the blood pump. The operation can be visualized by following through one complete cycle. Assume the pump has just completed ejecting blood (completed pump systole). The bladder and hydraulic actuator remain stationary (at bottom dead center) until the left ventricle contracts and ejects blood into the pump. This is accomplished by disconnecting the flywheel at bottom dead center and, secondly, by ensuring that the opening pressure on the outflow valve from the hydraulic actuator is

Figure F-2. Boiler Pressure Regulator

greater than the pressure exerted by the pressurizer. When the left ventricle ejects blood into the pump (which requires a pressure in excess of the pre-set pressurizer level), this causes the bladder and hydraulic actuator (hence, hydraulic pump and engine pistons) to be accelerated toward top dead center. The stronger the contraction of the left ventricle, the greater will be the pressure in the pump and the acceleration of the bladder and actuator. The stroke volume of the bladder will be intentionally designed to be slightly greater than the maximum stroke volume of the left ventricle. Therefore, the piston will not be returned completely to top dead center by the ejection of blood from the left ventricle. When the ventricle has completed systole and begins to relax, the inflow valve will close and the intra-pump pressure will tend to fall because of the inertia of the system (pistons, fluid, etc.). As the pressure falls below the pre-set threshold of the spring-loaded bellows in the pressurizer, the pressurizer piston will begin to move inward, compressing the bladder and maintaining the intra-pump pressure, and thus continuing the travel of the hydraulic actuator, hydraulic pump piston and engine piston toward top dead center. The stroking range of the pressurizer is greater than the variation from maximum stroke volume of the left ventricle. At top dead center, the engine intake valve will open and the engine will begin a power stroke, as before. The combination of engine and flywheel will guarantee a full stroke each cycle, thus ejecting all of the blood above a fixed residual inventory, regardless of peak systolic pressure. The initial travel of the hydraulic actuator will first re-set the pressurizer against the stops before ejecting blood into the aorta.

There are several objections to the direct-acting control system. First, the system is conceptually and mechanically complex. Second,

the large flywheel energy storage would require a speed increaser and/ or a heavy flywheel. In addition, the associated gyroscopic moments would certainly distress the host. For these reasons the control system described in this appendix was dropped from consideration.

APPENDIX G

CONTROLS FOR A LEFT VENTRICULAR ASSIST DEVICE

This appendix contains the report submitted by Whiteley Hydraulics, Inc., which summarizes the results of a study of blood pump control and power source coupling.

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APPENDIX G

CONTROLS FOR A LEFT VENTRICULAR ASSIST DEVICE TO BE USED WITH AN IMPLANTABLE POWER SUPPLY

By
Whiteley Hydraulics, Inc.

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I	Objective
II	Conclusions
III	Recommendations
IV	Background
V	Assist Device Function
VI	Assist Devices
	VI-1 Counterpulse and In-Phase
	VI-2 Dissipative Device - Throttling
	VI-3 Dissipative Device - Follower
	VI-4 Variable Flow Device
	VI-5 Direct Engine to Blood Pump Coupling
	VI-6 Variable Pressure Reservoir
VII	Advantages and Disadvantages of Follower Device
VIII	Appendix
	G-1 Requirements
	G-2 Efficiency Calculations

VIII APPENDIXES

The following pages contain supplementary information related to this report as follows:

Appendix G-1 - Requirements

Appendix G-2 - Efficiency Calculations

I OBJECTIVE

This report summarizes the work performed by Whiteley Hydraulics, Inc., as part of a conceptual study of the arrangement and control of a heart assist device for the Thermo Electron Corporation of Waltham, Massachusetts. The objective of the study was to define the control requirements and functions, based on verbal information, and written literature supplied by Thermo Electron; to consider several control system concepts; and to evaluate these with respect to the specified requirements.

II CONCLUSIONS

- 1) All of the requirements and functions (see Appendix G-1) for the left ventricle assist device can be met.
- 2) A constant pressure reservoir system offers the maximum engine to hydraulic system decoupling and affords the maximum flexibility for implanting location and suspension.
- 3) A single acting one - to - one follower device is the simplest and most physiological device considered. The left ventricle pressure-time relationship will be scheduled while the corresponding relationship in the systemic will be exactly physiological. (Refer to Section VI)
- 4) The follower system efficiency over its operating range can be made quite acceptable, by allowing the heart to assist it at the higher peak systemic

pressure conditions. By allowing the heart to assist the device at these conditions the power supply can be reduced in size.

- 5) Because of the independence of the power supply, and controls, the follower device can be easily studied and evaluated.
- 6) A variable flow system offers good potential for high hydraulic system efficiencies. However, this is at the expense of simplicity and reliability.

III RECOMMENDATIONS

The constant pressure reservoir, single acting follower device is recommended to meet the objectives previously set forth. It should be fully evaluated mechanically and physiologically prior to a detailed design. This would include a careful investigation into the feasibility of such a device as related to the arrangement and component design. The areas, spring constants, etc., should be sized and the system first evaluated on an analog computer. This will permit a complete investigation of the physiological response to simulated inputs, effects of frictional forces, stall conditions, transients, etc.

In addition to the above the following recommendations are also made:

- 1) A detailed evaluation should be made on optimizing the efficiency of the follower assist device, part-

icularly considering the effect of (a) regulated supply pressure variations (droop) and (b) allowing the actuator to stall out at a systemic peak pressure less than the 180 mm. Hg. maximum causing the heart to assist it under these conditions.

- 2) The control for the power supply should be studied further, with a view to overall system optimization.

IV BACKGROUND

Under certain circumstances it will become necessary to reduce the heart work load of an individual. Thermo Electron is working on an implantable assist device and power supply for this purpose. The approach being undertaken by Thermo Electron Corp. is to locate a hydraulically operated assist device between the apex of the left ventricle and the aorta. The power supply for this device would be located remotely in the abdominal cavity.

Since approximately 87% of the heart's work is done by the left ventricle, then the assist device need only reduce the work on this side. The difference in the work done by the right ventricle versus the left is essentially directly related to the peak pressures reflected in the respective sides of the heart. The peak pressure on the left side is approximately five times that of the right side.

The function of a left ventricle assist device is to pulse the blood from the heart to the aorta while permitting the heart to work at a lower pressure. The assist device must, therefore, be synchronized with the heart on a beat-by-beat basis.

The study undertaken by Whiteley Hydraulics was to define the control requirements of the assist device and to consider several control system concepts. This was to be done with careful consideration given to reliability, long life, efficiency and compatability.

Prior to the start of the study, meetings were held between members of Whiteley Hydraulics and Thermo Electron. The purpose was to define the scope of the study and to familiarize Whiteley Hydraulics with the total assist system and the physiological aspects of the cardiovascular system. Reports related to these discussions were made available. Thermo Electron also discussed the assist device concepts they had considered.

With this background information presented for study, the specific requirements (see Appendix G-1) of the assist device were formulated with particular emphasis placed on the physiological needs.

First considered, was a constant pressure counterpulse device. To meet the varying systemic pressure demands this device requires throttling, as well as means to trigger various mechanisms on essentially zero error signals (Refer to Section VI - 1 and VI - 2). This system would be relatively inefficient.

A variable flow system approach would be physiological and efficient. By varying the stroke of a positive displacement pump, the flow to the device would be made to be directly related to the cardiac output and the pressure would rise to

whatever was necessary to overcome systemic resistance. This approach, however, requires added complexity in the engine and hydraulic pump. (Refer to Section VI - 4).

Several devices, direct coupled to the power supply engine, pulsing on a beat -by- beat basis were considered. They, however, presented problems particularly with respect to the complexity of the required engine controls and synchronization with the biological heart.

To this point, all the devices considered required the heart to beat fully before the assist device would pulse. The signals available to operate the required mechanisms were small or required complex components. If the assist device could be pulsed at the same time as the heart beat, a simpler system could be devised.

This led to the follower system. This assist device pulses at the same time as the heart beats. It is a throttling device, but, the use of a double chambered "sac" permits the design of a simpler and satisfactory efficient system. (Refer to Section VI - 3).

V ASSIST DEVICE FUNCTION

The function of the left ventricle assist device is to perform the duties of the heart, in as physiological a manner as possible. It must allow the natural heart to perform unimpeded, while providing a means for the heart to pulse at a reduced pressure level.

The ideal system would provide for accepting varying stroke

volumes from the left ventricle and then pulsing this volume in a manner to exactly duplicate the pressure time and volume time relationships of the unassisted heart on a beat - by - beat basis.

Figure 1 shows the desired relationship of the time rate of change of pressure and volume of the assisted heart and the assist device. The respective curves are shown displaced for clarity, however, they could occur at any time between heart beats, providing no left ventricle back pressure or flow blockage exists.

The end diastole pressure is relatively constant at 80 mm. Hg. During heart systole this pressure will increase to between 120 and 180 mm. Hg. dependent on the stroke volume, heart systolic duration, and the systemic resistance level required to meet the body needs. The assist device must, therefore, be capable of adjusting its systolic duration to match that of the heart. If the stroke volume is pulsed from the assist device at too high (or too low) a rate, the peak systemic pressure level will correspondingly increase (or decrease) creating a non-physiological condition.

When the assist device can accommodate all the stroke volume and pulse it at a beat rate equal to the hearts, then the cardiac output will be matched.

VI ASSIST DEVICE

VI-I COUNTERPULSE AND IN-PHASE DEVICE

Initially the assist device was considered as one in

P_H	heart systolic pressure (fixed between 20 and 80 mm Hg).
P_{SYS}	peak systemic pressure (range between 120 to 180 mm Hg).
T_{SA}	assist device systolic duration = 20/Hr
W_S	average work per stroke = $.68 (P_{SYS})(S.V. MAX)$
$S.V. :$	stroke volume
$L.V. :$	left ventricle
$HR :$	heart rate
T_S	heart systolic duration
T_D	heart diastolic duration
T_O	$T_S + T_D \approx 3T_S =$ heart beat duration

Figure 1. Typical Pressure-Time and Volume-Time Relationships for the Assisted Left Ventricle and the Assist Device.

which a "sac" would be filled directly from the left ventricle and pulsed at the end of the fill. The device would counterpulse with the heart beat. During the heart systole the assist device would be filled from the left ventricle, while the fill rate (cardiac output) was being sensed. The resulting signal would be used to position a flow control device as a function of the cardiac output. At the end of the fill, the flow control device would remain positioned while a supply valve to the actuating device was opened. The actuator would then be pulsed and the blood forced into the system. The end of this pulse would be sensed to produce a signal for resetting the proper mechanisms, readying the system for the next heart pulse. The real difficulty in this approach is that some mechanism, e.g.; flow control device; must be positioned during fill, based on a maximum signal of flow or pressure and subsequently a valve would have to be cycled essentially on a zero error signal after the fill. Energy must be stored in the system for this purpose, resulting in a rather complex system.

An in-phase device would eliminate the above difficulty. since the time rate of change of pressure build-up of the fluid from the left ventricle could be used directly to simultaneously regulate fluid from a constant pressure reservoir to an actuator. This would be a high gain system allowing the actuator to follow along with the heart fill characteristics on a very low error signal.

VI-2 DISSIPATIVE DEVICE - THROTTLING

The dissipative assist device considered was one supplied from a constant pressure reservoir. A power supply and engine are used to drive a hydraulic pump. The pump pressurizes a reservoir, the pressure of which is regulated. Fluid is drawn from the reservoir as required to operate the assist device.

Figure 2 illustrates an arrangement of a throttling device shown ready to pulse. Valve B is closed and A is open. This causes a build up of pressure on the piston and an equivalent build up in the "sac". This pressure increase will maintain the inlet check valve closed. When the pressure builds to a value slightly above the systemic pressure (approximately 80 mm. Hg.) the discharge check valve will open and the piston will start to move. The velocity of the piston will be determined by (a) the variable orifice area of opening, set during the previous left ventricle discharge, (b) the piston area and (c) the drop in pressure across the orifice. The piston is allowed to travel to its stops. The flow from the "sac" will stop and as the "sac" pressure drops below the systemic pressure the discharge check valve will close. At this time valve A must be closed and B opened. The "sac" is now fully compressed and remains so until the start of the next heart systole. During the heart systole, the blood is pumped from the left ventricle into the "sac". This fluid acting on the piston plate forces the piston to return to its filled position as the fluid in the piston chamber is forced

out through a flow (cardiac output) detector. This detector repositions the orifice as a function of the cardiac output and also establishes the pressure peak, which the heart must work against. The piston is allowed to return to whatever position is required to handle stroke volume changes. At the end of the left ventricle discharge, valves A and B must be recycled permitting the device to again pulse.

Essentially this device requires a pressure drop to be established across a variable orifice opening between the reservoir and the piston of the assist device. The opening of the orifice is made proportional to the cardiac output. A flow match must exist such that the proper peak systemic pressure is produced. The need for this flow match can best be seen by consideration of a constant pressure reservoir, counterpulsing device, with no throttling. Assume, that the orifice does not exist, the valve between the reservoir and assist device piston can be opened instantaneously and the reservoir is large enough to be considered an infinite source, i.e; no reservoir pressure droop during the cycle. The piston would stroke almost instantaneously. This dictates that the reservoir pressure is on the piston at all time. The "sac" pressure is related to the piston pressure by the ratio of the respective effective areas for all steady state conditions. The force balance is simply - (reservoir pressure) (piston area) = (sac or systemic pressure) ("sac" effective area). From this relationship it can be seen that the systemic pressure would be constant. Its magnitude

Figure 2. Variable Orifice Throttling Device

would always be equal to the maximum peak systemic pressure, since the reservoir pressure and areas would have to be sized to meet the worst case condition to prevent stalling. This, of course, is not physiologically sound though it would be theoretically 100% efficient.

To be physiological, throttling is necessary. The stroke volume remains essentially constant and, therefore, the cardiac output varies mainly as a function of the heart rate. The peak systemic pressure varies, but is normally low at about 120 mm. Hg. To obtain a low pulse rate the variable orifice opening must be small. This in conjunction with a low systemic pressure will produce a large pressure drop to produce a low pressure on the piston.

Low systemic pressures will, therefore, produce the most inefficient operation, since throttling is wasted work. Because systemic pressures are low most of the time, the poorest efficiencies will exist most of the time.

The average efficiency for this device will be approximately 45% to 58% for 120 mm. Hg. and 180 mm. Hg. peak systemic pressures, respectively. (See Appendix B for calculations.)

VI-3 DISSIPATIVE DEVICE - FOLLOWER

A simpler more physiological device is a single acting piston, follower arrangement as shown in Figure 3. This in-phase device is schematically shown here with the symbols used for the efficiency calculations in the Appendix G-2.

This device has a double chambered "sac". During heart

systole, blood is discharged from the left ventricle, through the inlet check valve and into the first chamber of the "sac". The controls are activated by a signal directly related to the left ventricle flow, in a manner to permit fluid from the reservoir to move the piston at a matching rate. As the piston moves, the blood already in the second chamber is forced into the system through the discharge check valve and the first chamber fills. The check valve between chambers is held closed by the higher (systemic) pressure of the second chamber. The piston is permitted to travel the distance necessary to accommodate variations in stroke volume. At the end of this stroke; i.e.; start of heart diastole; the piston chamber is opened to the low pressure reservoir. The force produced by the return spring causes the piston to return. As the piston returns, the blood in the first chamber is forced through the intermediate check valve into the second chamber. This continues until the piston reaches its stop position. The device is now ready for the next pulse.

The addition of a third check valve for this device versus two in a counterpulse device is necessary. The inlet valve is required because of the flexibility of the aorta between the heart and the assist device. If this valve did not exist the blood would tend to flow back toward the heart during assist piston return.

As blood is pulsed into the assist device "sac", during heart systole, a bellows or similar device is activated.

The time rate of change of bellows length is directly related to the flow from the left ventricle; i.e., (ventricular flow) = (effective bellows area) (bellow velocity). The displacement of the bellows is related to the left ventricular pressure by the force balance; i.e., (effective bellows area) (ventricular pressure) = (bellows spring gradient) (bellows deflection).

The bellows movement is transmitted to a second bellows attached to a linkage. As the bellows starts to activate, the stationary piston connection acts as a pivot point allowing the spool valve to move. The piston remains stationary until the spool valve is activated and the pressure on the piston builds to a value to balance the resisting forces. The piston movement will tend to null the spool valve. The flow from the left ventricle will cause the bellows to continue deflecting at a rate directly related to it, causing the piston to follow at the same rate. When the left ventricle discharge is complete, the bellows will be fully activated. The forces resisting return of the piston will be essentially zero; i.e.; only that force necessary to move the mass of blood from one side of "sac" to the other and to overcome frictional forces. As the pressure on the bellows falls from the peak left ventricle value to the reference value of zero, the bellows will very quickly deactivate causing the three way spool valve to move to the drain position. A force imbalance, caused by the kinetic energy stored in the compressed return spring, now exists. The

piston will now retract, forcing the blood from one chamber to the other. The rate of return of the piston would be designed, based on the maximum heart beat rate of 200 per minute and the maximum stroke volume.

The arrangement as shown is a one - to - one follower of unity feedback. It is a high gain system working on very small error signals. Therefore, the following action of the piston to the bellows action will be essentially without lag.

At first look, the system does not appear to be any more efficient than that of the previously described throttling device. Due to the double "sac" arrangement of this in-phase device, its sizing will take into account an assist from the left ventricle. This is seen by writing a steady state force balance equation showing that, the pressure on the piston times the piston area is equal to the difference in pressures acting on the effective area of the "sac" times that area. The difference in pressures is the systemic minus the left ventricle pressure.

For the case of peak left ventricle pressure of 30 mm. Hg.; the average efficiency increase over the previously discussed throttling device is 1% to 4%. However, the follower device is not subject to stalling since the left ventricle could perform the "over" work necessary to prevent this from happening. This allows for designing based on a regulated pressure value with no margin against stall, or even basing the regulated pressure value on a peak systemic pressure

less than the maximum of 180 mm. Hg., i.e.; 150 mm. Hg. For peak systemic pressures above the 150 mm. Hg. the left ventricle would be allowed to work harder, e.g., for a peak systemic pressure of 180 mm. Hg., the left ventricle pressure would rise 30 mm. Hg. over the design value. The average efficiency will substantially improve to 70.9% at 120 mm. Hg. and 88% at 180 mm. Hg. (See Appendix G-2 for calculations).

Since the follower device normally will be assisted by the left ventricle, then the power supply drain's average efficiency will be higher than that of the system for heat dissipation evaluation. (This is shown in the calculations in Appendix G-2.) When considering the actual power drained from the implanted power supply, the assist from the left ventricle is not subtracted from the hydraulic power out, thus the efficiency is higher. This can be a decided advantage permitting the use of a smaller power source. The more "over" work the left ventricle is permitted, the larger the gain in this area.

This device will permit the left ventricle work level to be increased by simply reducing the level of the regulated pressure in the reservoir. This will have the effect of improving the average efficiency at all operating levels. This could be done in surgery, in the abdominal cavity.

In this follower system the impedance encountered by the left ventricle is a function of the spring gradient of the bellows or equivalent device. It, therefore, is scheduled and will be linear for a linear gradient spring as assumed

for the calculations of average efficiency shown in the appendix. As a result, the pressure-time relationship in the left ventricle will only approximate the physiological condition. This will still permit the designed peak left ventricular pressure to be obtained during each heart beat. By the use of non-linear devices, the pressure-time curve could be shaped to better approximate a physiological condition.

The systemic pressure-time and volume-time relationship will, however, be physiological regardless of the left ventricular schedule. The follower is a flow matching device in which the flow from it to the system is equal to the flow into it from the heart. Since the flow into the systemic is physiological, then the pressure build up must also be physiological.

VI-4 VARIABLE FLOW - VARIABLE PRESSURE DEVICE

The requirements of the left ventricle assist device could be met with a variable flow device. By varying the stroke of a variable displacement pump as a function of the cardiac output, the proper flow to the power piston could be obtained. With a positive displacement type pump the discharge pressure will rise to whatever value is necessary to overcome systemic pressures plus system friction losses and pressure drops. The fluid system, excluding the actual pump, can, therefore, be close to 100% efficient during the power stroke. By dumping the flow during piston return the efficiency is close to 100% at this time also. This is at the expense of increased complexity. Most of the components required in a

constant pressure counterpulse or in-phase device may still be needed, with the added complexity of a strokable pump and a power amplifier to stroke it. The advantage of maximum decoupling obtained by the constant pressure reservoir devices is lost. The unloading of the hydraulic pump during assist device diastole means that the engine is also unloaded. This will require that the generated energy during this time be stored as an increase of kinetic energy in a flywheel, thus requiring a larger and/or higher speed flywheel.

VI-5 DIRECT ENGINE TO BLOOD PUMP COUPLING

A direct coupling of the engine to blood pump was considered mechanically and hydraulically. The engine output would be controlled on a beat - by - beat basis.

In the mechanical approach, the hydraulic pump would not exist. The engine would be mechanically connected to the power piston and thus drive directly from it. To achieve the beat - by - beat synchronization would require a complex engine control. The close couple nature of this approach would mean that all components would have to be located in the thoracic cavity.

In the hydraulic approach, a flexible hydraulic coupling replaces the mechanical connection. The location of the power source and power piston no longer are restricted, however, the difficulty of synchronizing with the biologic heart still exists.

VI-6 VARIABLE PRESSURE RESERVOIR

A modified constant pressure reservoir system was considered. The primary action and requirements of the constant pressure system would still exist with the additional controls to vary the reservoir pressure as a function of the peak systemic pressure. At the lower peak systemic pressure conditions the reservoir pressure would be reduced. Because the cardiac output relationship must be met, throttling will still be necessary. However, due to the lower reservoir pressure, a smaller drop across an orifice is required, resulting in a substantial increase in efficiency. The controls for such a system become quite complex.

VII ADVANTAGES AND DISADVANTAGES OF FOLLOWER DEVICE

Due to the fact that the constant pressure reservoir follower system is recommended, the advantages and disadvantages are listed below:

I ADVANTAGES

- 1) Potentially very good life and reliability due to simplicity.
- 2) Maximum decoupling from the engine and engine controls.
- 3) Permits easy replacement of the power source.
- 4) Synchronized (in-phase) with the heart systole.
- 5) Quite physiological using a one - to - one follower based on the left ventricle pulse.
- 6) Allows a means of easily providing for dissipating heat to the blood.

- 7) Fail safe device. Failure of piston to operate in any position will allow the heart to function properly, but at systemic loads.
- 8) Device will not stall since the left ventricle can assist the device in the event of abnormally high peak systemic pressures.
- 9) The efficiency will improve as the left ventricle work load is allowed to increase.
- 10) The assist to the device from the left ventricle can permit the use of a smaller power supply.

II DISADVANTAGES

- 1) Relatively inefficient overall. Most inefficient at lower, more normal systemic pressures, i.e., largest heat dissipation necessary most of operating time.
- 2) Assist device "sac" would be slightly larger than that necessary in a counterpulse device.
- 3) Arrangement for through "sac" flow may prove difficult.
- 4) Three check valves are necessary versus two in a counterpulse device.

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APPENDIX G-1

I GENERAL REQUIREMENTS

The following requirements were generated to represent those of the assist device. They are summarized here and reflect the discussions held between Whiteley Hydraulics, Inc., and Thermo Electron Corp., and the research of the literature made available.

- 1) The assist device output must match the blood flow requirements, or cardiac output (C.O.) of the heart.

The cardiac output is equal to the heart rate (H.R.) times the volume of blood (stroke volume, S.V.) pumped from the heart during each heart beat, i.e.;

$$C.O. = (H.R.) (S.V.)$$

Where:

C.O = cardiac output, liters/min.

H.R. = heart rate, beats/min.

S.V. = stroke volume, liters/beat

- 2) The average work per stroke of the assist device must match that of the unassisted heart, i.e.; the shape of the time rate of change of left ventricular pressure and volume curves of the unassisted heart must be duplicated by the assist pump.

- 3) The control must synchronize the assist pump systole within each heart beat period, such that no pulse overlap occurs.

II SPECIFICATIONS

- 1) The assist device must be synchronized with the heart. It must pulse the entire stroke volume from it during each heart beat period. The assist device pulse duration must match that of the heart systole (T_s). The heart systolic duration is approximately one third that of each heart beat period.
- 2) Cardiac output and peak pressure demands must be matched by the assist device.
- 3) The assist device must be located downstream of the left ventricle and in series with it, and the aorta.
- 4) Peak systemic pressure can range from 120 to 180 mm. Hg. End diastole pressure will be approximately 80 mm. Hg.
- 5) Left ventricular systolic pressure must be capable of being varied from 20 to 80 mm. Hg. An adjustment must be provided to allow changing of this setting by the surgeon-cardiologist.
- 6) The assist device must be capable of pulsing at heart rates of 60 to 150 beats per minute, with

- provisions of going to 200 beats per minute under abnormal circumstances. Assist device pulses must not overlap normal heart pulses.
- 7) Provisions must be provided for stroke volume variations from 75 to 90 cc per beat.
 - 8) The normal left atrial pressure is from 4 to 6 mm. Hg.
 - 9) The control system must be hydromechanical.
 - 10) Pressurized water is available as the power source to the assist device.
 - 11) Water discharged from the left ventricle assist device must surround the device to permit heat transfer from the water to the blood for eventual dissipation through the body.

III DISCUSSION OF REQUIREMENTS

During the study several modifications were made to the original specifications. These resulted from an increasingly better understanding of what was needed. The changes produced more realistic and definitive requirements. While it was understood that all the requirements and limits may not apply to any one individual, they were necessary for complete evaluation of the control needs.

Initially it was considered that a device synchronized with the heart in a counterpulsing manner was necessary to meet the requirements. It soon became evident that a device

pulsing in-phase with the heart could apply as long as it pulsed once per heart beat and all the other requirements are met.

At the lower heart rates the left ventricle will pulse the stroke volume (c.c./beat) in approximately one third of a beat period. At the higher rates the time to pulse the stroke volume could increase to approximately one half of a beat. For evaluation the one third value was considered to apply at all heart rates.

The assist device must be capable of varying from 60 to 200 pulses per minute. It was further considered that a range of 60 to 150 beats per minute would be more realistic for most of the life of the device, but, that provisions for going to 200 beats per minute were necessary for abnormal conditions. The increase from 150 to 200, however, need not be as efficiently performed as at the lower rates.

Initially a fixed left ventricle pressure (back pressure) of between 20 to 30 mm. Hg. was considered. It was later decided that it would be desirable to set a back pressure, but, to include an adjustment to increase this pressure from 20 to 80 mm. Hg. as the damaged heart became stronger. This adjustment would be made by the surgeon-cardiologist.

APPENDIX G-2
DETERMINATION OF HYDRAULIC CONTROL EFFICIENCY

I Follower Device (refer to Figure 3 in Appendix G)

Assume that, as the blood is discharged into the aortic system, the pump discharged pressure increases linearly with the piston travel (X) from the minimum (mean systemic) value of pressure (P_{so}) required to open the discharge check valve to the maximum systemic pressure (P_{sm}).

$$P_s = P_{so} + K_{pl} X$$

$$K_{pl} = \frac{P_{sp} - P_{so}}{X_m}$$

The discharge flow of blood is given by

$$Q_s = A_p \frac{dX}{dt} \quad A_p V$$

The output power is given by

$$H_{out} = K_l P_s Q_s = K_l A_p V (P_{so} + K_{pl} X)$$

or

$$H_{out} = K_l A_p V \left[P_{so} + \left(\frac{P_{sp} - P_{so}}{X_m} \right) X \right]$$

Assume now that the left ventricle discharge pressure increases in a linear manner with increasing travel of the pumping piston from zero gage to a maximum of 30 mm of mercury. To make this assumption, it is necessary to assume that the force amplifier servo system is sufficiently fast and properly sized to meet these requirements. Then:

$$P_v = K_{p2} X$$

$$K_{p2} = \frac{P_{vp}}{X_m}$$

Next consider the force balance on the control piston

$$\begin{aligned}
 P_c A_c &= A_p (P_s - P_v) \\
 &= A_p (P_{so} + K_{p1} X - K_{p2} X) \\
 &= A_p (P_{so} + K_l X) \\
 K_p &= K_{p1} - K_{p2} = \frac{P_{sp} - P_{so} - P_{vp}}{X_m}
 \end{aligned}$$

$$\frac{A_p}{A_c} = \frac{P_c}{P_{so} + (P_{sp} - P_{so} - P_{vp}) \frac{X}{X_m}}$$

This area relationship is determined at max loading conditions ($X = X_m$) and ($P_c = P_{cm}$) thus

$$\frac{A_p}{A_c} = \frac{P_{cm}}{P_{sm} - P_{vm}}$$

The hydraulic power input is given by

$$H_h = K_l P_r A_c V$$

The power input as a result of the left ventricle assist is given by

$$H_a = K_l P_v A_p V = K_l A_p P_{vp} \left(\frac{X}{X_m} \right) V$$

Subtracting the assist power from the output power (assuming 100% efficient transfer) and dividing by hydraulic input power will give an equation proportional to overall efficiency.

$$\frac{H_{out} - H_a}{H_h} = \frac{A_p}{A_c P_r} \left[P_{so} + (P_{sp} - P_{so} - P_{vp}) \frac{X}{X_m} \right]$$

$$\frac{\epsilon}{100} = \left(\frac{P_{cm}}{P_r} \right) \left[\frac{P_{so} + (P_{sp} - P_{so} - P_{vp}) \frac{X}{X_m}}{(P_{sm} - P_{vm})} \right]$$

at $X = 0$

$$\frac{\epsilon}{100} = \left(\frac{P_{cm}}{P_r} \right) \left(\frac{P_{so}}{P_{sm} - P_{vm}} \right)$$

at $X = X_m$

$$\frac{\epsilon}{100} = \left(\frac{P_{cm}}{P_r} \right) \left(\frac{P_{sp} - P_{vp}}{P_{sm} - P_{vm}} \right)$$

$$\epsilon_{ave} = \frac{1}{2} \left[\left(\frac{\epsilon}{100} \right)_{X=0} + \left(\frac{\epsilon}{100} \right)_{X=X_m} \right]$$

or

$$\epsilon_{ave} = \frac{1}{2} \left(\frac{P_{cm}}{P_r} \right) \left(\frac{P_{so} + P_{sp} - P_{vp}}{P_{sm} - P_{vm}} \right) \quad (100)$$

This equation is used for determining heat rejection. However, it is not to be used for determining power drain.

Example Given: $P_{cm}/P_r = .8$, $P_{sm} = 180$, $P_{vm} = 30$, $P_{so} = 80$

Let: $P_{vp} = 30$, $P_{sp} = 120, 150$ and 180

at $P_{sp} = 180$

$$\epsilon_{ave} = \frac{1}{2} (80) \left(\frac{80 + 180 - 30}{180 - 30} \right) = 61.3\%$$

at $P_{sp} = 150$

$$\epsilon_{ave} = \frac{1}{2} (80) \left(\frac{80 + 150 - 30}{180 - 30} \right) = 53.3\%$$

at $P_{sp} = 120$

$$\epsilon_{ave} = \frac{1}{2} (80) \left(\frac{80 + 120 - 30}{180 - 30} \right) = 45.3\%$$

Assuming that the peak left ventricle discharge pressure remained constant then the above system efficiency variation could possibly be minimized by varying the control supply pressure (P_r) directly with the peak systemic pressure (P_{sp}).

Now is the assist power which was subtracted out for the above efficiency calculations is ignored, the effective power drain efficiency can be calculated.

$$P.e. = \frac{H_{out}}{H_h}$$

$$\frac{P.e.}{100} = \frac{A_p}{A_c P_r} \left[P_{so} + (P_{so} - P_{so}) \frac{X}{X_m} \right]$$

$$\frac{P.e.}{100} = \left(\frac{P_{cm}}{P_r} \right) \left[\frac{P_{so} + P_{sp} - P_{so} \frac{X}{X_m}}{P_{sm} - P_{vm}} \right]$$

at $X = 0$

$$\frac{P.e.}{100} = \left(\frac{P_{cm}}{P_r} \right) \left(\frac{P_{sp}}{P_{sm} - P_{vm}} \right)$$

at $X = X_m$

$$\frac{P.e.}{100} = \left(\frac{P_{cm}}{P_r} \right) \left(\frac{P_{sp}}{P_{sm} - P_{vm}} \right)$$

the average

$$\left(\frac{P.e.}{100} \right)_{ave} = \frac{1}{2} \left(\frac{P_{cm}}{P_r} \right) \left(\frac{P_{sp} + P_{so}}{P_{sm} - P_{vm}} \right)$$

For the conditions previously calculated.

at $P_{sp} = 180$

$$(P.e.)_{ave} = 40 \left(\frac{260}{150} \right) = 69.3\%$$

at $P_{sp} = 150$

$$(P.e.)_{ave} = 40 \left(\frac{230}{150} \right) = 61.3\%$$

at $P_{sp} = 120$

$$(P.e.)_{ave} = 40 \left(\frac{200}{150} \right) = 53.3\%$$

The average efficiencies can be increased by designing for a hydraulic stall condition less than the maximum peak systemic pressure of 180 mm Hg and allowing the left ventricle to provide the additional work required for the higher peaks.

It can be shown that the previously presented average efficiency equation will apply up to the peak systemic pressure used at hydraulic stall. Above this condition no additional hydraulic power is put in or taken out and, therefore, the efficiency remains constant at essentially 100%. The average efficiency above the design peak systemic pressure hydraulic stall condition is:

$$\left(\epsilon_{ave} \right)_{stall} \left(\frac{\epsilon_{ave}}{100} \right)_{P'_{sm}} \left(P'_{sm} - P_{so} \right) + \left(\frac{\epsilon_{ave}}{100} \right)_{P_{sp}} \left(P_{sp} - P'_{sm} \right)$$

Where P'_{sm} is the hydraulic stall design pressure.

Example Given: $P_{cm}/P_r = 1$, $P_{sm} = P'_{sm} = 150$, $P_{vm} = P'_{vm} = 30$, $P_{so} = 80$

Let: $P_{vp} = 30$, $P_{sp} = 120, 150$ and 180

at $P_{sp} = 120$

$$\epsilon_{ave} = \frac{1}{2} (100) \left(\frac{80 + 120 - 30}{150 - 30} \right) = 70.9\%$$

at $P_{sp} = 150$

$$\epsilon_{ave} = \frac{1}{2} (100) \left(\frac{80 + 150 - 30}{150 - 30} \right) = 83.3\%$$

at $P_{sp} = 180$

$$\epsilon_{ave} = 83.3 \left(\frac{150 - 80}{100} \right) + 100 \left(\frac{180 - 150}{100} \right) = 88.3\%$$

II Throttling Device

The efficiency for a throttling device without the left ventricle assist is simply proportional to the output power divided by the hydraulic power.

$$\frac{H_{out}}{H_h} = \frac{A_p}{A_c P_r} \left[P_{so} + P_{sp} - P_{so} \frac{X}{X_m} \right]$$

by proceeding as before:

$$e_{ave} = \frac{1}{2} \left(\frac{P_{cm}}{P_r} \right) \left(\frac{P_{so} + P_{sp}}{P_{sm}} \right) \quad (100)$$

for:

$$P_{cm}/P_r = .8, P_{so} = 80, P_{sm} = 180$$

then at $P_{sp} = 120$

$$e_{ave} = \frac{1}{2} (80) \left(\frac{200}{180} \right) = 44.4\%$$

at $P_{sp} = 150$

$$e_{ave} = \frac{1}{2} (80) \left(\frac{230}{180} \right) = 51.1\%$$

at $P_{sp} = 180$

$$e_{ave} = \frac{1}{2} (80) \left(\frac{260}{180} \right) = 57.8\%$$

NOMENCLATURE

A_c	Area of control piston
A_p	Area of pumping piston
ϵ	Instant control efficiency
ϵ_{ave}	Average control efficiency
H_a	Assist power from L. V.
H_h	Hydraulic power
H_{out}	Output power
K_l	Constant of proportionality
K_{pl}	Systemic pressure gradient
K_{p2}	Ventricular pressure gradient
P_c	Instant control pressure
P_{cm}	Maximum control pressure
$P.\epsilon.$	Instant power drain efficiency
$P.\epsilon_{ave}$	Average power drain efficiency
P_r	Regulated pressure
P_s	Instant systemic pressure
P_{sm}	Maximum systemic pressure
P_{so}	Minimum systemic pressure
P_{sp}	Peak systemic pressure
P_v	Instant ventricular pressure
P_{vp}	Peak ventricular pressure
P_{vm}	Maximum ventricular pressure

NOMENCLATURE (continued)

Q_s	System flow
V	Piston velocity
X	Instant piston travel
X_m	Maximum piston travel

APPENDIX H

INVERTER BLOOD PUMP CONTROL

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This appendix describes a compact device designed to operate from a hydraulic reservoir which:

- 1) Functions as a blood heat exchanger
- 2) Reduces the left ventricular pressure
- 3) Presents a near physiological pressure pulse to the systemic system in response to:
 - a) changes in heat rate
 - b) variations in stroke volume
 - c) shifts in cardiac output (implied in a and b)

System

A block diagram of the inverter assist device system is shown in Figure H-1.

Figure H-1. Block diagram of inverter assist device.

A sketch of the complete system is shown in Figure H-2.

Advantages

The salient feature of this design is that the servo mechanism follows the heart displacement which contains critical information on heart rate and stroke volume. Compared to the "follower" device discussed in Appendix G which acts 360 degrees out of phase with the left ventricle, this concept envisions an "inverter" counter-pulsing pump operating 180 degrees out of phase with the left ventricle.

Although both "inverter" and "follower" devices act on essentially the same information, the "inverter" approach discussed herein has these advantages relative to a "follower":

1. Only a single chamber is required (implies significant decrease in volume).
2. Two, rather than three check valves are used (reduces probability of clotting).
3. No bends in blood flow are required as in Appendix G (recall poor results with curved heat exchangers). (Curved heat exchangers had higher incidence of thrombosis than straight exchangers in joint NHI/AEC Additional Endogenous Heat Study.)
4. Pump bladder collapse is uniform as in present TECO-CHMC assist pump (unlike that suggested in Appendix G).

Disadvantages

However, the "inverter" device has the following disadvantages:

1. Dissipative (as are the "follower" devices). Note that spool valve logic could be made digital with improved efficiency.

Figure H-2. Implantable Inverter Control-Assist Pump-Heat Exchanger Device Powered by Hydraulic Reservoir.

2. The left ventricle pressure profile is modified by the assist pump. This is likely to be the case for any assist pump. There is no apparent reason why this should be objectionable.
3. The systemic systole and diastole times are switched. In this respect, the "inverter" system is less physiologic than the "follower" system. However, there is no apparent reason why the body should not accommodate itself to the reversal of the systole and diastole periods.

Hydraulic Reservoir

This assist pump is designed to be coupled to a hydraulic reservoir (or near constant pressure) because:

1. Hydraulic coupling is efficient.
2. Hydraulic reservoir decouples, to a large extent, the power source from the blood pump.
3. Pressure variations in hydraulic reservoir provide feedback to the engine to make it load responsive.
4. The mechanical potential energy stored in the hydraulic reservoir buffers the engine controls from the necessity of responding to fast transients.
5. Serves as part of the heat transfer loop to carry the reject heat from the power source to the blood pump.
6. Engine can be optimized for efficiency without regard to directly synchronizing and otherwise responding physiologically to the circulatory demands of variable heart rate, stroke volume and systemic pressure.

Subsystems

The inverter assist device consists of three primary components:

1. TECO-CHMC left ventricle assist pump.
2. Displacement sensor which senses change in heart volume.
This sensor would be placed as a cuff or garter around the outside of the heart.
3. Servo mechanism such as that discussed in Appendix G.

Blood Heat Exchanger

The inverter assist device is also used as a heat exchanger because:

1. Large surface area is available for exchanger interface with blood (which minimizes maximum rise in blood temperature).
2. Total prosthetic materials -- blood interface area is minimized since no other heat exchanger is required (reducing probability of clotting).
3. No additional cuts into a major artery are required for heat rejection (reducing trauma and complexity of surgery).
4. Dual use of assist device bladder for pumping and heat exchange reduces circulatory support system weight and volume.

Subsystem Description

Assist Pump -- The successful TECO-CHMC assist pump configuration has been chosen. This pump couples to the left ventricle through the apex of the heart. No significant modification of the pump, per se, is anticipated.

Heat Exchanger -- The pump bladder will be thinner in order to reduce the temperature drop across it. The bladder material will be switched from silicone rubber to stronger polyurethane.

Displacement Sensor -- Initially, pressure variations within the left ventricle and left atrium were considered as signals for the servo subsystem to follow. Unfortunately, during diastole both these signals are inadequate for servo control. During diastole significant changes in heart size occur even though pressure variations are small. Although the circumference variation is not simply related to cardiac output, it is not unreasonable to expect that it should be adequate for assist pump control-provided the elasticity, dimensions and placement of the sensor are properly tailored.

Use of a cuff or "garter" circumferential displacement sensor around the heart has several advantages:

1. The variation and magnitude of the displacement signal are suitable for control purposes during both systole diastole
2. No sensors that might interfere with cardiac function need to penetrate through the heart wall or be placed in any chamber of the heart.
3. Since the sensor, basically, detects displacement and not pressure (although some pressure variation may well be advantageous); the myocardial pressure is quite small. Therefore, irritation to the outside of the heart should be a solvable problem.
4. Clotting due to the sensor is non-existent since the sensor does not contact blood.

5. The sensor response can be tailored in a number of ways:
 - a. Dimensions (sensor wall thickness, diameter, circumference, topology)
 - b. Materials
 - c. Sensor Placement
 - d. Bellows dimensions and spring rate
6. Attachment flexibility. Sensor may be:
 - a. Sutured to myocardium
 - b. Harnessed to heart
 - c. Possibly, fluid coupled
7. Large area sensor gives minimum change in characteristics because of tissue growth.
8. Sensor placement requires no surgical incisions into any part of cardiovascular system.
9. Since thoracic cavity will be opened anyway, the heart will be readily accessible.
10. There is enough space around the heart that sensor size should not be critical.

It seems reasonable that a suitable sensor would be:

1. Constructed of prosthetic material
2. Light weight
3. Small size
4. Compliant
5. Coupled to bellows with very flexible tube.

Isovolumetric Contraction

The isovolumetric contraction presents a problem to the present control concept. During this period the volume does not change but there is a shortening of the heart between the base (atrioventricular septum) and the apex with a bulging or increase in the transverse diameter. Thus the heart becomes more spherical and the heart circumference increases (See Figure H-3) even though the volume does not change. For this short period of time, the sensor will misinterpret the circumference change as a volume change. Although this signal error may not be too significant, effort should be made to design and mount the sensor in such a way as to minimize this extraneous signal.

Servo Unit

The servo mechanism is essentially that discussed in Appendix G. This unit works in the following manner:

1. The displacement sensor's signal is transmitted to the bellows.
2. As the bellows actuates, the secondary piston connection acts as a pivot point which allows the spool valve to move.
3. The spool valve movement allows the pressure on the piston to build up to a value which gives a force balance. The piston movement is nulled by the spool valve.
4. The bellows and, hence, the drive piston "follows" the displacement changes due to blood flow into the heart.
5. The bellows and, hence, the drive piston will stop whenever the heart stops filling or when the drive piston reaches bottom dead center--whichever comes first.

Figure H-3. Time Variation of Assist Device Parameters.

6. The discharge from the left ventricle, aided by the spring under the drive piston, then returns the drive piston to a position determined by the stroke volume that the left ventricle ejects into the bladder of the assist pump.

APPENDIX I

DIFFERENTIAL EQUATION FOR ENGINE RESPONSE

The following analysis is useful for characterizing engine response. However this analysis gives no information about what happens within an engine cycle other than the change in mean speed.

The change in the flywheel energy, per engine cycle, is given by

$$\Delta W_{\text{Fly}} = \Delta W_{\text{Steam}} - \Delta W_{\text{Hydraulic}} - \Delta W_{\text{Dissipative}}$$

Now,

$$\Delta W_{\text{Fly}} = \Delta \left[\frac{1}{2} I \omega^2 (t) \right]$$

where

I = Inertia of flywheel

ω = Angular velocity (averaged over an engine cycle)

The steam work per stroke is

$$\Delta W_{\text{Steam}} = P_{\text{MEP}} A_s S_s$$

where

P_{MEP} = Mean effective pressure of steam during expansion

A_s = Area of steam piston

S_s = Stroke of steam piston

The work done by the hydraulic pump piston, per stroke, is given by

$$\Delta W_{\text{Hydraulic}} = \bar{P}_H(t) A_H S_H$$

where

$\bar{P}_H(t)$ = Mean effective hydraulic pressure over an engine cycle

A_H = Area of hydraulic piston

S_H = Stroke of hydraulic piston

The energy dissipated per stroke is represented by

$$\Delta W_{\text{Dissipative}} = B' \omega(t)$$

where

B' = Proportionality factor for lumped damping effects

Substitution into the first equation in this appendix yields

$$\therefore \Delta \left[\frac{1}{2} I \omega^2(t) \right] = P_{\text{MEP}} A_s S_s - \bar{P}_H(t) A_H S_H - B' \omega(t)$$

The time for cycle is $\Delta t = \frac{2\pi}{\omega}$.

Dividing the entire equation by ΔT yields:

$$\begin{aligned} \frac{\Delta \left[\frac{1}{2} I \omega^2(t) \right]}{\Delta t} &\cong \frac{d \left[\frac{1}{2} I \omega^2(t) \right]}{dt} \cong \frac{P_{\text{MEP}} A_s S_s \omega(t)}{2\pi} - \\ &- \frac{\bar{P}_H(t) A_H S_H \omega(t)}{2\pi} - \frac{B' \omega^2(t)}{2\pi} \end{aligned}$$

or, in the limit as Δt goes to zero,

$$\omega \frac{d\omega}{dt} \cong \frac{P_{\text{MEP}} A_s S_s \omega(t)}{2\pi I} - \frac{\bar{P}_H(t) A_H S_H \omega(t)}{2\pi I} - \frac{B' \omega^2(t)}{2\pi I}$$

It will be useful to express the mean hydraulic pressure, $\bar{P}_H$, in terms of other system parameters.

Figure I-1 High-Pressure Reservoir

K = spring constant (lbs/in.)

C_o = precompression of spring

From reference to Figure A.7.1, it is clear that

$$A_H = \frac{K}{A_p} (C_o + X), \quad \frac{dV}{dt} = Q_{in} - Q_{out} = A_p \frac{dx}{dt}$$

$$Q_{in} = \frac{A_H S_H}{\Delta t} = \frac{A_H S_H \omega}{2\pi}$$

$$X = \frac{1}{A_p} \int \left[\frac{A_H S_H \omega}{2\pi} - Q_{out}(t) \right] dt + C_1$$

C_1 = constant of integration

$$\bar{P}_H(t) = \frac{K}{A_p} \left(C_o + \frac{1}{A_p} \int \left[\frac{A_H S_H \omega}{2\pi} - Q_{out}(t) \right] dt + C_1 \right)$$

Substituting $\bar{P}_H(t)$ into the differential equation, differentiating each term with respect to time, and simplifying gives

$$\frac{d^2 \omega}{dt^2} + \left(\frac{B'}{2\pi I} \right) \frac{d\omega}{dt} + \left(\frac{K}{A_p^2} \right) \left(\frac{A_H^2 S_H^2}{4\pi^2 I} \right) \omega = \left(\frac{A_H S_H K}{2\pi I A_p^2} \right) Q_{out}(t)$$

which is a linear, second-order differential equation with constant coefficients whose solutions are well known.

Although the system response as a function of the engine parameters is of primary interest, the number and range of the parameters involved in the coefficients would give unwieldy results. However, all the parameters are not independent. Therefore the system constraints must be invoked to bracket the coefficient variations

$$A \frac{d^2 \omega}{dt^2} + B \frac{d\omega}{dt} + C\omega = D \cdot Q(t)$$

$$A = 1 \quad B = B'/2\pi I \quad C = \left(\frac{K}{A_p^2} \right) \left(\frac{A_H^2 S_H^2}{4\pi^2 I} \right)$$

$$D = \frac{A_H S_H K}{2\pi I A_p^2} \quad \eta_s = \text{Mechanical Efficiency of steam engine} = 1 - \frac{B' \omega}{P_{MEP} A_s S_s}$$

Constraint Equations

$$\overline{\text{RPM}} \cdot \overline{\eta}_s \cdot U \cdot P_{MEP} \cdot A_s \cdot S_s = S_p = \overline{\text{RPM}} \cdot \frac{1}{\overline{\eta}_H} \cdot U \cdot P_D \cdot A_H \cdot S_H$$

$\overline{\text{RPM}} \equiv$ mean cycles/min, $S_p \equiv$ steam engine shaft power at mean rpm

$U \equiv$ conversion factor, $P_{MEP} \equiv$ mean effective steam pressure

$A_s \equiv$ area of steam piston, $S_s \equiv$ stroke of steam piston

$\bar{\eta}_H \equiv$ efficiency of hydraulic pump at mean rpm

$P_D \equiv$ mean differential hydraulic pressure

$A_H \equiv$ hydraulic piston area, $S_H = S_s \equiv$ stroke of hydraulic pump piston

$\bar{\eta}_s \equiv$ mechanical efficiency of steam engine at mean rpm

Coefficients of Differential Equations

$$\boxed{B = B' / 2 \pi I} \quad C = \left(\frac{K}{A_p^2} \right) \left(\frac{A_H^2 S_H^2}{4 \pi^2 I} \right) \quad D = \frac{A_H S_H K}{2 \pi I A_p^2}$$

$$\boxed{C = D \cdot \frac{A_H S_H}{2 \pi}} \quad \boxed{D = \frac{B}{B'} \left(\frac{A_H S_H K}{A_p^2} \right)}$$

$$\bar{\eta}_s = 1 - \frac{B' \bar{\omega}}{P_{MEP} A_s S_s}, \quad \bar{\eta}_s P_{MEP} A_s S_s = P_{MEP} A_s S_s - B' \bar{\omega}$$

$$\bar{\eta}_s P_{MEP} A_s S_s = \frac{S_p}{RPM \cdot U}, \quad \omega = 2 \pi f \quad f = \frac{RPM}{60} \quad \omega = \frac{\pi RPM}{30}$$

$$\boxed{B' = \left(\frac{30}{\pi \cdot RPM} \right) \left[P_{MEP} A_s S_s - \frac{S_p}{RPM \cdot U} \right]}$$

High-Pressure Reservoir

$P_{LOW} \equiv$ lowest allowable hydraulic reservoir pressure (absolute)

$P_{HI} \equiv$ highest allowable hydraulic reservoir pressure (absolute)

$P_D \equiv$ mean differential hydraulic pressure

$P_M \equiv$ mean absolute hydraulic reservoir pressure

$$P_M = \frac{P_{LOW} + P_{HI}}{2} \quad \frac{P_{LOW}}{P_{HI}} = P_{RATIO}$$

$V_{pis} \equiv$ volume of hydraulic piston (for average S. V.)

$N \equiv$ number of piston volumes stored

$X_r \equiv$ full stroke of hydraulic reservoir piston

$$A_p X_r = N V_{pis} \quad P_{LOW} = \frac{K}{A_p} C_o \quad P_{HI} = \frac{K}{A_p} (C_o + X_r)$$

$$P_{RATIO} = \frac{C_o}{C_o + X_r} \quad P_{RATIO} C_o + P_{RATIO} X_r = C_o$$

$$C_o (1 - P_{RATIO}) = P_{RATIO} X_r \quad C_o = \frac{P_{RATIO} X_r}{1 - P_{RATIO}}$$

$$\boxed{X_r = \frac{N V_{pis}}{A_p}} \quad \boxed{C_o = \left(\frac{P_{RATIO}}{1 - P_{RATIO}} \right) \frac{N V_{pis}}{A_p}} \quad P_{LOW} = \frac{K}{A_p} C_o$$

$$K = \frac{P_{LOW} A_p}{C_o} = \frac{P_{LOW} A_p^2 (1 - P_{RATIO})}{P_{RATIO} N V_{pis}} = \boxed{\frac{A_p^2 \Delta P}{N V_{pis}}} = K$$

$$\frac{P_{LOW} (1 - P_{RATIO})}{P_{RATIO}} = \frac{(1 - P_{LOW}/P_{HI})}{1/P_{HI}} = P_{HI} - P_{LOW} \equiv \Delta P$$

Constraint and Coefficients

$$\overline{\text{RPM}} \cdot \frac{1}{\bar{\eta}_H} \cdot U P_D A_H S_H = S_p \quad A_H S_H = \frac{S_p \bar{\eta}_H}{\overline{\text{RPM}} \cdot U \cdot P_D}$$

$$D = \frac{B}{B'} \left(\frac{A_H S_H K}{A_p^2} \right) \quad \frac{K}{A_p^2} = \frac{\Delta P}{N V_{pis}}$$

$$D = \frac{B}{B'} \left(\frac{\Delta P}{N V_{pis}} \right) \left(\frac{S_p \bar{\eta}_H}{\overline{\text{RPM}} \cdot U \cdot P_D} \right) \quad C = D \cdot \frac{A_H S_H}{2\pi} = D \cdot \frac{S_p \bar{\eta}_H}{2\pi \overline{\text{RPM}} \cdot U \cdot P_D} = C$$

Power Out of Reservoir

$$\text{Power} = \overline{Q_{out}} \times \bar{P} \quad \overline{Q_{out}} = \frac{V_{pis}}{T_c} \quad T_c = 60/\text{HR} \quad \bar{P} = P_M$$

$\bar{\eta}_R \equiv$ efficiency of high-pressure reservoir at mean rpm

$$\bar{\eta}_H \bar{\eta}_R S_p = \overline{Q_{out}} \times \bar{P} = \frac{V_{pis}}{T_c} \times P_M$$

$T_c \equiv$ time for one heart cycle, seconds, for typical HR

$$\bar{\eta}_H \bar{\eta}_R S_p = \frac{V_{pis}}{T_c} \times P_M \times U \quad T_c = 60/\text{HR}$$

HR = heart rate, beats per minute, mean for typical man

$$V_{pis} = \frac{\bar{\eta}_H \bar{\eta}_R S_p T_c}{P_M \cdot U} = \frac{60 \bar{\eta}_H \bar{\eta}_R S_p}{P_M \cdot U \cdot \text{HR}}$$

$$D = \frac{B}{B'} \left(\frac{\Delta P}{N V_{pis}} \right) \left(\frac{S_p \bar{\eta}_H}{\overline{\text{RPM}} \cdot U \cdot P_D} \right) = \frac{B}{B'} \left(\frac{\Delta P \cdot P_M \cdot \text{HR}}{N 60} \right) \left(\frac{1}{\overline{\text{RPM}} \cdot P_D} \right)$$

$$D = \frac{B}{B} \left(\frac{\Delta P \cdot P_M \cdot HR}{N60 \bar{\eta}_R} \right) \left(\frac{1}{RPM P_D} \right)$$

$$\Delta P = P_{HI} - P_{LOW} \quad P_M = \frac{P_{HI} + P_{LOW}}{2} \quad \Delta P \cdot P_M = \frac{P_{HI}^2 - P_{LOW}^2}{2}$$

$P_{LPR} \equiv$ mean low-pressure reservoir pressure

$$P_D = P_M - P_{LPR} = \frac{P_{HI} + P_{LOW}}{2} - P_{LPR} \quad \frac{P_{HI} + P_{LOW}}{2} = P_D + P_{LPR} = P_M$$

$$D = \frac{B}{B} \left(\frac{[P_{HI}^2 - P_{LOW}^2] HR}{120 N \bar{\eta}_R} \right) \left(\frac{1}{RPM P_D} \right)$$

$$C = D \cdot \frac{S_p \bar{\eta}_H}{2\pi RPM \cdot U \cdot P_D}$$

Excitation Function $Q_{out}(t)$

Figure I-2 Excitation Function

$$2 T_s = \frac{T}{2} + 2 \Delta t \quad \frac{T}{2} = 2 T_s - 2 \Delta t \quad T = 4 (T_s - \Delta t)$$

$T_s \equiv$ systolic time of particular heart cycle (not to be confused with
 T_c cycle time of mean heart rate used in constraint equation)

$$2 \Delta t = 2 T_s - T/2 \quad \Delta t = T_s - T/4$$

$$\omega = \frac{2\pi}{T} = \frac{2\pi}{3T_s} \quad q(t) = \left| \sin \frac{2.09}{T_s} \cdot t - \sin \frac{2.09}{T_s} \Delta t \right|$$

$$\omega = 2.09/T_s$$

It is assumed that heart systole is $\frac{1}{3}$ of total heart cycle time

$$\Delta t = T_s - T/4 = T_s/4$$

$$q(t) = \left| \sin \left(\frac{2.09}{T_s} \cdot t \right) - \sin \frac{2.09}{T_s} \cdot \frac{T_s}{4} \right| = \left| \sin \left(\frac{2.09}{T_s} \right) - \sin (0.523) \right|$$

$$V_{pis} = 2 Q_{MAX} \int_0^T \left| \sin \left(\frac{2.09}{T_s} \cdot t \right) - 0.5 \right| dt$$

$$V_{pis} = 2 Q_{MAX} \int_0^{3T_s} \left| \sin \left(\frac{2.09}{T_s} \cdot t \right) - 0.5 \right| dt$$

$$\int_0^{3T_s} \left| \sin \frac{2.09}{T_s} \cdot t - 0.5 \right| dt = \int_0^{3T_s/2} \left| \sin \frac{2.09}{T_s} \cdot t - 0.5 \right| dt$$

$$V_{pis} = A_1 - A_2 \quad A_1 = 2 Q_{MAX} \int_0^{3T_s/2} \sin \frac{2.09}{T_s} \cdot t dt$$

$$A_1 = 2 Q_{MAX} \cdot \frac{T_s}{2.09} \left[-\cos \frac{2.09}{T_s} \cdot t \right]_0^{3T_s/2} = 1.91 Q_{MAX} T_s$$

$$\begin{aligned} A_1 - A_2 &= 2 Q_{MAX} \int_{\Delta t}^{3T_s/2 - \Delta t} \left[\sin \frac{2.09}{T_s} t - 0.5 \right] dt \\ &= 2 Q_{MAX} \int_{T_s/4}^{5T_s/4} \left[\sin \frac{2.09}{T_s} \cdot t - 0.5 \right] dt \end{aligned}$$

$$\frac{3T_s}{2} - \frac{T_s}{4} = \frac{5T_s}{4}$$

$$A_1 - A_2 = 2 Q_{MAX} \frac{T_s}{2.09} \left[\left(-\cos \frac{2.09}{T_s} \cdot t \right) \Big|_{T_s/4}^{5T_s/4} - (0.5t) \Big|_{T_s/4}^{5T_s/4} \right]$$

$$-\cos \frac{2.09}{T_s} \cdot \frac{5T_s}{4} = -\cos 2.615 = 0.866, \quad \cos \frac{2.09}{T_s} \cdot \frac{T_s}{4} = \cos 0.523 = 0.866$$

$$0.5 \left(\frac{5T_s}{4} \right) - 0.5 \left(\frac{T_s}{4} \right) = 0.5 T_s$$

$$A_1 - A_2 = 2 Q_{MAX} \left[1.732 \frac{T_s}{2.09} - 0.5 T_s \right]$$

$$A_1 - A_2 = 0.656 Q_{MAX} \cdot T_s = V_{pis} \quad Q_{MAX} = 1.52 V_{pis} / T_s$$

$$V_{pis} = \frac{60 \bar{\eta}_H \bar{\eta}_R S_p}{P_M U \cdot HR}$$

$$Q_{out}(t) = Q_{MAX} \quad q'(t) \quad q'(t) = 2 \left| \sin \frac{2.09}{T_s} \cdot t - 0.5 \right|$$

$$Q_{out}(t) = 2 Q_{MAX} \left| \sin \left(\frac{2.09}{T_s} \cdot t \right) - 0.5 \right|$$

$$Q_{out}(t) = \frac{180.24 \bar{\eta}_H \bar{\eta}_R S_p}{T_s \cdot P_M \cdot U \cdot HR} \left| \sin \frac{2.09}{T_s} \cdot t - 0.5 \right|$$

Finally, the system constraints result in the coefficients:

$$D \cdot Q_{out}(t) = \frac{B}{B'} \left(\frac{\Delta P}{60N} \right) \left(\frac{1}{RPM P_D} \right) \left(\frac{180.24 \bar{\eta}_H S_p}{T_s \cdot U} \right) \left| \sin \left(\frac{2.09}{T_s} \cdot t \right) - 0.5 \right|$$

$$\boxed{B = B' / 2\pi I} \quad \boxed{C = D \cdot \frac{S_p \bar{\eta}_H}{2\pi RPM \cdot U \cdot P_D}} \quad D = \frac{B}{B'} \left(\frac{[P_{HI}^2 - P_{LOW}^2]_{HR}}{120N \eta_R} \right) \left(\frac{1}{RPM P_D} \right)$$

for the equation in the form

$$\frac{d^2 \omega}{dt^2} + B \frac{d\omega}{dt} + C\omega = D \cdot Q_{out}(t)$$

SPRING RETURN ENGINE

The dynamics of the linear spring return engine were analyzed using a simple model. The engine characteristics were investigated as a function of spring constant, piston mass and cycle frequency. A sketch of the simplified linear engine is shown in Figure J-1.

Figure J-1. Simplified Linear Engine

Define:

P_s = Steam pressure acting on the steam piston

$$A_s = \text{Area of the steam piston over which the pressure } P_s \text{ acts}$$

- P_h = Average hydraulic fluid pressure acting on the piston of the hydraulic pump
 A_h = Area of the hydraulic piston over which the pressure P_h acts
 P_f = Average feed water pressure acting on the piston of the feed water pump
 A_f = Area of the feed water piston over which the pressure P_f acts
 K = Spring constant
 L = Piston stroke

The motion of the engine is divided into the power stroke and the return stroke. During the power stroke the steam expands and the piston travels from top dead center to bottom dead center. The piston goes from bottom dead center to top dead center on the return stroke.

During the power stroke both the hydraulic force and the feed water force resist the steam force; whereas during the return motion only energy stored in the spring moves the piston. The potential energy stored in the spring is converted into the Kinetic energy of the piston which is consumed in opening the inlet valve.

Power Stroke

Let y describe the motion of the piston from top dead center to bottom dead center (see Figure J-2).

Figure J-2. Spring-mass system during power stroke.

The equation of motion is

$$P_s A_s - P_h A_h - P_f A_f - K y = M \frac{d^2 y}{dt^2}$$

The driving force for the system is the steam force which varies with the displacement shown in Figure J-3.

Figure J-3. Indicator Diagram for Steam Engine

An isentropic expansion takes place after the inlet valve is closed.

For the isentropic expansion

$$pv^\gamma = \text{Constant}$$

$$\gamma = C_p / C_v$$

When both the inlet valve and the exhaust valve are closed the system acts as a closed system. Then

$$\text{Specific volume } v = \frac{1}{\frac{m}{\frac{\pi d^2}{4} (y + C_\ell)}}$$

Where:

C_ℓ = clearance distance

d = diameter of the piston

m = mass of the steam intake after the inlet valve is closed

Therefore

$$v = \frac{\pi d^2}{4} \frac{(y + C_\ell)}{m}$$

and

$$p_s \left[\frac{\pi d^2 (y + C_\ell)}{4 m} \right]^\gamma = \text{Constant}$$

Now suppose the cut-off conditions are known. If p_c is the release pressure, then $P_c v_c^\gamma = \text{constant}$

Then

$$p_s = \frac{\zeta}{(y + C_\ell)^\gamma} \quad \text{where } \zeta = p_c \left(\frac{4 m v_c}{\pi d^2} \right)^\gamma$$

The equation of motion becomes:

$$M \frac{d^2 y}{dt^2} + K y = \frac{\zeta}{(C_\ell + y)^\gamma} A_s - (P_f A_f + P_h A_h).$$

$$y_1 < y < L$$

The value of γ for steam is 1.28.

$$\text{and } M \frac{d^2 y}{dt^2} + Ky = P_B A_s - (P_f A_f + P_h A_h)$$

$$0 < y < y_1$$

P_B = boiler pressure or the high pressure at which the steam comes into the engine.

However, the above equation does not fall in any standard form.

In order to simplify the analysis the foregoing function was approximated as shown in Figure J-4.

Figure J-4. Approximated Indicator Diagram

The values of the P_{in} and P_{out} were chosen such that the area within the indicator diagrams are equal.

Calculations were made for the inlet steam conditions of 1200 psia and 1250°F, assuming the condenser pressure to be 15 psia.

The work done per pound of steam is 482 Btu/lb.

If v_e is specific volume at the end of the stroke, then the mass of the intake steam per stroke is $A_s L / v_e$.

$$\text{Therefore, the work per stroke} = 482 \times \frac{A_s L}{v_e} \frac{\text{Btu}}{\text{Stroke}}$$

$$= \text{Mep} \times A_s L$$

$$\text{Mep} = \frac{482 \text{ B/lb}}{8.3 \text{ cft/lb}} = \frac{482 \times 778 \times 12}{8.3 \times 1728} = 314 \text{ psi}$$

$$P_{out} = 15 \text{ psia,}$$

$$P_{in} = 15 + 2 (314) = 628 + 15 = \text{psia}$$

The forcing function is

$$P_s = 643 - 628 (Y/L)$$

and the equation of motion becomes

$$\begin{aligned} M \frac{d^2 y}{dt^2} + Ky &= P_s A_s - (P_f A_f + P_h A_h) \\ &= \{643 - 628 y/L\} A_s - (P_f A_f + P_h A_h) \end{aligned}$$

In general

$$M \frac{d^2 y}{dt^2} = [P_{in} - (P_{in} - P_{out}) y/L] A_s - P_f A_f - P_h A_h$$

Finally,

$$M \frac{d^2 y}{dt^2} + \left[K + \left(\frac{P_{in} - P_{out}}{L} \right) A_s \right] Y = P_{in} A_s - P_f A_f - P_h A_h.$$

The solution of the equation is

$$y = A_1 \sin \omega_p t + B_1 \cos \omega_p t + \left\{ \frac{P_{in} A_s - P_f A_f - P_h A_h}{K + \frac{P_{out}}{L} A_s} \right\}$$

The initial conditions to determine the unknown constants are

$$t = 0 \qquad y = 0 \qquad \dot{y} = 0$$

Therefore

$$y = \left[\frac{P_{in} A_s - P_f A_f - P_h A_h}{K + \frac{P_{out}}{L} A_s} \right] \left[1 - \cos \omega_p t \right]$$

$$\omega_p^2 = \frac{K + \frac{P_{out}}{L} A_s}{M}$$

At $t = \tau_p$ (i.e., at bottom dead center) the piston velocity should be zero. This implies

$$\therefore \boxed{\omega_p \tau_p = \pi}$$

Also at $t = \tau_p$, $y = L$ and

$$\frac{P_{in} A_s - P_f A_f - P_h A_h}{K + \frac{P_{in} - P_{out}}{L} A_s} = L/2$$

Return Stroke

Figure J-5. Spring-Mass System During Return Stroke

The spring-mass system during the return stroke is shown in Figure J-5. At the beginning of the return stroke, the spring force is KL .

The equation of motion during the return stroke is

$$M \frac{d^2 x}{dt^2} + Kx = KL$$

so that

$$x = L + A_2 \sin \omega_R t + B_2 \cos \omega_R t$$

$$\omega_R^2 = K/M$$

At $t = 0$ $x = 0$, $x = 0$

$$\therefore X = L (1 - \cos \omega_R t)$$

Suppose τ_R is the return timing

then,

$$t = \tau_R \quad X = L$$

$$L = L (1 - \cos \omega_R \tau_R)$$

Therefore,

$$\omega_R \tau_R = \frac{\pi}{2}$$

Engine Power

Assume the average power pumped into the blood is 1.75 watts and the blood pump efficiency is 75%. The net power delivered by the hydraulic pump is then 2.33 watts. Assuming a hydraulic pump efficiency of 85%, the power into the hydraulic pump is 2.74 watts.

Taking the controls efficiency to be 90% implies an engine power output of 3.05 watts.

The work done by the feed water pump was estimated by $v \Delta p$ for this case to be = 0.15 watts.

Thus, the steam piston output must be 3.20 watts.

If τ_p = Time taken to complete the power stroke.

τ_R = Time taken to complete the return stroke.

$$\therefore \text{Engine output/cycle} = 3.20 \times (\tau_p + \tau_R)$$

Also, the engine output/cycle

$$= \eta_m [\text{Mep}] \times \underline{\text{A. L.}}$$

where

$$\begin{aligned}\eta_m &= \text{Mechanical Efficiency} \\ &= 0.80\end{aligned}$$

The power considerations satisfy the following relation

$$\eta_m \cdot \text{Mep. A. L.} = 3.20 \times (\tau_p + \tau_R) \quad (1)$$

The power developed by the engine goes into the hydraulic pump, feedwater pump and the compression of the spring.

Out of 3.20 watts, 0.15 watts were estimated for the feedwater pump. From the experience of Thermo Electron with steam engines, it is expected that about 10% of the total cycle energy goes into the valving. The compression energy which goes into the spring should be enough to open the inlet valve. Therefore,

$$\begin{aligned}\text{Energy into Hydraulic pump/cycle} &= (3.20 - 0.15) \times 0.9 (\tau_p + \tau_R) \\ &= P_h A_h \cdot L\end{aligned}$$

∴ To design the Linear Engine consider the following equations

$$\begin{aligned}\eta_m \text{ Mep. A. L.} &= 3.20 (\tau_p + \tau_R). \quad (1) \\ &(\text{Engine power balance})\end{aligned}$$

$$3.05 \times 0.9 \times (\tau_p + \tau_R) = P_h A_h \cdot L \quad (2)$$

(Hydraulic pump power balance)

$$\frac{P_{in} A_s - P_f A_f - P_h A_h}{K + \frac{(P_{in} - P_{out}) A_s}{L}} = L/2 \quad (3)$$

(Power stroke equation of motion)

$$\omega_p \tau_p = \pi \quad \begin{array}{l} \text{(Velocity at bottom dead center} \\ \text{is zero during power stroke)} \end{array} \quad (4)$$

$$\omega_p = \sqrt{\frac{K + \frac{(P_{in} - P_{out}) A_s}{L}}{M}} \quad (5)$$

$$\omega_R \tau_R = \frac{\pi}{2} \quad \text{(Return stroke equation of motion)} \quad (6)$$

$$\omega_R = \sqrt{\frac{k}{M}} \quad (7)$$

$$\therefore \frac{\tau_R}{\tau_p} = \frac{1}{2} \sqrt{1 + \frac{(P_{in} - P_{out}) A_s}{KL}} \quad (8)$$

$$\begin{aligned} 1/2 KL^2 &= E_v = \text{Energy to open inlet valve} \\ &= 0.1 \times 3.20 \times (\tau_p + \tau_R) \cdot \end{aligned} \quad (9)$$

Assume the stroke and diameter of steam piston to be 0.25 inches

Using equations (1) and (2),

$$P_h A_h = 9.65 \text{ lb}_f$$

Assuming that 10% of the energy goes to open the inlet valve,

$$1/2 KL^2 = \frac{9.65 \times 1/4}{10} = 0.241 \text{ in lbs.}$$

Therefore

$$K = 7.712 \text{ lb/in}$$

From equation (8),

$$\frac{\tau_R}{\tau_P} = 2.07$$

From equation (2),

$$\tau_R + \tau_P = 0.099 \text{ which corresponds to a speed of 606 RPM.}$$

$$\tau_R = \frac{2.07}{3.07} \times 0.099 = 0.067$$

$$\therefore M = K \frac{\tau_R^2}{\pi} \times g_o \times 4$$

$$= \frac{(0.067)^2}{\pi} \times 7.712 \times 386 \times 4 = 5.35 \text{ lbs}$$

When the hydraulic pump work is done on the power stroke, the characteristic features of the system are given in Table 1.1.

By changing the length of the stroke and diameter of piston, lower values of $(\tau_R + \tau_P)$ and hence lower value of mass of system could be obtained. However to go much below the piston length and diameter of the sample case would probably not be practical.

TABLE 1.1

CHARACTERISTICS OF LINEAR, SPRING RETURN STEAM ENGINE
(HYDRAULIC PUMP WORK ON POWER STROKE)

Mass of the system	5.35 lbs
RPM	606
Spring Constant	7.712
τ_R / τ_p	2.07
Length of Stroke	0.25 inch
Diameter of the piston	0.25 inch
Pumping power into blood	1.75 watts

Another possible case is for the hydraulic pump working on the return stroke. In this arrangement, another variable which could be adjusted to have the desirable characteristics and thus a less stiff spring could be used to absorb the same amount than a spring with no precompression.

By the same procedure as used in the previous case, the equations of motion of piston on the power and return stroke are as follows:

Power Stroke

$$Y = Y_0 (1 - \cos \omega_p t) \quad \text{Initial conditions}$$

$$t = 0, \quad Y = 0, \quad \dot{Y} = 0$$

$$Y_0 = \frac{P_{in} A_s - P_f A_f - KC}{\left[K + \frac{P_{in} - P_{out}}{L} A_s \right]}$$

$$C = \text{Pre-compression in the spring}$$

At bottom dead center, the velocity of the piston is desired to be zero so that

$$\omega_p \tau_p = \pi$$

Return Stroke

$$x = x_0 (1 - \cos W_R t)$$

where

$$x_0 = \frac{K(C + L) - P_h A_h}{K} \quad \begin{array}{l} \text{Initial conditions} \\ t = 0, x = 0, \dot{x} = 0 \end{array}$$

(This equation is approximate because just before return stroke ends near top dead center some energy goes to open the inlet valve. The energy involved in opening the inlet valve is assumed negligible compared to that which goes into the hydraulic pump, and does not affect the equation of motion.)

Because the feedwater pump works on the power stroke and out of 3.20 watts produced 0.15 watts go to feedwater pump and rest stored in compression of the spring. On the basis of energy/cycle,

$$3.05 (\tau_p + \tau_R) = 1/2 KL^2 + KCL$$

Also from the mean effective pressure,

$$\eta_m \left(\frac{P_{in} - P_{out}}{2} \right) \cdot A_s \cdot L = 3.20 (\tau_p + \tau_R)$$

For the return stroke, assuming 10% of energy stored in the spring goes in opening the inlet valve and 90% to the hydraulic pump. Therefore,

$$3.05 \times 0.9 \times (\tau_p + \tau_R) = P_h A_h \cdot L$$

The equations to design the system are:

$$\eta_m \frac{(P_{in} - P_{out})}{2} A_s \cdot L = 3.20 (\tau_p + \tau_R)$$

$$3.05 (\tau_p + \tau_R) = 1/2 K L^2 + K C L$$

$$3.05 \times 0.9 \times (\tau_p + \tau_R) = P_h A_h \cdot L$$

$$P_{in} A_s + P_{out} A_s - 2 P_f A_f - (2 C + L) K = 0$$

$$[Y_o = L/2, t = \tau_p, y = L]$$

$$(2 C + L) K - 2 P_h A_h = 0$$

$$[x_o = L/2, t = \tau_R, x = L]$$

$$\therefore K = \frac{2 P_h A_h}{2 C + L}$$

$$\omega_R \tau_R = \pi$$

$$\omega_p \tau_p = \pi$$

$$\frac{\tau_R}{\tau_p} = \sqrt{1 + \frac{P_{in} - P_{out}}{K L} A_s}$$

Assuming:

Stroke of piston = 0.25 inch

Diameter of piston = 0.25 inch

Precompression = 2 inch

$$p_h A_h = 10.7 \text{ lb}_f$$

$$K = 5.02 \text{ lb/in.}$$

$$\tau_R / \tau_p = \sqrt{1 + \frac{2 \times 314}{5.02 \times 1/4} \times \frac{\pi}{64}} = 5.0$$

$$\tau_R + \tau_p = 0.112 \text{ sec}$$

$$\text{RPM} = 535$$

$$\tau_p = 5/6 \times 0.112 = 0.093 \text{ sec}$$

$$M = \left(\frac{0.093}{\pi} \right)^2 \times K g_o$$

$$= 1.702 \text{ lbs}$$

There is no apparent advantage in pumping on the return stroke.

APPENDIX K

UNIFLOW ANALYSIS

This Appendix compares the performance of a uniflow engine (exhaust ports) with that of the Thermo Electron exhaust valve which stays open the entire return stroke. The indicator diagrams for the two engines are given in Figures K-1 and K-2.

Figure K-1. Indicator Diagram
of Thermo Electron
Engine

Figure K-2. Indicator Diagram
of "Uniflow" Engine

In the analysis the following symbols will be used:

- t = time
- p = pressure
- T = temperature
- h = enthalpy
- u = internal energy
- s = entropy
- v = specific volume
- V = piston swept volume
- M = mass of working fluid
- W = engine work output

The subscripts that are used refer to:

- 1 - state of working fluid when the intake valve closes
- 2 - state of working fluid when the exhaust valve opens
- 3 - state of working fluid after it exhausts down to condenser pressure
- 4 - state of working fluid when the intake valve opens
- s - fluid exhausted from engine at each stroke
- c - fluid recompressed during return stroke

I. THERMO ELECTRON ENGINE

The work per cycle is given by:

$$W_{TECO} = \oint p dV = p_1 V_1 + \int_1^2 p dV - p_3 V_3$$

Substituting for the volume terms:

$$V_1 = M_s v_1$$

$$V_2 = V_3 = M_s v_2$$

The work per pound per cycle is given by

$$\frac{W}{M_s}_{TECO} = p_1 v_1 + \int_1^2 p dv - p_3 v_2$$

The expansion from state 1 to state 2 can be treated as an isentropic expansion for the ideal case. Then the property relationship,

$$T ds = du + p dv = 0$$

results in:

$$p \, dv = -du$$

$$\therefore \left(\frac{W}{M_s} \right)_{\text{TECO}} = p_1 v_1 + u_1 - u_2 - p_3 v_2$$

$$h = u + pv$$

$$\left(\frac{W}{M_s} \right)_{\text{TECO}} = h_1 - h_2 + v_2 (p_2 - p_3)$$

II. UNIFLOW ENGINE

In the uniflow engine, not all of the steam that expands from 1 to 2 exhausts. Some of the flow is recompressed to 4.

$$M_S \equiv \text{net steam flow exhausted}$$

$$M_C \equiv \text{recompressed steam flow}$$

Consider the steam still above the piston when the exhaust port opens (see Figure K-3). The piston can be treated as stationary during the exhaust process because the exhaust process is completed while the piston changes its position only slightly.

Figure K-3. Uniflow Engine Configuration

The first law of thermodynamics applied to Figure K-3 gives:

$$h \frac{dM}{dt} = \frac{\partial}{\partial t} (Mu)_{C.V.}$$

$$h \frac{dM}{dt} = M \frac{du}{dt} + u \frac{dM}{dt}$$

$$(h - u) \frac{dM}{dt} = M \frac{du}{dt}$$

$$\therefore \frac{dM}{M} = \frac{du}{h - u}$$

The volume relationship gives:

$$V = Mv = \text{constant}$$

$$Mdv + v dM = 0$$

$$\therefore \frac{dM}{M} = - \frac{dv}{v}$$

Substituting:

$$\frac{dv}{v} = \frac{du}{h - u} = - \frac{du}{pv}$$

$$\therefore du + pdv = 0$$

The property relationship for entropy is:

$$T ds = du + p dv = 0$$

Therefore, this is an isentropic expansion for the gas above the piston from 2 to 3. However, no net work is obtained as in expanding against the piston as from 1 to 2.

In order to avoid inefficient mixing, the conditions at state 4 should be the same as at state 1. Since the expansion from state 1 to state 3 was isentropic, an isentropic compression from the pressure at state 3 to the pressure at state 1 will result in the optimum state 4.

The work output for the net exhausted flow, M_s , is the same as for the TECO engine.

$$W_{M_s} = M_s [h_1 - h_2 + v_2 (p_2 - p_3)]$$

The recompressed portion of the flow, M_c , has the same work output. However, the recompression work input exceeds the expansion work. The work of recompression is $(h_1 - h_3)$ for the flow M_c .

$$\therefore W_{M_c} = M_c [h_1 - h_2 + v_2 (p_2 - p_3) - (h_1 - h_3)]$$

These expressions can be added to give the net work output per pound for the uniflow engine.

$$\left(\frac{W}{M_s} \right)_{\text{UNIFLOW}} = h_1 - h_2 + v_2 (p_2 - p_3) - \frac{M_c}{M_s} \left[(h_2 - h_3) - v_2 (p_2 - p_3) \right]$$

A comparison of the expressions for the work per pound of net flow per cycle for the Thermo Electron engine and the uniflow engine readily shows that the Thermo Electron engine is always more efficient. Its work output exceeds the uniflow work output by the expression:

$$\frac{M_c}{M_s} \left[\left(h_2 - h_3 \right) - v_2 \left(p_2 - p_3 \right) \right]$$

The magnitude of the various terms can be demonstrated graphically. The term M_c/M_s can be expressed in terms of specific volume by using the volume relationship $V_2 = V_3$.

$$V_2 = \left(M_s + M_c \right) v_2 = V_3 = M_c v_3$$

$$\therefore \frac{M_c}{M_s} = \frac{v_2}{v_3 - v_2}$$

A fluid property diagram of pressure versus specific volume for an isentropic expansion can be used to illustrate the various parameters (see Figure K-4).

Figure K-4. Pressure-Specific Volume Diagram

The shaded area represents $\left(h_2 - h_3 \right) - v_2 (P_2 - P_3)$.

The unshaded area between the dotted horizontal lines represent

$$h_1 - h_2 + v_2 (P_2 - P_3).$$

(Note that h_3 represents enthalpy above the piston. The outlet "quasi-steady" enthalpy h_3 is a value between h_2 and h_3 .)

APPENDIX L

THERMAL LOSSES

Types of Heat Loss

In analyzing the total heat loss of the engine, three types of heat transfer losses must be considered: conduction, convection, and radiation.

Conduction losses will occur essentially from the heat source (boiler) to the heat sink (condenser). The paths the heat flux will take are the vacuum seal divider, the engine exhaust pipe, and the engine cylinder support. Conduction losses through the steam piston and shaft due to piston ring contact with the cylinder are negligible.

The convection losses will occur within the steam cylinder due to the temperature difference between the steam and the cylinder wall-piston and in the exhaust line due to the temperature difference between the exhausted steam and the exhaust pipe-piston. The basic convection heat transfer equation is: $q = hA(T_w - T_\infty)$. The quantity "q" is the total heat transferred from a hot wall to a cool fluid flowing past the wall. The other quantities are defined as follows:

A ≡ total surface area of wall

h ≡ convection heat transfer coefficient from wall to the particular fluid

T_w ≡ absolute temperature of wall

T_∞ ≡ absolute temperature of fluid

The coefficient "h" is difficult to determine except by experiment. In the case of the steam in the steam engine cylinder, T_∞ is a function of time and this further complicates the problem. Therefore, no attempt will be made to quantitize the loss. However, two things can be said qualitatively. The first thing is that the cylinder wall should always be hotter than the steam. This means that heat will be transferred from the wall

to the steam rather than from the steam to the wall. Keeping the wall temperature above the steam temperature is essential in obtaining a high thermodynamic cycle efficiency. Otherwise cylinder condensation occurs and efficiency drops. Secondly, the cylinder wall temperature distribution can be adjusted over a wide range by adjusting the relative heat conductions of the engine parts.

Radiation losses, like the convection losses, are difficult to analyze. Figure L-1 shows that radiation can occur within the steam engine cylinder. The basic equation for radiation heat transfer is:

$$q = F_{\epsilon} F_G \sigma A_1 (T_1^4 - T_2^4).$$

The quantity "q" is the total heat radiated from body "1" to body "2". The quantity in parentheses is the difference between the fourth power of the absolute temperatures of body "1" and body "2". The other quantities are defined as follows:

- $A_1 \equiv$ radiating area of body "1"
- $F_{\epsilon} \equiv$ emissivity function which depends on the color and texture of surfaces "1" and "2"
- $F_G \equiv$ geometric view factor which depends on the angle between the two radiating surfaces.
- $\sigma \equiv$ proportionality constant (Stefan Boltzmann constant)

Figure L-1. Radiation Within Steam Cylinder.

In Figure L-1, most of the radiation will occur between surface 1 and surface 4 and between surface 2 and surface 3 because the view factor and temperature difference are the highest between these surfaces. A complication is that surfaces 3 and 4 are not always exposed because of the motion of the piston. In addition F_g is difficult to determine except by experiment. However, it will be low because the cylinder will be lined with low emissivity chromium.

Without experimentation, only a qualitative analysis can be made. Some of the energy radiated will be absorbed by the steam, raising the steam temperature. The rest of the energy will be radiated from the top of the cylinder down to the bottom tending to reduce the temperature gradient along the cylinder. Radiation will be limited because of the chromium lining and piston motion.

Radiation heat losses may also occur between convolutions of a single engine part such as the vacuum seal divider or cylinder support. However, by making these parts out of polished metal, F_g can be minimized. In addition the radiation will be limited because the temperature differences are not as great as those involved in the engine cylinder.

Radiation losses will be minimized because of the relatively low operating temperature of the Rankine cycle (recall that radiation losses are proportional to the fourth power of the absolute temperature). In addition, the compact engine design limits the area of highest temperature.

First Approach - Minimizing Conduction Losses

The first approach taken was to minimize the conduction losses by making the thermal conduction of all engine parts as small as possible. The general formula for thermal conduction is:

$$G = \frac{KA}{L}$$

In this formula, L is the length of heat flow, K is the thermal conductivity of the specific material, and A is the cross-sectional area of heat flow.

Therefore, to minimize heat conduction losses for each engine part, L must be large and K and A must be small. The length of heat flow L is limited by the maximum size of the capsule. However, in some cases a longer heat path can be achieved without increasing the overall size

of the package by convoluting the parts. The cross-section area, A , cannot be made too small. Some parts must have a certain minimum cross-sectional area so that they will not fail under stress. Other parts have a minimum A due to machining limitations. The thermal conductivity, K , is limited by the materials available. Any material used must be able to withstand a maximum temperature of 900°F . It must be strong enough so that what is gained by a low thermal conductivity is not lost by the necessity for a large cross-sectional area. Titanium, whose thermal conductivity is $.118 \text{ watt/in-}^{\circ}\text{F}$, is the best material available for this application.

Figure L -2 shows the electrical analog for the engine heat conduction losses. The quantities are defined as follows:

T_{EX}	$\equiv$	temperature of exhaust pipe at outlet		
R_{ins}	$\equiv$	thermal resistance of vacuum foil insulation		
R_{uc}	$\equiv$	"	"	" upper cylinder
R_{LC1}	$\equiv$	"	"	" lower cylinder (lower portion)
R_{LC2}	$\equiv$	"	"	" lower cylinder (upper portion)
R_{DIV}	$\equiv$	"	"	" vacuum seal divider
R_{EX}	$\equiv$	"	"	" exhaust pipe
$R_{\text{cyl sup}}$	$\equiv$	"	"	" cylinder support
ΣQ	$\equiv$	total conduction heat loss		
Q_{ins}	$\equiv$	heat flux through vacuum foil insulation		
Q_{uc}	$\equiv$	"	"	" upper cylinder
Q_{LC}	$\equiv$	"	"	" lower cylinder (lower part)
Q_{DIV}	$\equiv$	"	"	" vacuum seal divider
Q_{EX}	$\equiv$	"	"	" exhaust pipe
$Q_{\text{cyl sup}}$	$\equiv$	"	"	" cylinder support

Figure L-2. Conduction Heat Loss Analog

The heat source (boiler) temperature is 900°F and the heat sink (condenser) temperature is 140°F. Heat conduction through the piston rings has been neglected. The vacuum seal divider can be attached either at the bottom of the upper cylinder or at the exhaust pipe.

Table L - 1 is a list of the thermal resistances of the various engine parts derived by minimizing A and maximizing L within the constraints described. The thermal conductivity of titanium was used in all cases. Since all parts are essentially hollow cylinders, their cross-sectional area, A , was approximated as $\pi 2R\Delta r$ (see Figure L-3).

Figure L-3. Approximate Cross-Sectional Area of Hollow Cylinder

Table L - 1.

Thermal Resistances of Engine Parts

Part	K	L	2R	Δr	$R = \frac{L}{K2\pi R \Delta r}$
insulation	$\rightarrow 0$	---	---	---	$\rightarrow \infty$
upper cylinder	$.118 \frac{\text{watt}}{\text{in-}^\circ\text{F}}$	.2"	.55"	.010"	$98 \frac{^\circ\text{F}}{\text{watt}}$
lower cylinder (top)	"	.8"	.55"	.010"	$392 \frac{^\circ\text{F}}{\text{watt}}$
lower cylinder (bot.)	"	.6"	.55"	.010"	$294 \frac{^\circ\text{F}}{\text{watt}}$
divider	"	4.2" (convoluted)	.8"	.005"	$2.86 \times 10^3 \frac{^\circ\text{F}}{\text{watt}}$
exhaust	"	4. "	.125"	.005"	$17.25 \times 10^3 \frac{^\circ\text{F}}{\text{watt}}$
cylinder support	"	2.0" (convoluted)	.85"	.020"	$318 \frac{^\circ\text{F}}{\text{watt}}$

The analog is redrawn in Figure L-4 with the calculated resistances. In the model, temperature is analagous to voltage, heat flux is analagous to current, and of course, thermal resistance is analagous to electrical resistance. All heat fluxes and tempe ratures can be found using general circuit theory. The actual calculations will be omitted here. The results are summarized in Table L-2.

Figure L-4. Conduction Analog Parameters

Table L - 2.

Summary of Conduction Losses - I

upper cylinder:	$Q_{uc} = .915$ watts
vacuum seal divider:	$Q_{DIV} = .235$ watts
lower cylinder:	$Q_{LC} = .68$ watts
exhaust pipe:	$Q_{EX} = .023$ watts
cylinder support:	$Q_{cyl\ sup} = .657$ watts
total heat loss:	$\Sigma Q = .915$ watts
exhaust temperature:	$T_{EX} = 542^\circ F$

In order to determine whether the steam in the cylinder is always cooler than the cylinder wall, the temperature of the steam, T_s must be determined (see Figure L-5).

Figure L-5. Change in Steam Temperature

$x \equiv$ instantaneous displacement of piston from top dead center
 $S_s \equiv$ total displacement of piston during first half of cycle
 $v \equiv$ velocity of piston
 $t \equiv$ elapsed time from top dead center

$$v = \frac{\text{RPM}}{60} (2S_s) = \frac{\text{RPM} \cdot S_s}{30} ; t = \frac{x}{v} = \frac{30x}{\text{RPM} \cdot S_s}$$

This result is valid if it is assumed that $d^2 x/dt^2 \approx 0$. Figure L-6 shows what this means. The steam engine flywheel will tend to make $d^2 x/dt^2$ small. Therefore, to simplify this analysis, the assumption will be made.

Figure L-6. Piston Displacement vs. Time

The thermodynamic process involved is polytropic. The equation describing the process is: $T_1 v_1^{n-1} = T_2 v_2^{n-1}$

$T_i, v_i \equiv$ the Rankine temperature and steam volume respectively at any arbitrary point, i , during the piston stroke.

$n-1 \equiv$ constant which describes the process

$$T_2 = T_1 \left(\frac{v_1}{v_2} \right)^{n-1} \quad n-1 = .241 \text{ (from efficiency estimates)}$$

$$T_1 = 1360^\circ\text{R (inlet steam temperature)} \quad \therefore T_2 = T_s, v_2 = v_s$$

$$T_s = 1360^\circ R \left(\frac{v_1}{v_s} \right)^{.241} \quad v_1 = \pi V_p^2 X_1 \equiv \text{initial volume of steam}$$

$$v_s = \pi V_p^2 (X + X_1) \quad \frac{v_1}{v} = \frac{X_1}{X + X_1} \quad \text{at BDC: } \frac{X_1}{X + X_1} = \frac{1}{20}$$

$$\text{at BDC: } X = S_s \quad \frac{1}{20} = \frac{X_1}{S_s + X_1} \quad X_1 = \frac{S_s}{19}$$

$$T_s = 1360 \left(\frac{X_1}{X + X_1} \right)^{.241} = 1360 \left(\frac{S_s/19}{S_s/19 + \frac{\text{RPM} \cdot S_s \cdot t}{30}} \right)^{.241}$$

$$\text{or simplifying: } T_s = 1360^\circ R \left(\frac{1}{1 + (19/30) \text{RPM} \cdot t} \right)^{.241}$$

If $\overline{\text{RPM}} = 500/\text{min}$, then the results can be summarized as follows:

$$\text{downstroke: } T_s(t) = 1360^\circ R \left(\frac{1}{1 + 316.67t} \right)^{.241} \quad 0 \leq t \leq .06$$

$$\text{upstroke: } T_s(t) = \text{constant} = T_s(.06) \quad .06 \leq t \leq .12$$

Figure L-7 shows a plot of $T_s(t)$. During the downstroke T_s decreases and during the upstroke T_s remains constant. The cylinder wall temperature, T_w , must now be determined as a function of x . The Figure L-8 illustrates the simple method. The temperature, T_w , will change linearly from the inlet to bottom dead center (BDC). The temperature of the wall at BDC can be determined from the heat conduction analog.

Figure L-8. Determining T_w as a function of x .

$$S_s = .45 \text{ in.} \quad X_1 = \frac{S_s}{19} = .0237 \text{ in.}$$

$$T_{EX} = 542^\circ\text{F} = 1002^\circ\text{R.} \quad \Delta T = 358^\circ\text{R (TDC to Exhaust)}$$

$$T_{BDC} = 1360^\circ\text{R} - \left(\frac{S_s + X_1}{.9 \text{ in.}} \right) 358^\circ\text{R} = 1172^\circ\text{R.}$$

$$T_{X_1} = 1360^\circ\text{R} - \left(\frac{X_1}{.9 \text{ in.}} \right) 358^\circ\text{R} = 1350.6^\circ\text{R}$$

11-7

Figure L-7. Steam and Cylinder Wall Temperatures.

Bringing Exhaust Temperature to 660°R

Figure 1 is a thermal circuit diagram of a gas turbine engine. It shows a network of thermal resistances (R) and heat flows (Q). On the left, a heat source at 900°F provides heat ΣQ . This heat flows through a resistance R_{uc} ($4/7$) to a junction. From this junction, heat flows through R_{ins} ($\rightarrow \infty$) to a cold sink at $T_c = 140^{\circ}\text{F}$, and through R_{uc} to a junction at $T_{EX} = 200^{\circ}\text{F}$. From T_{EX} , heat flows through R_{LC} ($3/7$) to a final junction, and through R_{DIV} ($\rightarrow \infty$) to the cold sink. The final junction has heat flows through R_{EX} ($\rightarrow \infty$) to the cold sink and through $R_{cyl\ sup}$ to a cylinder. The cylinder has heat flows $Q_{cyl\ sup}$ and $R_{cyl\ sup}$. A dashed line indicates a resistance R_l between the T_{EX} junction and the cold sink.

Figure L - 9 shows the conduction analog with constraints. The most obvious ones are $T_{EX} = 200^{\circ}F = 660^{\circ}R$ and $R_{ins} \rightarrow \infty$ (and consequently $Q_{ins} \rightarrow 0$). The less obvious constraints are $R_{DIV} \rightarrow \infty$ and $R_{EX} \rightarrow \infty$. However, since $60^{\circ}F$

must be dropped across equivalent resistance R_1 compared to 700°F across $R_{uc} + \frac{4}{7} R_{LC}$, R_1 must be less than one-tenth of $R_{uc} + \frac{4}{7} R_{LC}$. To obtain this thermal resistance ratio, $\frac{3}{7} R_{LC}$ and R_{sup}^{cyl} can be made much smaller than they were in the previous analysis. This effectively "shorts out" R_{DIV} and R_{EX} which are much larger than $R_{uc} + \frac{4}{7} R_{LC}$ and much larger than $\frac{3}{7} R_{LC} + R_{sup}^{cyl}$. With these assumptions, the simplified analysis follows.

The thicknesses of the upper cylinder and upper portion of the lower cylinder will be kept at 10 mils (.01").

$$\therefore R_{uc} + \frac{4}{7} R_{LC} = 98 \frac{^\circ\text{F}}{\text{watt}} + 392 \frac{^\circ\text{F}}{\text{watt}} = 490 \frac{^\circ\text{F}}{\text{watt}}$$

$$\Delta T = 760^\circ\text{F} \qquad \Sigma R = R_{uc} + \frac{4}{7} R_{LC} + R_1$$

$$\frac{490}{\Sigma R} (760^\circ\text{F}) = 700^\circ\text{F} \qquad \Sigma R = 532 \frac{^\circ\text{F}}{\text{watt}}$$

$$R_1 = R_{sup}^{cyl} + \frac{3}{7} R_{LC} = 42 \frac{^\circ\text{F}}{\text{watt}} \qquad \Sigma Q = \frac{\Delta T}{\Sigma R} = 1.43 \text{ watts}$$

The result shows that both R_{sup}^{cyl} and $\frac{3}{7} R_{LC}$ should have a small thermal resistance. The most convenient way to decrease thermal resistance is to increase the cross-sectional area of the member. This gives an added advantage since greater cross-sectional area implies greater structural support.

Table L-3 is a summary of the conduction heat loss using the analog of Figure L-9. The total conduction heat loss is 50 percent higher than the previous case. However, there will be little or no convection heat loss at the exhaust. In the previous case, the cool steam carried heat away from the cylinder exhaust.

Table L - 3

Summary of Conduction Losses - II

upper cylinder:	$Q_{uc} = 1.43 \text{ watts}$
vacuum seal divider:	$Q_{DIV} = 0$
lower cylinder:	$Q_{LC} = 1.43 \text{ watts}$
exhaust pipe:	$Q_{EX} = 0$
cylinder support:	$Q_{cyl \sup} = 1.43 \text{ watts}$
total heat loss:	$\Sigma Q = 1.43 \text{ watts}$
exhaust temperature:	$T_{EX} = 200^{\circ}F$

Figure L-11 shows that the wall temperature of the cylinder is always higher than the temperature of the stream. Thus, the two desired design features have been achieved.

Another feature which may be added is glass filament to reinforce the cylinder. The filament would be wound around the cylinder in such a way that the sides of individual hoops do not contact. Figure L-10 shows how the filament would be arranged. The glass fiber would help to relieve the hoop stress on the cylinder due to the high steam pressure within. However, because of the fiber's low thermal conductivity and low contact area, the added heat conduction losses would be negligible.

Figure L-10. Cylinder Wound Glass Fibers

Figure L-11. Steam and Cylinder Wall Temperatures - Improved Design.

APPENDIX M

LEAKAGE ANALYSIS

All working pistons in the implantable power system will have losses due to fluid leaking past them. These pistons include that of the steam engine, hydraulic pump, feedwater pump, sump pump, and high pressure reservoir.

As currently envisioned the steam piston will be the only one with rings. All others will be straight pistons within precision machined cylinders. A separate leakage analysis must be done for both the piston ring and straight piston cases.

Piston Without Rings

In this analysis laminar flow is assumed. The quantities are defined as follows:

$D \equiv$ diameter of cylinder (in.)

$C \equiv$ radial clearance between piston and cylinder (in.)

$t_o \equiv \frac{C}{2}$, (in.)

Figure M-1. Cross Section of Straight Piston in Cylinder

Figure M-2. Laminar Flow Through Radial Clearance Along Cylinder Length.

- u $\equiv$ velocity of fluid at any point (in./sec)
 τ $\equiv$ shear stress between two layers of fluid ($\text{lbs}_f/\text{in.}^2$)
 t $\equiv$ ordinate of laminar flow path
 x $\equiv$ abscissa of laminar flow path
 p $\equiv$ pressure of fluid at any point ($\text{lbs}_f/\text{in.}^2$)
 ρ $\equiv$ density of fluid ($\text{lbs}_m/\text{in.}^3$)
 N $\equiv$ viscosity of fluid $\left(\frac{\text{lbf} \cdot \text{sec}}{\text{in}^2} \right)$
 L $\equiv$ total length of piston (in.)
 Δp $\equiv$ pressure drop across piston length ($\text{lbs}_f/\text{in.}^2$)

Figure M-3. Incremental Length of Cylinder, dx , unfolded into Parallel Plates.

The cross section of a straight piston within its cylinder is shown in Figure M-1. The laminar flow which is assumed to exist through the radial clearance along the length of the cylinder is shown in Figure M-2. The radial clearance, $2t$, around the circumference of the cylinder can be unfolded into two parallel plates. Figure M-3 shows this for an incremental cylinder length, dx . The summation of forces in the x direction must be equal to zero.

$$\tau (\text{shear area}) = dp (\text{pressure area})$$

$$\tau (2dx) (\pi D) = dp (2t \pi D)$$

$$\tau = \left(\frac{dp}{dx} \right) t = \eta \frac{\partial u}{\partial t}$$

$$\int_u^{u=0} \frac{\partial u}{\partial t} = \frac{1}{\eta} \left(\frac{dp}{dx} \right) \int_t^{t_o} t dt$$

$$-u = \frac{1}{2\eta} \left(\frac{dp}{dx} \right) \left(t_o^2 - t^2 \right)$$

$$u = \left[\frac{(dp/dx)}{2\eta} \right] t_o^2 \left[\left(\frac{t}{t_o} \right)^2 - 1 \right]$$

$$\frac{dp}{dx} = - \frac{\Delta p}{L}$$

$$u = \frac{\Delta p t_o^2}{2\eta L} \left(1 - \left(\frac{t}{t_o} \right)^2 \right)$$

$$\bar{v} \equiv \frac{1}{2t_o} \int_{-t_o}^{t_o} u dt \equiv \text{mean velocity of fluid across clearance}$$

$$W_{LEAKAGE} \equiv \text{amount of fluid leaked} \quad \frac{\text{lbs m}}{\text{sec}} = \rho \bar{V} A$$

$$A = \pi D (2 t_o)$$

$$\rho \bar{V} A = \rho \pi D (2 t_o) \frac{1}{2 t_o} \int_{-t_o}^{t_o} u dt$$

$$W_{LEAKAGE} = \rho \pi D \int_{-t_o}^{t_o} u dt$$

$$= \frac{\rho \pi D \Delta p t_o^2}{2 \eta L} \left[t - \frac{t^3}{3 t_o^2} \right]_{-t_o}^{t_o}$$

$$= \left(\frac{\rho \pi D}{2 \eta L} \right) \Delta p t_o^2 (2) \left[t_o - \frac{t_o}{3} \right]$$

$$= \left(\frac{2 \rho \pi D}{3 \eta L} \right) \Delta p t_o^3$$

$$\boxed{W_{LEAKAGE} = \left(\frac{2 \pi \rho}{3 \eta} \right) \left(\frac{D}{L} \right) t_o^3 \Delta p} \quad \text{for a straight piston}$$

within a tightly fitted cylinder.

All of the quantities can be considered as constants except D , L , and t_o . Therefore, the leakage losses can be minimized not only by a small clearance but also by a small D/L ratio. In order for steam leakage not to seriously degrade the cycle efficiency, the clearance, t_o , would have to be maintained below 0.002 inch.

Piston Ring Case

The formula derived above can be applied to the case of a steam piston with no rings. The Δp across the piston will be large when the inlet valve opens - about 400 psi. The viscosity of the steam, N , is much lower than that of water. These two factors will cause a significant amount of leakage. This will have a greater effect on the overall efficiency than would the same percentage of leakage at a pump piston. No mathematical analysis is presently available because of the complexity of such an analysis. However, piston rings are used on the implantable power source engine because experience in building larger steam engines (1500 watt) has demonstrated that piston rings can greatly decrease leakage.

Materials

In the straight piston with no rings, leakage can be minimized by using a steel piston in a graphite cylinder liner. Experience has shown this to significantly reduce leakage if the piston is working on a liquid.

The best material for the steam cylinder is titanium because of its high strength, low thermal conductivity, fabricability and compatibility with water. The larger steam engines built by Thermo Electron have given satisfactory results using steel rings on a graphite cylinder liner. However, the graphite liner has a high thermal conductance and cannot be used in the heart engine where heat loss is critical. For the present application graphite rings in a chromium plated titanium cylinder will significantly reduce steam leakage and limit cylinder heat loss.

Leakage Loss Calculations

$$\% \text{ Leakage} \equiv \frac{V_L \times 100\%}{V_p}$$

$$V_p \equiv \text{volume of pump} = AS, (\text{in.}^3)$$

$$A \equiv \text{area of pump piston} (\text{in.}^2)$$

$$S \equiv \text{stroke of pump} (\text{in.})$$

$$V_L \equiv \text{volume of fluid leaked} (\text{in.}^3)$$

Assume that the pump downstroke duration is equal to the upstroke duration.

$$N \equiv \text{cycles/sec of steam engine} = \frac{\text{RPM}}{60}$$

$$T \equiv \text{duration of downstroke} = \frac{1}{2} \times \frac{1}{N} = \frac{30}{\text{RPM}}, (\text{sec})$$

$$V_L \equiv W_{\text{leakage}} T / \rho = 30 W_{\text{leakage}} / \rho (\text{RPM})$$

Therefore, the % leakage will decrease as RPM increases.

Hydraulic Pump: $A = .11 \text{ in.}^2$, $S = .45 \text{ in.}$

$$V_p = AS = (.11)(.45) \text{ in.}^3 = .0495 \text{ in.}^3 = 4.95 \times 10^{-2} \text{ in.}^3$$

$$W_{\text{LEAKAGE}} = \left(\frac{2\pi\rho}{3\eta} \right) \left(\frac{D}{L} \right) t_o^3 \Delta p$$

$t_o = .0002''$

$$D = .375''$$

$$L = .5''$$

$$\Delta p = 80 \text{ psi}$$

$$\rho = \frac{61.4 \text{ lbs m}}{\text{ft}^3} = \frac{61.4 \text{ lbs m}}{1728 \text{ in.}^3} = 3.55 \times 10^{-2} \frac{\text{lbs m}}{\text{in.}^3} / T_c = 140^\circ \text{F}$$

T_c = condenser temperature

$$\begin{aligned} \eta &= 1.14 \frac{\text{lb m}}{\text{ft-hr}} = \frac{1.14 \text{ lb m}}{12 \text{ in} \cdot 60 \text{ sec}} = \left(\frac{1.14}{720} \right) \left(\frac{1}{32.2 \times 12} \right) \frac{\text{lb f} \cdot \text{sec}}{\text{in.}^2} \\ &= 4.1 \times 10^{-6} \frac{\text{lb f} \cdot \text{sec}}{\text{in.}^2} \end{aligned}$$

$$W_{\text{LEAKAGE}} = \frac{2\pi(3.55) \times 10^{-2}}{3(4.1 \times 10^{-4})} \cdot \frac{.375}{.5} (2 \times 10^{-4})^3 80 \frac{\text{lbs m}}{\text{sec.}}$$

$$(1.81 \times 10^4)(.75)(8.0) \times 10^{-6} (.80) = 8.72 \times 10^{-6} \frac{\text{lbs m}}{\text{sec}}$$

Let $\text{RPM} = \overline{\text{RPM}} = 500 \text{ cycles/min.}$

$$V_L = 30 (8.72 \times 10^{-6}) / (3.55 \times 10^{-2} \times 500) = \left(\frac{242.16}{17.75} \right) \times 10^{-6} \text{ in.}^3$$

$$V_L = 13.68 \times 10^{-6} \text{ in.}^3$$

$$\% \text{ Leakage} = \left(\frac{13.68 \times 10^{-6}}{4.95 \times 10^{-2}} \right) \times 100\% = \boxed{.0276\% = \% \text{ Leakage}}$$

Feedwater Pump: $A = .012 \text{ in.}^2$, $S = .040''$

$$V_p = AS = (1.2)(4) \times 10^{-4} \text{ in.}^3 = 4.8 \times 10^{-4} \text{ in.}^3$$

$$\boxed{t_o = .0002''} \quad D = .125'' \quad L = .2'' \quad \Delta p = 400 \text{ psi}$$

$$\begin{aligned}
 W_{\text{LEAKAGE}} &= (1.81 \times 10^4) \left(\frac{.125}{.2} \right) \left(2 \times 10^{-4} \right)^3 400 \frac{\text{lbs m}}{\text{sec}} \\
 &= 1.81 \times 10^4 (.625) (.8 \times 10^{-5}) (4.00) \frac{\text{lbs m}}{\text{sec}} \\
 &= 3.62 \times 10^{-5} \frac{\text{lbs m}}{\text{sec}}
 \end{aligned}$$

$$V_L = 30 (3.62 \times 10^{-5}) / (3.55 \times 10^{-2} \times 500) = \frac{108.7}{17.75} \times 10^{-5} \text{ in.}^3 = 6.2 \times 10^{-5} \text{ in.}^3$$

$$\% \text{ Leakage} = \frac{6.12 \times 10^{-1}}{4.8 \times 10^{-4}} \times 100\% = \boxed{12.76\% = \% \text{ Leakage}}$$

If

$$t_o = .0001''$$

,

$$\% \text{ Leakage} = 1.6\%$$

Sump Pump: $A = .012 \text{ in.}^2$, $S = .040''$, $V_p = 4.8 \times 10^{-4} \text{ in.}^3$

$$t_o = .002''$$

$$D = .125''$$

$$L = .2''$$

$$\Delta p = 15 \text{ psi}$$

$$W_{\text{LEAKAGE}} = (1.81 \times 10^4) (.625) (.8 \times 10^{-6}) 1.5 = 1.36 \times 10^{-6} \frac{\text{lbs m}}{\text{sec}}$$

$$V_L = 30 (1.36 \times 10^{-6}) / (3.55 \times 10^{-2} \times 500) = \frac{40.8}{17.75} \times 10^{-6} = 2.29 \times 10^{-6} \text{ in.}^3$$

$$\% \text{ Leakage} = \left(\frac{2.29 \times 10^{-2}}{4.8 \times 10^{-4}} \right) \times 100\% = \boxed{.478\% = \% \text{ Leakage}}$$

High Pressure Reservoir. $A = 1 \text{ in.}^2$ $S = \frac{V_o}{A}$

$V_o =$ nominal hydraulic volume needed to pump blood once (in.^3)

$$S = \frac{.15 \text{ in.}^3}{1 \text{ in.}} = .15 \text{ in.}$$

$$V_p = AS = (1 \text{ in.}^2)(.15 \text{ in.}) = V_o = .15 \text{ in.}^3$$

$$\boxed{t_o = .0002"} \quad D = 1.125" \quad L = .25" \quad \Delta p = 65 \text{ psi}$$

$$\begin{aligned} W_{\text{LEAKAGE}} &= \left(1.81 \times 10^4 \right) \left(\frac{1.125}{.25} \right) \left(2 \times 10^{-4} \right)^3 65 \frac{\text{lbs m}}{\text{sec}} \\ &= 1.81 \times 10^4 (4.5)(.8) \times 10^{-5} \times .65 \frac{\text{lbs m}}{\text{sec}} \\ &= 4.23 \times 10^{-5} \frac{\text{lbs m}}{\text{sec}} \end{aligned}$$

$$V_L = 30 (4.23 \times 10^{-5}) / 17.75 = \left(\frac{126.9}{17.75} \right) \times 10^{-5} = 7.15 \times 10^{-5} \text{ in}^3$$

$$\% \text{ Leakage} = \left(\frac{7.15 \times 10^{-2}}{1.5 \times 10^{-1}} \right) \times 100\% - \boxed{.0475\% = \% \text{ Leakage}}$$

Summary:

	<u>t_o (Radial Clearance/2)</u>	<u>% Leakage</u>
Hydraulic Pump	.0002"	.0276%
Feedwater Pump	.0002" (.0001")	12.76% (1.6%)
Sump Pump	.0002"	.478%
High Pressure Reservoir	.0002"	.0475%

Steam Piston with No Rings

$$T_B \equiv \text{boiler temperature} = 900^\circ\text{F}$$

$$T_C \equiv \text{condenser temperature} = 140^\circ\text{F}$$

$$T_{EX} \equiv \text{exhaust temperature} = 200^\circ\text{F}$$

$$\bar{T} \equiv \text{average temperature of steam} \quad \frac{T_B + T_{EX}}{2} = 550^\circ\text{F}$$

$$P_{MEP} \equiv \text{mean effective steam pressure} = 58.2 \text{ psi}$$

$$\text{For steam at } 550^\circ\text{F and } 58.2 \text{ psi: } N = 4.3 \times 10^{-7} \frac{\text{lb f} - \text{sec}}{\text{ft}^2}$$

$$\eta = \frac{430 \times 10^{-9} \frac{\text{lb f} - \text{sec}}{\text{in}^2}}{144 \text{ in}^2} = 2.98 \times 10^{-9} \frac{\text{lb f} - \text{sec}}{\text{in}^2}$$

$$v = \frac{1}{\rho} = 10.2 \frac{\text{ft}^3}{\text{lb m}} = (10.2) 1728 \frac{\text{in}^3}{\text{lb m}} = .176 \times 10^5 \frac{\text{in}^3}{\text{lb m}}$$

$$\rho = 5.67 \times 10^{-5} \frac{\text{lb m}}{\text{in}^3}$$

$$W_{\text{LEAKAGE}} = \left(\frac{2\pi\rho}{3\eta} \right) \left(\frac{D}{L} \right) t_o^3 \Delta p$$

$$\Delta p = p_{MEP} = 58.2 \frac{\text{lbs f}}{\text{in}^2} \quad \boxed{t_o = .0002 \text{ in.}}$$

$$D = .5" \quad L = 1.0"$$

$$\begin{aligned} W_{\text{LEAKAGE}} &= \left(\frac{2\pi(5.67 \times 10^{-5})}{3 \times 2.98 \times 10^{-4}} \right) \left(\frac{1}{2} \right) \left(8 \times 10^{-12} \right) 58.2 \frac{\text{lbs m}}{\text{sec}} \\ &= (3.98 \times 10^4) \left(\frac{1}{2} \right) (.80 \times 10^{-5}) .582 \frac{\text{lbs m}}{\text{sec}} \end{aligned}$$

$$W_{\text{LEAKAGE}} = .925 \times 10^{-5} \frac{\text{lbs m}}{\text{sec}}$$

$$V_L = 30 W_{\text{LEAKAGE}} / \rho \text{ (RPM)} = 30 (.925 \times 10^{-5}) / (5.67 \times 10^{-2} \times .500) \text{ in.}^3$$

$$V_L = \left(\frac{27.75}{2.83} \right) \times 10^{-3} = 9.81 \times 10^{-3} \text{ in.}^3$$

$$V_p = AS = (.196 \text{ in.}^2) (.45 \text{ in.}) = .0881 \text{ in.}^3 = 8.81 \times 10^{-2} \text{ in.}^3$$

$$\% \text{ Leakage} = \left(\frac{.981}{8.81} \right) \times 100\% = \boxed{11.13\% = \% \text{ Leakage}}$$

$$I. \quad \boxed{t_o = .0001" \quad \% \text{ Leakage} = 1.39\%}$$

The analysis shows that rings must be used on the steam piston if cycle efficiency is to be maintained. An attempt to achieve an effective clearance value, t_o , of one-tenth mil without piston rings would probably cause the engine to "sieve" on heating because of differential thermal expansion. Piston ring design, as evidenced in the development of the internal combustion engine, is largely empirical. The modern automobile power plant as well as Thermo Electron's 1500-watt steam engine show that piston rings can greatly reduce the effective clearance between the piston and cylinder.