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**EVALUATION OF PRACTICABILITY OF A RADIOISOTOPE
THERMAL CONVERTER FOR AN ARTIFICIAL HEART DEVICE**

Phase 1 Final Report

April 1972

TRW Systems Group
Redondo Beach, California

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EVALUATION OF PRACTICABILITY OF A RADIOISOTOPE THERMAL CONVERTER FOR AN ARTIFICIAL HEART DEVICE

PHASE 1 FINAL REPORT

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Abstract

The objective of the program was to determine the practicability of developing a radioisotope thermal converter for an artificial heart device. The thermal converter, including the radioisotope heat source, and all associated power conditioning and control systems are to be fully implantable and capable of functioning for ten years with high reliability. The device must supply the necessary mechanical, hydraulic, or pneumatic power for a blood pump which provides 100% of the left ventricular pumping function.

The principal design groundrules included a daily average power level of 2.8 watts delivered to the blood pump and the following design maxima: 60-watt heat source, 1.5-liter volume, 3.0-kilogram weight, and peak power level of 4.44 watts.

The concepts evaluated included various mechanizations of gas reciprocating, vapor reciprocating, vapor rotary, and thermoelectric cycles, as well as various combinations of these devices. A total of eight different thermal converter systems were found capable of meeting all the design groundrules.

These eight systems were compared using previously-defined evaluation criteria and scoring procedures. The preferred concept is an all-electric-output hybrid engine which consists of a thermoelectric stage operated thermally in series and electrically in parallel with an organic vapor turbogenerator. The hybrid engine powers a motor/reciprocator unit which in turn mechanically actuates a Kwan-Gett-type blood pump. The weight and volume of the entire thermal converter (exclusive of the blood pump) are 1.83 kg and 1.33 liters. The thermal converter requires a 49-watt heat source and has an overall heat-to-mechanical conversion efficiency of a little over nine percent.

1. INTRODUCTION AND SUMMARY

The overall objective of the program is to determine the practicability of developing a fully implantable radioisotope thermal converter to power a heart-assist pump that is capable of assuming 100 percent of the left-ventricular pumping function, and has a high probability of operating acceptably for 10 years. The objective of this first phase of the study was to evaluate all the possible concepts and recommend a preferred system for more detailed evaluation in Phase II.

The Phase I practicability evaluation of a radioisotope thermal converter was carried out under the design groundrules listed in Table 1-1. In addition to these physical and performance specifications, we imposed the requirement that all candidate systems utilize fibrous thermal insulation

Table 1-1. Prime Design Groundrules

• Power supply to support blood pump in 100%-assist left ventricular functional replacement mode (i.e., to take over 100% of left ventricular work not simply capture 100% of cardiac output)	
• Nominal average output power (to 60% efficient Kwan-Gett-type of blood pump)	2.81 watts
• Maximum heat source	60 watts (thermal)
• Output power range	2.22 to 4.44 watts according to daily power profile
• Energy storage	5.58 watt-hours at input to blood pump
• Maximum blood flow rate capability	12 liters/minute
Blood pressure at pump output	100 mm mercury (mean)
Maximum pump cycle rate	120 cycles/minute
• Package	fully implantable, totally self contained
Maximum total volume (except blood pump)	1.5 liters
Maximum Weight	3.0 kilograms
• Durability	system lifetime 10 years minimum with reasonable level of reliability
• Filling performance	left arterial pressure maintained between 0 and 18 mm mercury gauge

with an inert gas fill since we do not feel confident that vacuum superinsulation can be made reliable enough for 10-year operation. We also imposed the requirement that the thermal converter surface area be adequate to permit rejecting all waste heat directly to the body fluids. We assumed a value of 0.07 watt/cm^2 for the allowable heat flux across the container. However, we did assess the effects of using superinsulation or alternate waste heat rejection techniques.

We found a total of eight candidate thermal converter systems capable of meeting all the design groundrules. The main characteristics of these eight systems are summarized in Table 1-2. More complete descriptions are contained in Section 5 of this report. When they are compared on the basis of 24 evaluation criteria, the all-electric hybrid thermoelectric and rotary vapor system (without a battery) emerged as the most satisfactory system concept.

Since the candidate designs using a thermal to electrical conversion stage generate a constant level of power, any energy storage must be downstream of the main thermal engine. State-of-the-art nickel-cadmium storage batteries are too heavy, while the promising high-energy-density solid electrolyte batteries are not yet sufficiently well developed. In a certain sense, one can consider the thermoelectric stage in the selected design as substituting for the energy storage function on an interim basis

There is one particular major system-level advantage to all of the candidate electrical systems. This pertains to their ability to efficiently match the variation in the load on the system during the period of a single ejection stroke. During this first phase of the study, we assumed, as directed, that the blood pump operates against a constant backpressure of 100 mm Hg. But in reality, the pressure varies from about 80 mm at the beginning of systole to something between 120 and 180 mm Hg at the end of systole depending on the output power level. Therefore, the candidate approaches with constant-force rather than constant displacement characteristics must be preset to deliver the output at peak rather than average pressure. All except the electrical candidates have this mismatched constant-force type of output characteristic. Since the excess work is not recoverable, the true power requirements for these engines will be somewhere between 20%

Table 1-2. Candidate Systems

	Synchronous		Nonsynchronous		
	TESM	None	TESM	Battery	None
Gas Reciprocating	Long life engines with mechanical output requires design concepts not yet proved in this size range		48.9 watts 1.23 liters 1.75 kg 120 bpm		54.4 watts 1.36 liters 1.57 kg 120 bpm
Vapor Reciprocating	48 watts 1.42 liters 1.78 kg Variable Frequency 60 - 120 bpm	57 watts 1.40 liters 1.34 kg 120 bpm	Large heat source required		Large heat source required
Rotary Vapor				41 watts 1.06 liters 1.46 kg 120 bpm (most compact)	Most simple design but too inefficient
Thermoelectrics				54 watts 1.38 liters 1.51 kg 120 bpm	Theoretically most-reliable but large heat source required
Thermoelectrics and Rotary Vapor				37 watts 1.4 liters 1.84 kg 120 bpm (most efficient)	49 watts 1.33 liters 1.83 kg 120 bpm (most practicable)

and 80% greater than calculated according to the Phase I groundrules.

For the all-electrical systems, however, the electromechanical energy storage and transfer characteristics are such that the system will follow the variations in the backpressure and, therefore, the blood pump energy requirements can legitimately be calculated for the specified 100 mm average backpressure.

1.1 DESCRIPTION OF PREFERRED SYSTEM CONCEPT

The hybrid thermal converter system is comprised of three major components, namely:

- Hybrid Heat Engine
- Motor/Reciprocator Unit
- Automatic Actuator

This system has projected weight of 1.83 kg and occupies a total volume of 1.33 liters.

1.1.1 Hybrid Heat Engine

The hybrid heat engine subsystem (Figure 1-1) includes a 49-watt heat source which incorporates a vent and capillary tube assembly for release of the helium vented from the heat source into the abdominal cavity. Mounted on each end of this cylindrical capsule are two cascaded thermoelectric converters. These converters are composed of 24 silicon-germanium high-temperature couples operating with a hot junction temperature of 1742°F (950°C) and a cold junction temperature of 932°F (500°C). The cascaded low-temperature converter consists of 60 2N/TAGS couples operating at a hot junction temperature of 887°F (475°C) and a cold junction temperature of 464°F (240°C). Under these conditions, these converters provide an output power of 3.21 watts_e at 3.64 volts and an overall efficiency of 6.55%. Combined with this thermoelectric converter is a rotary vapor cycle turbogenerator operating at a peak cycle temperature of 430°F . This temperature is compatible with the cold junction temperature of the 2N/TAGS thermoelectric couples. The rotary vapor cycle system uses CP-34 (thiophene) as a working fluid. Operating between a peak cycle temperature of 430°F (at 250 psia) and a condensing temperature of 116°F (4 psia),

this unit is capable of providing 7.24 watts_e at 15 volts and achieves an overall efficiency of 15.75%. The main components of this rotary vapor cycle system are shown in Figure 1-1. When implanted in the body, the engine will be mounted with the turbogenerator shaft in the vertical axis. The turbogenerator operates at 96,000 rpm and is similar in size and design to units already built by Airezsearch Manufacturing Co. which operate at speeds up to 220,000 rpm.

The engine unit is insulated with fibrous Min-K insulation filled with xenon gas. The waste heat is rejected through condenser tubes on the inside of the titanium container. The surface areas of both of the container packages is sufficient to permit heat rejection to the local body tissues at flux levels less than 0.07 watt/cm².

In order to combine the outputs from the two electrical generators, the output from the thermoelectric unit must be converted to 15 volts, reducing the electrical power output from 3.21 to 2.61 watts_e. The overall conversion efficiency (thermal to electrical) achieved by the hybrid heat engine is 20.5%.

Figure 1-1. Hybrid Engine

Figure 1-2. Motor/Reciprocator Unit

1.1.2 Motor/Reciprocator Unit

The motor/reciprocator unit is shown in Figure 1-2. This unit consists of a brushless dc motor operating at 15 vdc and 9,000 rpm. The motor is similar in design to units used for the Minuteman guidance and control gyrocompass. These units have been operating continuously at 16,000 rpm for over 4 years now. The rotor is of the permanent magnetic field type and the stator resembles a conventional armature winding without a mechanical commutator. The motor operates on gas bearings in an hermetically sealed housing. The motor is coupled to a set of reduction gears through a magnetic coupling. The reduction gears decrease the motor output speed to 120 rpm and drive a rotary cam. This rotary motion is converted to reciprocating motion by a cam-follower operating in a continuous elliptical groove provided on the inner surface of the rotary cam. The cam follower imparts reciprocating motion to a rod which has an overall stroke of 1.3 inches. A bellows seal is provided to retain the cam and reciprocating cam follower lubricant. An electronic control system is also provided to implement various control functions. This is shown in Figure 1-3. A turbogenerator speed control is provided.

Figure 1-3. Electronic Controls Block Diagram

When the output voltage, which is directly related to generator speed, rises above a certain set point, a circuit consisting of a filter, a comparator with hysteresis, and a power switch, turn on a parasitic load. When the load is added, the turbogenerator speed is reduced which reduces the output voltage. The circuit is configured to provide speed control within $\pm 2\%$. As was previously mentioned, a dc-to-dc converter is included to increase the thermoelectric converter output voltage from 3.64 to 15 volts. To avoid a mismatch of impedances with the two parallel power supplies, a load-sharing electronics circuit is provided. This permits switching from one power source to the other at a very high rate. Finally, a motor overspeed control is used to ensure constant speed output from the brushless dc motor. With the exception of the speed-controller, all the electronic controls will be mounted on the motor-cam reciprocator system. The overall efficiency of the motor/reciprocator is 59%. Several alternate motor/reciprocator configurations are discussed in Section 3.1.4.2.

1.1.3 Automatic Actuator

The combined blood pump/actuator unit is shown in Figure 1-4. The dimensions of the pump were derived by scaling the Kwan-Gett design for a stroke volume displacement of 125 mls. (Volumetric efficiency assumed to be 80% and a blood volume per stroke of 100 ml - 12 l/min at 120 cpm.) We have considered it permissible to include a part of the blood pump actuator within the physical envelope of the pump unit, provided that we do not change the functional or material interface between the blood pump and the blood.

The electrical motor/reciprocator unit generates a mechanical back-and-forth motion which is transmitted to the pump actuator through a flexible plastic-coated braided metal cable. A spring within the actuator unit

Figure 1-4. Combined Blood Pump/
Automatic Actuator Unit

provides a force bias which effectively maintains the cable in tension which is the preferred operating mode during all phases of the pumping cycle.

With this design, no sensors are required to maintain control of the pumping action, the output power is automatically modulated by a variation in the pumping duty cycle (between 25 and 50% at a fixed rate of 120 beats/minute). The duty cycle is automatically regulated by the novel actuator design and the controlled constant-speed characteristic of the electric motor.

The most important feature of the automatic actuator is the additional fluid reservoir which is contained behind the flexible membrane located inside the perforated casing which forms the lower surface of the combined blood pump/actuator unit. This flexible reservoir, or make-up reservoir as shown on the figure, allows the power piston in the unit to be physically decoupled from the blood pumping membrane. Therefore, the piston goes through a displacement cycle which is not locked to the blood

inflow rate and the resulting displacement of the pumping membrane. Thus the power piston is continuously cycled through a fixed volumetric displacement at a constant rate of 120 cycles per minute.

At the onset of the filling phase, the pumping membrane and the reservoir membrane are maximally displaced. As the power piston begins its downward displacement, the blood pump begins to fill at a rate determined by the left atrial pressure, and therefore, corresponds to the physiological demand. If this inflow rate is less than the maximum, the constant suction generated by the power piston will begin to reduce the pressure in the blood pump. However, as soon as the pressure begins to try to fall below body ambient, the membrane enclosing the make-up reservoir begins to collapse, maintaining the pump pressure very close to the body reference pressure.

By the end of the filling cycle, the make-up reservoir has been depleted by an amount equal to the difference between the maximum stroke volume and the value corresponding to the actual demand at this time. When the power piston reverses direction, the ejection phase begins with initial flow back into the low back-pressure of the make-up reservoir. When the reservoir chamber is refilled, the pressure rises until the forward-flow valve in the blood pump opens and the blood is discharged into the aorta. At the completion of the piston upstroke, the pumping membrane is returned exactly to the maximal displacement position, and all of the blood which was admitted during filling is discharged.

In short, the actuator automatically delivers a partial stroke, according to the physiological demand; it provides the positive filling action necessary to maintain system control ("passive autoregulation") with the mandatory antisuction control; and it presents a very simple mechanical interface to the actuator drive unit.

Careful management of the fluid volume behind the power piston is necessary in order to avoid significant power losses. A gas fill (nitrogen or carbon dioxide) which is contained in a high quality rubber membrane sac (butyl or possibly viton) is nominally used to accommodate the volumetric change, partly by expanding into a membrane-enclosed space surrounding the power connection cable and partly by gas compression.

The power piston itself is specially designed so that it fits very closely behind the blood pumping membrane, leaving only a very thin layer of liquid to even out the pumping stresses and to preclude direct physical contact during pumping. The overall diameter of the piston is about 4 inches and it requires a stroke of 1.33 inches to displace the full 125 ml. The piston has a double membrane construction with a small trapped volume of liquid or light grease. This feature prevents the membranes from tending to fold or wrinkle as the effective length of the membranes change during the pumping cycle. With the trapped volume, the membranes are smoothly bowed out into a more convex configuration in the intermediate positions. At the top and bottom positions, the inner membrane is pulled almost flat. Without the double layer, the swept volume would be reduced by the membranes being deflected into a reversing concave shape by the flow resistances of the displaced fluids.

The overall efficiency of the automatic actuator is 76%. This value, coupled with the 59% efficiency of the motor/actuator, transforms the 9.85 watts of electrical power from the hybrid engine into the required 4.44 watts of hydraulic power into the blood pump. Models of the hybrid engine, motor/reciprocator, and automatic actuator are shown together in Figure 1-5.

1.2 GROWTH CAPABILITY

One of the most attractive aspects of our preferred concept is its growth capability. As can be seen in Figure 1-6, there are a number of different ways that the system can "grow" in terms of performance and reliability. (The hybrid system already has the highest reliability rating among the candidates).

One option would be to go to a pure rotary system when minor improvements are made in either engine, pump, or actuator efficiency. The heat source size required to do this with presently predicted or assumed efficiencies is only a few watts over the 60-watt limit.

Other opportunities arise if the high-energy-density solid electrolyte batteries now under development at TRW (sodium-sulfur) and Argonne National Laboratory (lithium-selenium) become state of the art. (The

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Figure 1-5. Thermal Converter

Figure 1-6. Growth Capability

energy density of NiCd batteries is too low to permit their utilization.) One option would be to retain the hybrid design, but reduce the heat source to 37 watts. Two other options are to go pure rotary or pure thermoelectric. The latter option would probably yield the highest possible system reliability.

If 10-year-lifetime superinsulation packages become available, the potential exists for a further reduction in size and weight. However, we have found that for the more compact systems, a decreased insulation thickness will not result in a corresponding reduction in converter volume since the converter size is close to optimum when specific gravity and allowable heat flux factors are taken into account.

1.3 SENSITIVITY ANALYSIS

An analysis was carried out to determine the sensitivity of our selection process to subjective assumptions. While the initial percentage-point spread among the top candidates was small, all the logically induced perturbations in scoring left the hybrid system in the top-ranked position by an increased margin.

1.4 CONCLUSIONS

In summary the preferred candidate approach can be seen to demonstrate several clear advantages.

- All of the components are capable of providing 10-year durability using existing, demonstrated state-of-the-art technology.
- The output of the system provides an efficient match with the load characteristic of the positive displacement blood pump. This reduces the heat source requirement by a factor of almost two compared with non-matched systems.
- The novel actuator design efficiently provides overall system control using the experimentally-proved 'passive autoregulation' approach.
- The design approach is technically flexible and capable of incorporating and taking advantage of technology such as solid electrolyte batteries which will probably be developed in the relatively near future.

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2. SYSTEM DESIGN CONSIDERATIONS

2.1 GROUND RULES

The prime design groundrules for the Phase I study are listed in Table 2.1-1. In contrast with previous design studies, these groundrules emphasize the capability of the design for long life and high reliability in addition to acceptable functional performance, minimum size and weight, and high overall thermal conversion efficiency. Since the objective of the project is to demonstrate practicability, in addition to feasibility, we have deliberately emphasized systems utilizing technology with good development predictability.

Considering the key reliability questions of wear-out and premature failure, which are discussed in some detail in Section 4, it is clear that these fundamental system qualities tend to be in conflict with several specified performance goals. Reliability can be enhanced by conservative design and carefully controlled fabrication at the expense of parameters such as size, weight and unit production cost.

As a first step towards the reliability goals, all of the stressed parts in the systems, and the seals in particular, must be designed with conservative stress margins. This requirement will be particularly hard to meet in the case of the synchronous reciprocating gas cycle engines because of the relatively high force levels which must be developed by the prime mover during the ejection phases. The mechanical stresses are lower in the nonsynchronous reciprocators because the stroke work levels are lower at the higher engine speeds. The technical feasibility of the nonsynchronous gas cycle engines has been demonstrated in thermal engine programs that have been underway for several years. Several key areas relating to the question of a practical design for long life and high reliability are still under study.

In looking for alternate candidates, a system with an all-static thermal converter is an obvious approach with a clear potential reliability edge, although the nature of the system output dictates that all of the systems will have at least some moving parts. Another area where recent advances relating to long-life dynamic converters justify

Table 2.1-1. Prime Design Groundrules

- Power supply to support blood pump in 100%-assist left ventricular functional replacement mode (i.e., to take over 100% of left ventricular work not simply capture 100% of cardiac output)
- Nominal average output power 2.81 watts
(to 60% efficient Kwan-Gett type of blood pump)
- Maximum heat source size 60 watts (thermal)
- Output power range 2.22 to 4.44 watts according to specified daily power profile
- Energy storage capacity 5.58 watt-hours, at input to blood pump
- Maximum blood flow rate capability 12 liters/minute
 - Blood pressure at pump output 100 mm mercury (mean)
 - Maximum pump cycle rate 120 cycles/minute
- Package fully implantable, totally self-contained
 - Maximum total volume (except blood pump) 1.5 liters
 - Maximum total weight 3.0 kilograms
- Durability system lifetime 10 years minimum with reasonable level of reliability
- Filling Performance left arterial pressure maintained between 0 and 18 mm mercury gauge.

consideration for this application is that of hydrodynamic bearings for rotating machinery. These techniques which are now considered to be state-of-the-art for commercial products, have excellent promise for very low wear and long life.

2.2 SYSTEM TECHNOLOGY

The terminology which has been used throughout this study is shown in Figure 2.2-1. The total thermal converter system which is the mechanism by which thermal energy from the radioisotope heat source is converted to another energy form suitable for pumping blood is subdivided into two primary subsystems:

- The engine subsystem which is defined as containing the prime power converter (or engine) and engine modulator (if any), and the energy storage unit (thermal energy storage material or electrochemical battery).
- The power conditioning and control subsystem which is defined as containing the power conditioning units (if any), all necessary control sensors and signal conditioners, and the blood pump actuator.

2.3 ALTERNATE CONCEPTUAL APPROACHES

There are several basically different approaches to designing a thermal converter system to power a positive displacement type of artificial blood pump. The primary power converter can be designed to generate pulsatile power that can be applied directly to the pump in a so-called synchronous operating mode; or the main power unit can be designed to take a role more like that of a "boilerhouse" supplying a more or less constant level of power that is converted by a second stage into the required pulsatile output format. Systems of this second type are said to operate in the nonsynchronous mode.

Other basic differences include, of course, the type of thermal engine used for the primary conversion and the different ways and extent to which the various systems can store energy in order to most efficiently meet the need to vary both the mean and the instantaneous output power demands. This latter feature which relates to the time-varying nature of the load presented to the system by this type of blood pump, has been generally neglected in virtually all of the documented

Figure 2.2-1. System Terminology

system studies to date. Yet in terms of the impact on overall efficiency, this particular characteristic could prove to be more significant than the differences in conversion efficiency between most of the candidate heat engines.

Another generally neglected system-level consideration also relates to the time-varying nature of the load presented by the blood pump and that is the discontinuous volumetric displacement that will result from a compressible pneumatic link at any point in the power transmission chain. A direct consequence of this factor is the serious disadvantage inherent in the use of a synchronous reciprocating gas cycle engine, such as the free piston modified Stirling engine, with an output in the form of a gas pressure drive.

Thus several important general conclusions regarding selection of the best system design approach can be drawn from consideration of the output power requirements and the time-varying nature of the load presented by the blood pump.

2.3.1 System Load Characteristics

There are three aspects to the load characteristic which have particular bearing on maximizing the performance of the total system. First, the back-pressure during the blood ejection stroke varies typically between 80 and 120 mm Hg at the lower levels of physical activity and may reach peak levels as high as 160 to 180 mm Hg during the more active periods. As a result, systems that produce constant force levels (which include most of the candidate approaches), must be set so that they are providing excessive force (and therefore excessive power) for some part of the time.

A second important aspect of the load characteristic is the step-function changes in the blood pump back pressure between the end of the ejection phase and the beginning of the filling phase, and again between the end of the filling phase and the beginning of ejection. If there is a compressible pneumatic link anywhere in the power transmission train between the prime mover and the blood pump, the finite time required to change the pressure in this link either up or down, leads to a differential pressure situation which causes the blood pump to "pause." In the

case of a synchronous reciprocating gas cycle engine, we found that the combined sum of these "dwell" periods could not be reduced to less than 50% of the time available in the total cycle. Half of the period allocated to ejection passed before the power piston in the actuator driving the blood pump could begin actual ejection. A similar situation was found to occur on the filling stroke with the power piston stationary for the first half of this period also. This phenomenon is caused by the approximately sinusoidal nature of the output pressure drive. There appeared to be no way to design this type of engine to provide an output which is more of a square wave, without generating excessive power and thus sacrificing a great deal of conversion efficiency.

The third aspect is one which is fairly commonly acknowledged, and that is the variation in the mean power demand associated with different levels of physical activity. Systems that have the ability to store considerable amounts of energy at some part of the conversion chain can, to a greater or lesser extent depending on the storage capacity, deliver levels of power which match the daily variation in demand. Systems with no long-term storage capacity must continuously generate power at the level corresponding to the maximum specified demand.

There will always be a practical problem associated with specifying the amount of storage necessary to provide for not only different life styles of the recipients, but day-to-day and longer-term variations in the activity profiles of different individuals. Therefore, energy storage systems capable of efficiently providing additional storage over and above the reasonable minimum are preferred. There are two practical methods of storing energy within the system and these are discussed more specifically in Section 2.8.

2.3.1.1 Output Force Matching

Figure 2.3-1 shows a simplified model of the system load which illustrates the essential characteristics being discussed. During the ejection phase, the back pressure reflected back through the blood pump to the output power piston rises from a low (diastolic) level to a peak (systolic) level, and the work performed during this period is substantially that required to pump the blood up to this potential energy level.

The mean functional head which is used to compute the equivalent stroke work represents an average value of the back pressure during the ejection period.

Thus the force required to just overcome the back pressure and eject the blood follows the changing back pressure and likewise increase during the period of ejection. If a constant force is applied to the power piston during ejection, then it must be set high enough to overcome the peak back pressure otherwise it will stall and fail to discharge some fraction of the stroke volume. Ensuring that all of the blood is always discharged during periods of increased activity when the peak back pressure may rise

Figure 2.3-1. Simplified Model of Load Characteristics

to say 180 mm Hg, requires that the system be set so that it is providing a considerable excess of force, and therefore power, for most of the time

The output characteristic which comes closest to the ideal would eject the blood at a near constant flow rate. This requires that the power piston be driven as from a crank with near-constant angular velocity. A rotary-drive system, with mechanical inertia sufficient to absorb the

back pressure variation without appreciably changing speed, has such a characteristic and thus matches the delivered power to the instantaneous demand. By way of contrast, the synchronous ram steam engine, the synchronous expansion steam engine, hydraulic and pneumatic pressure storage systems, solenoid and piezoelectric systems all have mismatched output force characteristics. The overall efficiency penalty associated with this force mismatch can be estimated by comparing the back pressure corresponding to the actual preset force level (which should be close to 180 mm Hg), with the daily average of the mean functional back pressure, which is specified as 100 mm Hg. Another drawback of systems with a mismatched force output is that the excess power is dissipated kinetically in the blood during the ejection period, contributing to the risk of mechanically damaging the blood cells.

2.3.1.2 Pneumatic Power Drive

If the pressure wave output from a synchronous reciprocating gas cycle engine is used to drive the power piston, the filling and ejection phases of the pumping cycle are significantly shortened by adverse pressure relationships which cause the power piston, and thus the blood pump, to "pause." The resulting distortion of the blood pump displacement characteristic is shown in Figure 2.3-2.

The second undesirable feature of pneumatic driving is common to both synchronous systems, where gas or vapor is used in the coupling between the engine piston and the output power piston, and the non-synchronous systems which store fluid energy at a constant pressure. The compression work contained in the fluid used as the positive displacement medium cannot be efficiently recovered, since the compressed gas must be removed rapidly at the end of the ejection stroke in order to achieve the low back pressure in the blood pump required for adequate filling performance.

In current pneumatic-powered blood pumping systems using external power supplies, this extra energy is discarded by dumping the gas to a large sink such as the atmosphere in order to meet the filling phase requirements.

Figure 2.3-2. "Dwell" Periods With Gas Drive

In summary, consideration of the nature of the system load characteristics lead us to the following general conclusions:

- (1) Systems which drive the output power piston at, or close to, constant angular velocity with short-term energy storage in the form of a mechanical "flywheel," or the equivalent, are capable of providing the ideal load matching characteristic. All the electrical candidates have been designed with this characteristic.
- (2) Systems with force-limited output drive characteristics, such as the nonsynchronous designs that use stored fluid energy must be designed to meet the maximum (systolic) back pressure rather than the mean pressure levels. This results in an additional efficiency factor which cannot be greater than 0.83 (i. e., 100/120) and may be as low as 0.56 (i. e., 100/180) in order to perform adequately under the specified range of physical activity levels.
- (3) Synchronous engines must be designed with a mechanical or hydraulic (incompressible) coupling medium between the engine piston and the output power piston in order to avoid dwell periods and a consequent reduction in both filling and ejection periods.
- (4) Pneumatic links in the output power train lead to additional power losses resulting from nonrecovery of the compression work.

2.3.1.3 Summary of General System Considerations

The approximate overall system efficiency multipliers to take account of the losses associated with both lack of energy storage capacity and output force mismatches are given in Table 2.3-1. The 0.70 value represents a value between the numbers derived earlier (0.56 and 0.83) and corresponds to a peak (or systolic) pressure of 143 mm Hg.

The most obvious potential advantage of systems using a synchronous engine is avoiding the compounding of two or more stages of power conversion. However, under the realistic groundrules of this study, the potential advantage in overall conversion efficiency is apparently outweighed by a number of other factors relating to the difficulty of designing a single-stage converter that adequately meets all of the system requirements. The nonsynchronous design approach allows

Table 2.3-1. Impact of Load Matching and Energy Storage Characteristics on Overall System Efficiency

	<u>System Efficiency Equals</u>
All of candidates using electrical generators and battery storage are perfectly matched.	1.0 x power conversion efficiency
Synchronous and non-synchronous reciprocating engine candidates using TESM can accommodate daily changes in mean output power but cannot match changes in back pressure during pumping.	0.70
Electrical generator candidates without battery storage cannot accommodate changes in mean output power level.	0.60
Synchronous and non-synchronous reciprocating engine candidates without TESM cannot accommodate either changes in mean output power or changes in back pressure during pumping.	0.42

consideration of non-reciprocating thermal converters and selection of an optimum speed range for the engine.

Particular design problems with synchronous engines include achieving high efficiency when the output power is modulated over a 2:1 range, efficiently controlling the power modulation and achieving adequate control of the back pressure and volumetric variations during the filling phase.

2.4 WASTE HEAT MANAGEMENT

Physiological studies using several different species of experimental animals have established guidelines for the levels of heat flux that can be tolerated at the surfaces of artificial devices implanted within the body. These findings are discussed in detail in Section 2.4.2.

2.4.1 Heat Generation Rates

Generally it has been assumed that at least some of the waste heat from an artificial heart device power supply must be transferred directly into the main blood stream through either a tube-type of heat exchanger placed in series with the blood pump, or by a special modification of the pump itself. The rationale for this approach appears to be, first, that there is a natural, physiological precedent since the heart itself gives up 10 to 15 watts of heat to the throughflow; and second, the logical consideration that with only a small temperature rise acceptable, the greatest level of heat rejection can be attained where the mass flow rate of coolant is greatest. The experimental evidence summarized in 2.4.2 is that heat fluxes in the range of 0.8 to 0.9 watt/cm² can be rejected to the main blood flow while the limiting rate to the surrounding musculature and other body tissues is in the range of 0.05 to 0.10 watt/cm².

Consideration of several factors relating to the groundrules and requirements of this specific study, however, lead us to the following general conclusions on which we propose an initial design approach to waste heat management which is simply to reject all of the waste heat through the metal container to the surrounding body tissues.

- For a 70-kgm subject, the maximum allowable thermal inventory of the heat source is specified to be 60 watts. The surface area of a sphere equivalent to the specified volumetric limit of 1.5 liters is 634 cm^2 . The corresponding mean flux level of 0.095 watt/cm^2 , indicates the possibility of achieving mean thermal flux levels in the experimentally verified range of 0.05 to 0.10 watts/ cm^2 .

Actual candidate system sizes are below the 1.5-liter limit but they also use heat sources less than 60 watts and package geometries that have surface areas greater than the volume-equivalent sphere.

- The design goal of an overall specific gravity close to unity will tend to maintain this heat source/surface area ratio fairly constant, even as overall system efficiency is improved during development.
- Even with a special-purpose heat exchanger in the system design, there will be an inevitable amount of heat leakage (estimated by an earlier study contractor to be about 40 to 50%) through the insulation and into the surrounding tissues. This is particularly true within our system design groundrules since the low specific gravity requirement makes the use of the less compact fibrous insulation a very attractive design approach. Because of these heat-leakage paths, it would be difficult to transport more than about 50% of the waste heat to a separate heat-exchanger.

There is a disadvantage in using a heat exchanger in the blood pump because unlike the natural heart which rejects the greater part of its thermal load to the coronary circulation and thus directly into the blood circulating through the lungs, the heat rejected to a left heart assist pump would be carried into the systemic circulation and thus largely back into the abdominal cavity. An abdominally located tube-type heat exchanger would be a logical engineering solution to this question. The arguments against this approach center largely around the risk associated with another site of surgical anastomosis, the inclusion of another potentially damaging material interface with the blood, and the additional complication at installation. Our medical consulting group was unanimous in their preference for using the outer wall of the container as the prime body heat exchanger.

In addition to these special system considerations, there is a good physiological case for rejecting heat via the conduction and convection paths provided by the surrounding tissues and body fluids. The heat production in the trunk core usually accounts for over half of the body's basal heat production so there are established natural physiological

pathways for significant levels of heat flow from this region. The always-present tissue fluids ensure good thermal coupling between the implanted package and the surrounding muscles and viscera. Perhaps the most pertinent experimental evidence to date on allowable heat flux levels comes from long-term studies carried out on dogs by the Boston Group (Reference 7). Two isotope sources (16 and 24 watts) were attached to tube-type heat exchangers that were connected into the abdominal aorta. The blood-exposed surface had an area of 11.4 cm^2 and the external surface to which the tissue was exposed had an area of about 120 cm^2 . According to one of the authors (page 909, reference 7), as much as 50% of the heat was rejected through the surface areas surrounded by tissue. For the two heat source sizes, this results in a chronic thermal flux level of 0.07 and 0.10 watt/ cm^2 , respectively in the tissues. The experiments were continued for 22 and 26 months, respectively, before they were terminated for causes not related to the experimental exposure. The tissues showed no effects that could be attributed to the exposure. The balance of the heat was rejected directly to the blood stream via the inside surface of the implant at a flux density of 0.7 and 1.0 watt/ cm^2 , respectively.

The Pierce Foundation group specifically studied this mode of dissipation using electrically-heated flat packages implanted within the abdominal cavity of 40-kg sheep. They found that at levels of about 0.6 watt/kgm (equivalent to 42 watts for a nominal 70 kg human) the animals equilibrated with only marginal increases in body temperature. Again, the excess heat was rejected via the respiratory system, since sheep are not provided with any body surface heat rejection mechanisms.

2.4.2 Related Experimental Studies

The experimental studies relating to maximum total levels of additional endogenous heat and maximum thermal flux densities at the tissues that can be physiologically accommodated have been carried out primarily at three centers: Battelle's Pacific Northwest Laboratories in Richland, Washington (References 1 - 3); by a group in Boston comprised of staff members from The Children's Hospital Medical Center, Sears Surgical Research Laboratories and Thermo Electron Corporation (References 4 - 8); and by the John B. Pierce Foundation Laboratories in New Haven, Connecticut (References 9 - 12).

All three groups have investigated fairly extensively the ability of dogs (and in the case of PNL, Hanford miniature swine) to accommodate tube-type blood-cooled heat exchangers connected into the descending aorta. Their findings (which included a number extending over periods of 12 to 24 months) generally supported the conclusion that the animal subjects can adapt chronically to an additional thermal load up to about 0.8 to 1.1 watts/kgm of body weight, although for the larger sources there was an initial tolerance problem during the first several days. In dogs, the excess heat is rejected via the respiratory route which is a preferred mode for the dog, rather than through the skin. If the findings could be extrapolated directly, this would correspond to a tolerable heat source in the range of 55 to 70 watts (nominal 70 kgm body weight). The PNL group's findings were somewhat less confident about thermal burdens in excess of 20 watts even though they reported 2 to 3-month survival for animals with 80- and 100-watt sources. They reported that 50 to 80 kgm swine with 60-watt sources had problems accommodating unusually warm environmental temperatures and mild infections.

None of the groups found evidence of gross physiological, biochemical or histological anomalies that could be related to the thermal burden, though there were numerous problems associated with the thrombogenic nature of the foreign material surfaces exposed to the blood and some difficulties with physical acceptance of the implanted packages.

Dr. Rawson of the John B. Pierce Foundation Laboratories is quoted on Page VIa of Reference 5 as stating that vascularized tissue can be conditioned to dissipate a heat flux of 0.1 watt/cm^2 with a source temperature of 43°C . Their study, however, exposed a very significant new facet to the question of waste heat management, which is the relatively rapid encapsulation of the warm implant with highly vascularized tissues rather than the more typical avascular, thermally insulating scar tissue capsule. Of major significance is the fact that this is currently being investigated (Reference 8) as a technique for stimulating collateral circulation in the damaged heart muscle, using local heating via a venous catheter. Previous to these reports, temperatures of about 45°C were considered noxious and potentially damaging to body tissues.

It has been speculated that the capillary formation is triggered by a local shortage of oxygen in the tissues brought about by increasing the local rate of tissue metabolism and, therefore, an excessive consumption of oxygen.

While it is not possible to state categorically at this time that large, warm implants in humans will behave in an analogous manner, these findings suggest that heat can be rejected safely to body tissues at levels of about 0.07 watts/cm^2 . However, if an alternate mode of heat rejection is preferred, the most attractive approach is to add a heat exchange surface to the blood pump. Adopting this design option would increase the volume of the nominal selected design by about 4% and the total weight by about 6.6%. As discussed later, there would be a negligible increase in the power required.

2.4.3 Peak Heat Rejection Rates

The heat sources for the eight candidate systems that were finally evaluated lie in a range between 37 and 57 watts. For the physical body size of the recipient model specified for the study, there is evidence from experiments with animals to the effect that these levels are acceptable as a chronic thermal body burden. However, how the body will adapt to temporary thermal loads higher than this for relatively short periods throughout the day is not so easy to predict. Systems using thermal energy storage reject heat at levels from 28 to 56% higher than the mean during periods of maximum physical activity. The mean is approximately the same as the thermal power of the heat source.

At first consideration, it might be argued that in relative terms this is not likely to be a problem, since the body itself is generating more heat at these times and the natural heat dissipation channels are fully opened. However, there could be a problem in coupling this extra heat from the site in the abdominal cavity into the natural heat distribution channels because of a redistribution of the blood flow between the different body segments. During exercise, the flow to the abdominal organs tends to increase only marginally with significant increases in the flow to the skin and the outer body layers. Unless there is a natural compensation, the additional rejected heat could lead to locally elevated temperatures. This question does not arise with the non-modulated systems or those using battery storage since the rejected heat level does not vary appreciably at different output power levels.

2.5 PACKAGING CONSIDERATIONS

Some packaging considerations cannot be adequately evaluated at this time. For example, conformity of the package with body spaces in the chest and the abdomen, the exact nature of the interconnections between the packages, and the nature of any external surface mountings have been assumed to be design considerations common to all of the candidates. Therefore no attempt has been made to differentiate the candidates on these points. Other considerations, including overall volume, specific gravity and maximum anterior-posterior dimension, have been specifically evaluated.

For consistency in arriving at a set of comparable overall system designs we have followed a set of common guidelines.

2.5.1 Packaging Configuration

Within the volume, specific gravity, A-P dimension and to some extent, the surface area constraint associated with heat rejection to the surrounding body tissues, we have assumed that the complete system will consist of at least two interconnected parts. The pumping unit must be located as close as possible to the site of the natural heart, while the power unit itself can probably be more readily accommodated in the lower abdomen. Since those preferred sites are at least 5 to 7 inches apart in the mature human, we have allowed nominal configurations with a third, smaller package located in the intermediate area. All of the package configurations are considered reasonable nominal designs in the absence of more specific requirements. They are partly responsive to certain of the primary design requirements but they are not proposed at this time as optimum configurations.

Because the power unit will be prefueled and operating at the time of installation, we have, where feasible, included the capacity to mechanically uncouple the pump drive unit (the actuator) from the power unit or power conditioner, since we considered that this would provide a convenient way of allowing the surgeon to connect and prime the blood pump. This mechanical coupling is a design feature to be explored.

2.5.2 Packaging Materials

The main package of all of the candidate systems is enclosed in a hermetically sealed can of commercially pure titanium. There are three general areas of consideration that lead to the choice of titanium. First, it is potentially capable of withstanding the corrosive environment within the body for 10 years, second it is not harmful to body tissues, and third, it can be fabricated using conventional techniques.

The body offers a surprisingly hostile environment to metals, and only those with considerable resistance to corrosion in saline solutions, such as titanium, cobalt-chromium alloys, and to a lesser extent, stainless steel can be considered. Titanium is far more resistant than

its general reactivity (as indicated by its position in the electrochemical series) would suggest. This is because, like stainless steel, it forms a protective oxide layer when it is exposed to an electrolytic solution. Of the materials which are most commonly used for internal prostheses, pure titanium is the most corrosion-resistant, being virtually immune to attack. Both stainless steel and the cobalt-chrome alloys degenerate under certain stress conditions. Stainless steel is usually the most seriously affected.

In the absence of corrosion, there is little to choose between titanium and the other materials with respect to tissue acceptability. As with all metal implants there will be a slow migration of metallic ions into the tissue. However, this slow dissolution does not appear to be harmful.

It is anticipated that the outermost surface of the implanted package will be a "jacket" of an open-weave inert textile material such as Dacron mesh impregnated with medical grade silicone rubber. This will allow the surrounding tissues to grow into the porous surface providing a strong mechanical bond.

2.6 GAS MANAGEMENT

Gaseous isotopic decay products are vented from the heat source and conducted to the outside of the power unit through a metallic capillary tube. If the vent design incorporated into the heat source is of the nonselective type a small portion of the vented gas, the radon fraction, will be radioactive. However, provided that the path length of the capillary is greater than 5 or 6 inches the calculated flow rate is such that the activity level will have decayed to an acceptable level (16 half-lives), before it reaches the outside of the container.

The major gaseous decay product is helium which is not radioactive and presents no biological hazard. It will be released at a more or less steady rate of about 2×10^{-6} standard cc per second. If all of the helium generated over 10 years within a 50-watt capsule is released from the fuel it will amount to a total of about 625 sccs.

Helium is compatible with body tissues and the proposed gas management approach is to leak the gas to the surrounding body

environment through a helium-permeable membrane. The capillary will terminate on the outside of the main container, where it will pass through a length of soft silicone rubber into the lumen of a small diameter plastic tube which will act as a reservoir from which the helium can diffuse into the surrounding body tissues. The tubing will be mounted around the outside of the container and appropriately integrated with the final Dacron-mesh and silicone-rubber "jacket." The external length of the gas-vent capillary penetrates the soft rubber stopper which will provide an adequate fluid resistant seal excluding body fluids from the gas space within the plastic tube. There are several candidate materials for the plastic diffuser such as Teflon, polyethylene, and polyethylene terephthalate which are compatible with body tissues. Assuming that only one side of the plastic acts as a diffusion "window" a 12-inch length of 1/4-inch OD polyethylene tube with a 0.050-inch-thick wall will adequately match the maximum anticipated helium venting rate.

2.7 THERMAL INSULATION

The advantages and disadvantages of fibrous and foil type insulations were investigated to determine the optimum insulation for use in the engine subsystems. The important parameters studied were weight, size, thermal conductivity, fabricability, ease of assembly, and reliability.

2.7.1 Fibrous Insulation

Min-K 2020 (Reference 13) is a nonmetallic insulation which consists primarily of submicron silica, titanium dioxide opacifier for radiation blockage, and two types of silica fibers (microquartz and astroquartz) for reinforcement. The insulation contains no binder, therefore, no preconditioning is required to remove the binder which in the past caused contamination with usage in various systems. In fact, the Min-K formulations 1999, 2002, and 2020 were specifically designed for lead telluride and silicon-germanium thermoelectric energy conversion devices to reduce excessive outgassing and potential poisoning of the thermoelectric elements by impurities within the insulation.

Thermal conductivity data are available for Min-K 2020 in air,

argon, vacuum (Reference 14) and xenon. The data are plotted in Figure 2.7-1.

Figure 2.7-1. Thermal Conductivity Versus Mean Temperature

As can be seen from this figure, the use of xenon as the Min-K fill gas results in lower insulation thermal conductivities than are attainable with air or argon fill gas.* Min-K in a vacuum, of course, has lowest conductivity; however, maintaining a static vacuum for 10 years is difficult, although the conductivity increase of fibrous insulation with increasing gas pressure is not catastrophic as it is with superinsulation. Ease of fabrication and assembly of Min-K is also an advantage since the insulation can be molded and machined into various geometries.

The nominal density of Min-K 2020 is 20 lb/ft³. Lower Min-K densities are available at the expense of mechanical strength. Load-

*However, the difference is not substantial. For the candidate concept, if argon were substituted for xenon, the heat losses would increase by 1.4 watts.

bearing requirements for the candidate thermal converter systems are small and the Min-K 2020 should be suitable for this application.

Since the Min-K insulation absorbs moisture from the air, assembly procedures would include a bakeout at temperature in a vacuum with assembly of the insulation and welding of the thermal converter system package in a dry box flooded with xenon gas. If electron beam welding is required for package sealing, the welding would be done in a vacuum and then the package back-filled with xenon.

Shrinkage does not appear to be a problem even at the highest anticipated system operating temperatures. For example, a SiGe thermoelectric converter using Min-K 2020 operating at 950-1000°C, which has been on test for 24,000 hours, shows no increase in heat loss. (Reference 15). Shorter-term tests have demonstrated no measurable shrinkage below about 850°C.

2.7.2 Foil Insulation

One type of foil insulation consists of metallic foils which have been sprayed with a thin coating of ceramic oxide. The foils are rolled or formed into cylinders with the required number of layers and then packaged into a leak-tight container and evacuated. The oxide coating has a low thermal conductivity and acts as a spacer between the metal foils. Heat from inside the insulation can only be conducted through the areas where the oxide contacts the next layer of foil and by radiation. A high vacuum (1×10^{-3} mm Hg) is required for efficient performance. This type of foil insulation is being developed by Thermo Electron Corporation.

A similar type of foil insulation is being developed by Linde Division of Union Carbide Corporation. Fibrous glass or quartz paper is used between the metal foils instead of the sprayed ceramic oxide. This type also requires a high vacuum.

The mean thermal conductivities for both of these types (References 16 and 17) are shown in Figure 2.7.2 versus source temperature. A comparison of Figures 2.7-1 and 2.7-2 shows that the thermal conductivity for the foil type insulations is lower than for the fibrous

Figure 2.7-2. Vacuum Foil Insulation Thermal Conductivity Versus Source Temperature

insulations by a factor of approximately 100. Therefore, if vacuum foil insulation were used to insulate the candidate thermal converter systems, the insulation thickness could be appreciably decreased.

The approximate densities of the various foil insulations are as follows:

<u>Foils</u>	<u>Insulation Densities (lb/ft³)</u>
Al	7.9
Stainless Steel	22.7
Ti	13.4
Cu	26.3

Comparing these densities with the Min-K density of 20 lb/ft³ shows that a vacuum foil insulation system for the candidate thermal converter systems would be a lighter weight system because of the thinner insulation thickness required; in some cases, depending on the metallic foil used, the density would also be lower.

Application of vacuum foil insulation to anything but flat surfaces is difficult. If tubes are to be brought through the insulation or if the surface to be insulated has protuberances, fabrication and assembly become complex and a good static vacuum is difficult to maintain. Also, losses at corners and edges where the foils are discontinuous substantially reduce the overall insulation efficiency. Separate structural support of the insulated components must also be provided since the load bearing characteristics of the vacuum foil insulation are poor and the heat losses increase as loading on the foils increases. Structural supports provide additional heat leaks and further reduce overall system efficiency.

2.7.3 Thermal Insulation Selection

Based on the foregoing advantages and disadvantages of fibrous versus superinsulation, a decision was made to use only the fibrous type of insulation. The reliability associated with the 10-year maintenance under static conditions of the high vacuum necessary for the effective performance of superinsulation appeared to be low. In addition, the insulation efficiency improvement potentially available with superinsulation is only partly relizable because of edge losses and the inability of the superinsulation to provide load-bearing capability. Finally, system-level analyses showed that the possible size reduction is largely offset by a considerable increase in the specific gravity of the package. However, for the sake of completeness we did compare the characteristics of the selected system concept with both fibrous insulation and superinsulation.

2.7.4 Overtemperature Protection

In the event of a system malfunction or failure, some method of preventing heat source overheating must be provided. For the selected system concept, a simple passive thermal fuse appears practical and is described in Section 3.2.5.4.

2.8 ENERGY STORAGE

By providing certain types of long-term energy storage, the engine subsystems can be designed to generate power at a level corresponding to the average daily blood pump requirements rather than the demands associated with maximum physical activities. In order to accommodate variations in the specified daily power demand profile, the energy storage device must be sufficient to deliver and store at least 5.6 watt-hours of energy, as measured at the input to the blood pump.

Mechanical, pneumatic, and hydraulic methods do not provide sufficient energy density to meet the above requirement. Energy storage in the form of thermal energy storage material (TESM) or electro-chemical storage in the form of secondary batteries are the two forms that have been considered. Because the TESH must be located at the same position in the conversion chain as the heat source, it suffers the conversion inefficiency of the entire thermal converter (engine subsystem plus power conditioning and control subsystem). The secondary battery, however, is downstream of the electrical generator and, therefore, a smaller amount of stored energy is needed to provide the equivalent of 5.6 watt-hours at the blood pump. Using typical figures for the system conversion efficiencies to illustrate this factor, we find that

- TESH requires approximately 73 watt-hours based on a 7.5% overall thermal converter efficiency.
- Batteries require approximately 12.5 watt-hours based on a 45% PCCS efficiency.

2 8 1 Thermal Energy Storage Material (TESM)

The TESH uses the latent heat of fusion of the material to absorb excess thermal energy isothermally. This process is completely reversible and theoretically there is no limit to the number of times it can be repeated. The temperature of the TESH used must be compatible with the engine and working fluid. Ideally, the heat source of the thermal converter is sized for the average power requirement of the blood pump. Then when the blood pump power is below the daily average, the TESH will absorb the excess thermal energy which will tend to melt the TESH.

During periods when the blood pump power requirements are above the daily average, the additional heat energy will be supplied by the solidification of the TESM. Utilization of TESM presumes that the engine can be operated not only in a variable power mode but at reasonably constant efficiency. Variable power can be obtained either by varying the power output per stroke at fixed frequency or varying the frequency if the power per stroke is fixed.

If the TESM is sized for the daily load profile and the recipient power profile meets the daily average, no additional controls will be required other than engine controls. However, if the daily power profile is not maintained, some device such as a heat pipe may be required to prevent heat source temperature excursions. A heat pipe could be placed between the TESM and the outside container. The heat pipe would be pressurized so that it would not transmit heat until the temperature of the TESM reaches a given value. Once the high temperature is reached, the TESM heat can be transferred via the heat pipe to this container which will reject the heat directly to the body.

The TESM materials considered for use with the candidate systems are shown in table 2.8-1. Figures 2.8-1 and 2.8-2 indicate volume and weight of the TESM material versus watt-hours.

Table 2.8-1. TESM Materials Considered

Material	Melting Temp (°F)	Cal/cc	Cal/gram
LiF/LiCl	930	362	165
Al	1216	207.5	76.8
LiF/NaF	1250	442	170
LiH	1252	480	610
Cu	1981	375	42

Figure 2.8-1. TESM Volume Versus Watt-Hours

Figure 2.8-2. TESM Weight Versus Watt-Hours

2.8.2 Electrochemical Energy Storage

The secondary electrochemical energy storage devices must be capable of supplying additional power when blood pump requirements are in excess of the daily average and storing power when the blood pump power requirements fall below the daily average. This method of energy storage is particularly attractive since the engine can be sized for the average power level and operated at fixed load. The cycle life, specific energy (w-hr/lb), and energy density (w-hr/cm³) should be as high as possible. If the secondary battery is to be in close contact with the isotope heat source, as in the case of several of the designs, then radiation effects must also be considered as must the practical problems related to controlling the charging rate of the battery.

On the basis of these requirements, three types of secondary cells are considered; nickel-cadmium, lithium-selenium and sodium-sulfur systems. Other batteries such as silver-cadmium and silver-zinc are not considered because of their poor cycle life characteristics.

Extensive development work has been performed on the nickel-cadmium battery by the aerospace industry and a great deal of performance information is available. Figure 2.8.3 shows the depth-of-discharge versus the cycle life of a nickel-cadmium cell. These data indicate that for a 20% depth-of-discharge at 100°F the nickel-cadmium cell will last for only 3000 cycles, and the low specific energy (15 w-hr/lb) indicates that a heavy unit be required. At 20% depth-of-discharge, the weight of the battery to provide 12.5 w-hours of stored energy would be about 4.0 lbs. At 40% depth-of-discharge, this weight could be reduced to 2 lbs but the cycle life would decrease to about 1500 cycles. A further drawback to nickel cadmium systems is the relatively complicated electronics required to control the charging conditions.

The other two types of batteries utilize fused salt or solid electrolyte cells which are more efficient at higher temperatures. These cells can be operated to 100% depth-of-discharge and they do not require elaborate charging control circuits. Unlike the nickel-cadmium cell which contains a polymeric material as a separator, these cells do not need a conventional separator. The lithium-selenium cell uses a LiF-LiCl-LiI eutectic in an immobilized molten state as an electrolyte. A specific energy close to

Figure 2.8-3. Secondary Battery-NiCd-Cycle Life

50 w-hrs/lb has been demonstrated at Argonne National Laboratory with the cell operating at 710°F and tested through 300 charge/discharge cycles.

The sodium-sulfur cell uses a thin film of sintered polycrystalline alumina as the electrolyte. The cell is operated at 570°F with projected specific energy of about 150 w-hrs/lb. These systems are under development at TRW, Ford, GE, and the Navy Department.

A comparison between lithium-selenium and sodium-sulfur cells shows that the sodium-sulfur cell has a number of advantages.

- It is less expensive (i.e., cost of sodium is \$0.25/lb vs. \$8.00/lb for lithium; sodium is abundant and sulphur costs about \$0.07/lb).
- It can be oriented in any direction because of complete separation of the anode and cathode compartments.
- It can be hermetically sealed because it does not offgas during charging and discharging.
- It is not sensitive to storage temperature.

- It can be discharged at both high and low rates.
- It has a long shelf life.

The neutron and gamma radiations at the levels anticipated will not damage the cell materials. Operating cells at higher temperature does not affect their performance but it may decrease reliability because of the material compatibility problems.

A sodium-sulphur cell weighing about 0.15 lb is capable of meeting the energy storage requirements for the candidate electrical systems. Figures 2.8-4 and 2.85-5 compare the weight and volume of the alternate systems.

Figure 2.8-4. Candidate Battery Weight Versus Energy Storage

Figure 2.8-5. Candidate Battery Volume Versus Energy Storage

There are still several practical problems associated with the fabrication of sodium-sulphur cells. These problems are associated with:

- Selecting the solid electrolyte (i.e., sodium aluminate at several different compositions can be used).
- Reproducibility in fabricating the electrolyte.
- Developing ceramic-to-metal seals and ceramic-to-ceramic seals.
- Selecting a proper current collector.
- Optimizing the design for minimum weight and volume.

While the required battery technology currently exists at the laboratory demonstration level, there is no off-the-shelf battery system which is capable of meeting the weight and life requirements for this application. Based on a review of the current technical status, it is estimated that a prototype battery could be developed within 2 years. A production model could be made available in about 3 years.

2.9 RADIOISOTOPE CAPSULE DESIGN

The design characteristics of the vented radioisotope capsule used throughout the study were supplied by the AEC and are shown below:

Fuel Pellet: $^{238}\text{PuO}_2$ ¹⁶ (90% ^{238}Pu)

Density	9.5 g/cc
Specific Power	0.45 watt/g

Ta-10W Container

Internal Volume taken as Pellet Volume Plus	10%
Wall Thickness	0.15 cm (0.060 in)
Density Ta-10W	16.83 g/cc

Pt-20Rh Container

Clearance between Containers taken as	0.025 cm (0.010 in)
Wall Thickness	0.05 cm (0.020 in)
Density Pt-20Rh	18.74 g/cc

The AEC-supplied values for capsule weight and volume as a function of thermal inventory are as follows:

Thermal, Power, Watts	30	40	50	60
Capsule Diameter, cm L/D = 1	2.59	2.79	2.99	3.15
Capsule Volume, cm ³	13.6	17.1	21.0	24.6
Capsule Weight, Grams	156	196	234	274

3. COMPONENT SUBSYSTEMS

3.1 POWER CONDITIONING AND CONTROL

In addition to the general requirements that cover all system components, such as long life and high reliability, there is a further set of requirements for each of the power conditioning and control subsystems determined by constraints of the blood pump interface and the converter output requirements. A third set of requirements results from the specific control and power conditioning needs of the corresponding engine subsystem.

3.1.1 Blood Pump Interface

The Kwan-Gett type of blood pump was specified as the unit on which to model the interface for the purpose of this study. This is a positive displacement unit, intermediate between a diaphragm displacer and a piston displacer design. Since a prime goal of the overall system design is to minimize the total implanted volume we have considered it legitimate to insert a part of the thermal converter system within the envelope of the pump unit, provided that the pump function and physical appearance of the unit remain identical as seen by the blood. Because of the fragility of the pump membranes, we disallowed the possibility of directly actuating the membrane without an intermediate fluid layer to uniformly distribute the mechanical stresses. A simplified drawing of this type of pump unit is illustrated in Figure 3.1-1.

A volumetric efficiency of 80% is assumed as a reasonable proportion of the specified 60% overall power transfer efficiency of the pump. Therefore, to accommodate the required maximum blood flow rate of 12 liters/minute at the specified maximum pump cycle rate of 120 cpm, the design has been scaled to provide a stroke volume displacement of 125 milliliters. The remaining mechanical/hydraulic efficiency factor of 75% results in a back pressure during pumping of 133.3 mm of mercury (mm Hg) at the input to the blood pump, corresponding to the specified mean functional back pressure of 100 mm Hg at the output of the blood pump (the aorta).

PNEUMATIC DRIVE VERSION SHOWN, SCALED FOR 125 cm^3 STROKE VOLUME

Figure 3.1-1. Kwan-Gett Positive Displacement Blood Pump

Since the pump will probably be installed in the chest and the power unit in the abdomen, it is desirable to provide a coupling in the connecting line which can be made quickly and easily during the installation procedure. It is envisioned that the power unit will be already in operation at this time, so a simple mechanical connection would be convenient in allowing the surgeon to prime the blood pump by hand before it is energized by connecting it to the power unit. We have referred to this requirement as a "dry" coupling indicating a preference for a mechanical linkage. Connecting fluid lines with their inherent installation problems that may involve balancing fluid inventories are not an attractive alternate.

Two important constraints on the PCCS designs have already been discussed as overall system design considerations in Section 2. The first, which is the need to match the power format with the load profile, precludes (as we discussed) the use of intermediate power storage in devices such as pneumatic or mechanical springs which have force-limited characteristics, unless they are combined or integrated with other devices to produce a satisfactory net force characteristic. The second constraint also relates to the blood pump load characteristic and requires that the force rise-time during ejection, and the force fall-time at the beginning of the fill period, be sufficiently rapid so that there is a minimum reduction

in the net length of each of these phases. As we showed in Section 2-3, a noncompressible positive-displacement linkage in the power train reduces this problem. Compressible pneumatic segments call for careful design to ensure that the effective power duty cycle and filling periods are not appreciably degraded.

3.1.2 Blood Pump Filling Requirements

Several approaches to overall control of the blood pump (i. e. , ensuring that the output blood flow rate corresponds to the physiological demand at each moment in time) have been proposed. The method adopted for the purposes of this study, "passive autoregulation," probably represents the simplest and most practical approach. The technique has proved to be satisfactory in recent experiments where animals have been supported for periods of several hundred hours with total heart replacement pumps.

The essential requirements of the "passive autoregulation" approach are (a) to maintain the blood in-flow rate proportional to the source (or in-flow) pressure head in the left atrium, and (b) to ensure that all of the admitted blood is discharged during the following ejection phase. To accomplish this satisfactorily and ensure an overall filling rate commensurate with the required maximum flow rate, the system must provide a positive filling force to offset the hydraulic resistances and inertia inherent in all artificial blood pumps. Since the source pressure is affected by changes in the body ambient pressure in the immediately surrounding body fluids (postural effects, natural respiration, changes in atmospheric pressure, etc.), the bias pressure introduced by this positive filling action must similarly be referenced to (intrathoracic) body ambient, preferably from a site close to the filling source, i. e. , the left atrium/pulmonary vein complex. One particularly important constraint, however, complicates the design of the positive filling action, and that is the need to avoid generating suction (negative pressure) at the input to the blood pump. Negative pressures lead to undesirable physiological complications within the pulmonary circulation and also to the possibility of the veins collapsing and damaging the pump in-flow valve.

There are two basically different ways of operating the blood pump: either by maintaining a fixed pump cycle rate and allowing the stroke volume to vary as a result of different in-flow rates (net flow rate will then reflect the metabolic demand); or allow the pump to fill completely each cycle, then initiate the discharge phase by using an end-of-fill detector. In this case the pump frequency reflects the metabolic demand and determines the output flow rate. For the nominal daily power profile specified for the study, both approaches are satisfactory. The variable frequency mode would be more complicated to implement if a greater than 3:1 flow ratio were required. This is the result of the pump operation being limited within a frequency range which is constrained at the high end by the filling performance of the blood pump, and at the low end by the medical requirement to maintain a minimum pulse rate in the range of 45 to 50 beats per minute.

Some designs are more easily implemented at variable frequency and some at fixed frequency. For example, synchronous reciprocating engines have a fixed stroke length and so are easier to implement on a variable frequency basis. Devices such as solenoids and piezoelectric drivers can be readily designed to produce fixed pulses of power that also fit more readily into the variable frequency regime.

3.1.3 Actuation of the Blood Pump

The evolution of the actuator design concepts developed for the two pump operating modes and each of the specific power conversion approaches is illustrated in Figure 3.1-2. The goal was to develop practical designs for units which efficiently and automatically perform as many as possible of the functional requirements just described. A second goal was to minimize the complexity of the interface with the power conversion stage in either the engine (synchronous systems) or the power conditioning unit (nonsynchronous systems).

Two families of automatic actuators were developed. The Model 1 versions can be used in either the fixed or the variable frequency systems and the Model 2 versions only where the operating frequency is fixed.

The simplest approach is to pump hydraulic fluid between the abdominal power unit and the blood pump. However this requires actuators

NOTE: THE PHYSICAL SIZE OF THE MODIFIED BLOOD PUMP IS PROJECTED TO BE ABOUT 170 cm³ (10 in³). THIS FIGURE HAS BEEN SUBTRACTED FROM THE ESTIMATED VOLUMES OF THE CONFIGURATIONS SHOWN HERE TO ARRIVE AT THE VALUES INDICATED.

KWAN-GETT BLOOD PUMP
 80% VOLUMETRIC EFFICIENCY
 (60% OVERALL)
 MAXIMUM FREQUENCY 120 BPM
 STROKE VOLUME 125 cm³

MODEL 1
 (ANTI-SUCTION VALVE DESIGNS)

580 cm³
 0.798 Kgms

MODEL 2
 (MAKE-UP RESERVOIR DESIGNS)

650 cm³
 0.871 Kgms

MODEL 1M
 MECHANICAL
 ACTUATOR

267 cm³
 0.367 Kgms

MODEL 1F
 FLUIDIC
 ACTUATOR

277 cm³
 0.381 Kgms

MODEL 1B
 CLOSED
 HYDRAULIC
 DRIVE

MODEL 2M
 MECHANICAL
 ACTUATOR

103 cm³
 0.159 Kgms

MODEL 2F
 FLUIDIC
 ACTUATOR

236 cm³
 0.336 Kgms

MODEL 2B
 CLOSED
 HYDRAULIC
 DRIVE

Figure 3.1-2. Blood Pump/Actuator Units

which are large and heavy with respect to the total system budgets. Bellows must be used as displacement amplifiers when the engine cannot provide the stroke length required by the output power piston. Certain special types of bellows are feasible for 10-year lifetimes, but they must be operated with very low displacement per convolution and therefore are relatively bulky. Generally a more compact actuator can be designed using mechanical devices (cables and push rods) to transfer the power.

Careful management of the fluid behind the output power piston is required to avoid considerable power losses, and all of the designs make use of flexible (compliance) reservoirs located at the outer surface of the unit which expand displacing body fluids. All of the designs use gas, rather than liquid, in this space in order to minimize friction and inertial losses. There are several acceptable gas/membrane combinations that appear capable of providing satisfactory performance over a 10-year period. The degradation of performance, with the maximum specified changes in ambient pressure, falls within the acceptable limits.

All of the Model 1 automatic actuator designs use a special valve operated by body fluid pressure to achieve both antisuction protection and automatic output flow rate control. The Model 2 designs use a reservoir of fluid maintained at body ambient pressure to achieve the same goals. The second design approach is potentially more compact than the first since the power piston can share some of the volume within the blood pump envelope.

3.1.3.1 Model 1 Automatic Actuators

The working principle of the Model 1 automatic actuators is illustrated in Figure 3.1-3. A collapsible "segment" is interposed between the fluid volume in the blood pump and the output power piston which provides the back and forth volumetric displacement. The outer wall of the pressure-sensitive segment is exposed to the local body fluid pressure so that it tends to collapse, thus stopping flow and isolating the blood pump from the power piston, whenever the fluid pressure within the "segment" falls below body ambient. Since blood will flow at a more or less steady rate during filling, the valving action will tend to be a series of short on-off cycles, resulting in a "flutter" mode of operation with the power piston being displaced under a steady applied force at a mean rate matching the blood in-flow rate.

- ANTIVAC VALVE PROVIDES PROTECTION AGAINST SUCTION DURING POSITIVE ACTION FILLING
- DISPLACEMENT OF POWER PISTON IS ADEQUATE MEASURE OF FILLING RATE
- AUTOMATIC CONTROL ACTION IF ENGINE OPERATED AT END OF STROKE (VARIABLE FREQUENCY) OR IF ENGINE RUN FIXED FREQUENCY (VARIABLE STROKE)

Figure 3.1-3. Principle of Model 1 Automatic Actuators

The antisuction valve concept, therefore, requires a positive filling action such as a spring-loaded return stroke on the power piston, but it prevents the generation of a sustained negative pressure behind the blood pump membrane. At the same time, this valve action couples the displacement rate of the power piston to the blood filling-rate-controlled displacement of the blood pump membrane.

Automatic system operation can be achieved either by using an end-of-stroke sensor operated by the power piston to initiate a series of identical power pulses or by simply allowing the high pressure fluid drive to discharge the part-full blood pump at fixed intervals. The variable length stroke automatically meters the correct amount of fluid from the high pressure accumulator and thus regulates the amount of power delivered per stroke.

Three versions of this actuator design were developed; one for short mechanical strokes, one for long strokes, and one for the high pressure fluid drive. The synchronous vapor cycle engine operates at variable frequency (60-120 cpm) and furnishes a full 1.33-inch mechanical stroke on initiation by a noncontacting pump-full sensor. The Model 1M automatic actuator (Figure 3.1-4) is designed to transfer this power pulse to

Figure 3.1-4. Model 1M Automatic Actuator

the blood pump. Positive filling action results from a pulling force on the mechanical coupling provided by a spring action within the engine subsystem.

The nonsynchronous gas cycle engine operates at a fixed frequency of 120 cpm which is preset by adjustment of a hydraulically powered automatic timer/valve which will be described later. The antisuction valve action ensures that the pump fills during the (approximately) 0.25-second filling phase, to a level determined by the in-flow blood pressure. A positive filling force is provided by permanently connecting the upper face of the reciprocator piston (Figure 3.1-5) which has a smaller effective area than the lower face, to the high pressure side of the accumulator. This force acts continuously, requiring an additional force to neutralize this bias during the ejection phase. This mechanism is long because of the need to support the rolling seal, but it is more efficient than using an equivalent return spring. In this design the option of a "dry" connection was sacrificed to reduce size and weight.

Figure 3.1-5. Model 1F Automatic Actuator

A system using a synchronous gas cycle engine must be operated in a variable frequency mode because it is impracticable to operate with a variable stroke length or stroke power. The variable speed engine design requires a control signal for a heat flow regulator consisting of a pair of gas-displacement-controlled heat pipes integral with the engine/heat source interface. Obtaining an appropriate control signal requires a complex actuator design using a combination of a push rod with a spring-controlled separation section to allow measurement of the mismatch between the engine frequency and the power demand, and a bellows displacement amplifier to match the short stroke available from the engine. The design of this actuator is not shown because this system design was eliminated before the final evaluation.

3.1.3.2 Model 2 Automatic Actuators

A novel feature of the second group of automatic actuator designs is the addition of a fluid reservoir which allows the power piston to go through a displacement cycle which is not locked to the blood in-flow rate and the resulting displacement of the pumping membrane. This decoupling

between the power piston and pumping membrane results in a unit which varies slightly in physical size according to the flow rate as well as the usual cyclic change. A membrane within a cage-like container surrounding the main body of the blood pump/actuator is used to form a make-up reservoir and it is this membrane which undergoes additional small physical pulsations according to the flow rate. In this physical configuration the fluid within the reservoir is maintained at a pressure close to body ambient.

The power piston in this design of actuator (Figures 3.1-6 and 3.1-7) is cycled through a fixed volumetric displacement at a constant frequency (120 cpm). At the beginning of the filling phase, the pumping membrane and the reservoir membrane are at their maximum displacement positions. The downward displacement rate of the power piston is fixed and equal to the in-flow rate corresponding to the maximum blood pumping rate (12 l/min). If the physiological demand is such that blood flows into the pump, under the controlled pressure gradient (left-atrium pressure to body ambient) at a slower rate, the pressure in the blood pump will begin to fall. However, as soon as the pressure within the blood pump begins to fall below body ambient, the membrane enclosing the make-up reservoir begins to collapse maintaining the pressure very close to ambient. At the end of the filling cycle, the make-up reservoir has been depleted by an amount equal to the difference between the maximum stroke volume (125 cm^3) and the actual stroke volume, corresponding to the physiological demand at that particular time.

When the power piston reverses direction and begins the ejection phase, the initial flow is back into the make-up reservoir, until the reservoir is refilled at which time the pressure within the pump rapidly rises to the net pump back-pressure level and blood ejection begins. At the end of the piston stroke the pumping membrane is returned exactly to its initial maximum displacement position and all of the admitted blood is discharged.

The make-up reservoir design thus automatically delivers a partial stroke according to the physiological demand, provides positive filling action with adequate antisuction control, and presents a very simple mechanical requirement to the actuator drive unit (i. e. , fixed frequency, fixed stroke reciprocating motion).

SUITABLE FOR ELECTRICAL NON-SYNCHRONOUS SYSTEMS AND
NON-MODULATED VERSIONS OF ALL CANDIDATE SYSTEMS

MAKE-UP RESERVOIR PROVIDES PROTECTION AGAINST SUCTION DURING POSITIVE ACTION FILLING
AT FIXED FREQUENCY THIS DESIGN ALLOWS A FIXED DISPLACEMENT AND PRESENTS A LOAD WITH A VARIABLE DUTY CYCLE TO THE UNIT DRIVING THE POWER PISTON

Figure 3.1-6. Principle of Model 2 Automatic Actuators

Figure 3.1-7. Complete Pump/Actuator Cycle

The power required to discharge each stroke corresponds to the output blood flow rate and the mean back pressure — provided that the power piston displacement rate is maintained constant when the make-up reservoir, which presents a negligible load, is being refilled. If the piston speeds up appreciably when it is not loaded, additional power will be dissipated kinetically within the actuator.

The initial version of this actuator design, the Model 2M, Figure 3.1-8, was designed for the nonsynchronous (electrical) candidate systems. The power piston is specially designed so that it fits very close behind the pumping membrane, with a minimum protective layer of fluid to even out the pumping stresses and prevent direct physical contact during pumping. The effective diameter of the piston is about 4 inches and it requires a 1.33-inch stroke to displace 125 cm³ of fluid. A double membrane design is used for the piston. At the top and bottom positions the inner membrane is pulled almost flat while the outer membrane is deflected into a convex or bulged shape by a volume of trapped liquid. Because of the particular geometry of the design, the effective width of

Figure 3.1-8. Model 2M Automatic Actuator

the piston changes during the displacement cycle. The liquid trapped between the two membrane layers, however, smoothly bows the membranes into a "fatter" section at the mid-position where the effective diameter is at a minimum, thus precluding any tendency to fold or wrinkle which would reduce the durability of the unit.

For comparable efficiency, mechanical power transfer results in the most compact actuator designs. A plastic-coated stainless steel cable is used to transfer back-and-forth motion with a 1.33-inch stroke from the electrically driven reciprocator. It is not necessary to transmit a push-then-pull force through this cable because a force bias is provided by a spring to maintain the cable in tension during the entire cycle.

The swept volume behind the power piston is filled with a gas such as carbon dioxide which is partly compressed and partly pumped into the volume between the power transmission cable and the outer membrane. Final optimization of this particular design feature requires more detailed study.

A push-pull version of the Model 2M actuator (Figure 3.1-9) is used with the nonmodulated linear vapor system. Two more versions, Model 2B and 2F were designed for a synchronous gas reciprocating system and a nonsynchronous gas reciprocating system, respectively. The Model 2F is shown in Figure 3.1-10. The synchronous gas reciprocating candidate was not considered in the final evaluation.

The essential characteristics of the six actuator designs are summarized in Table 3.1-1.

3.1.4 Power Conditioning

The synchronous engine candidates interface directly through a mechanical connection with the appropriate blood pump actuator. No intermediate power conditioning or conversion is necessary. Both the electrical and hydraulic nonsynchronous candidates require further processing. The nonsynchronous gas cycle engine (gas reciprocator/ TESM system) interfaces with a hydraulic power storage unit which also functions as a sensor for the engine power output modulator. In the case of the candidates with electrical outputs, the electrical power from the engine subsystem must be converted into the reciprocating mechanical

Figure 3.1-9. Push-Pull Version of Model 2M Automatic Actuator

Figure 3.1-10. Model 2F Automatic Actuator

Table 3.1-1. Actuator Design Characteristics

Model	Candidate System	Size (cm ³)	Wt (kg)	Effi- ciency (%)	Comments
1M	Linear Vapor/TESM	267	0.367	76	Complex, control loop losses
1F	Gas Reciprocating/TESM	277	0.381	76	
1B*	Synchronous Gas Reciprocating/TESM	325	0.494	59	
2M	All Modulated and Non- modulated Electrical Candidates and Linear Vapor	103	0.159	76	
2F	Gas Reciprocating	236	0.336	76	Spring rate losses
2B*	Nonmodulated, Synchronous Gas Reciprocating Cycle	226	0.322	69	

*These candidate systems were eliminated before the final evaluation.

motion required to operate the blood pump actuator. The energy storage unit (battery) is defined as part of the engine subsystem.

3.1.4.1 Hydraulic Power Conditioning

The design of the hydraulic power conditioning unit is shown in Figure 3.1-11. It consists of a combined high pressure/low pressure accumulator which accepts the 750 cpm pumping output from the engine subsystem and a preset hydraulic timer/rotary valve. The essential features of the design are (a) the large size of the pneumatic spring required to provide the necessary force bias, and (b) the compact design of the hydraulic timer/switch unit.

The accumulator presents a design challenge even assuming relatively optimistic response times from the engine. Storage for 7.5 cm³ of fluid at the intermediate fluid pressure (180 psig), which represents power for only 2 to 4 strokes of the blood pump, requires a unit with an outer diameter of almost 3 inches. The force bias required for this unit is beyond the capability of even the best mechanical springs and a bellows

Figure 3.1-11. Hydraulic Power Conditioner

pneumatic spring is proposed. However, a very conservatively rated design is required since with an engine speed of 750 cpm the output ripple over 10 years will cycle the device through small excursions about 4×10^9 times.

For the gas reciprocating system, the accumulator must be provided with a check-valve controlled dissipative bypass. For the modulated, gas reciprocating/TESM system a mechanical output from the bellows is required to operate the engine modulation unit (speed control). These design details are not shown in the figure.

A gear-vane motor running from the high pressure supply cycles the rotary valve, through a gear reduction, at 120 cpm. If silicone fluid is used rather than water as the intermediate high pressure hydraulic fluid, then relatively high efficiency and 10-year life can be projected for this design.

The basic dimensions of the combined unit are 5.9 cms by 7.9 cms in diameter. This basic volume (234 cm^3) is increased to 261 cm^3 when the dissipative bypass for the gas reciprocating system is included and to 285 cm^3 when the volume required for the additional modulator for the gas reciprocating/TESM system is added. The weights corresponding to these modifications are 0.408, 0.463, and 0.508 kg respectively. The net power transfer efficiency for all versions is projected to be 83%.

3.1.4.2 Electric Motor/Reciprocator Unit

There are several alternate ways in which the electrical power output from the engine subsystems can be converted to the reciprocating mechanical motion required by the automatic actuator. Among the design approaches that have been explored specifically for electrically driven blood pumps are piezoelectric conversion, high-efficiency solenoids, variable reluctance oscillators and various motor-driven devices. For several reasons which have been discussed in Section 2.3, we have selected a design approach using a constant-speed motor driving a drum-cam reciprocator, which when combined with the novel (Model 2M) design of automatic actuator offers a number of significant advantages. In summary these advantages are:

- Short-Term Load Matching. The constant speed operation allows the mechanical inertia of rotating parts to be used to provide the total system with an output characteristic which is not force-limited. Even if the back pressure during ejection varies, the actuator will not stall and it will draw power from the system at a level corresponding to the mean rather than the peak back pressure. Also it is possible to achieve higher overall conversion efficiencies with a constant speed of operation rather than with a control mode requiring start-stop or variable-speed operation.
- Simple, Automatic Operation. No specific control sensors or power-modulating devices are required with this particular combination of components. The motor/reciprocator runs at a preset constant speed (and mechanical stroke) even when the delivered output power is varied over a 2:1 range. The load variation is passed to the engine subsystem as simply a variation in the electrical load, i. e., a varying demand for electric current.

- Uses More Compact Automatic Actuator. The constant frequency operating mode which is possible with the Model 2 automatic actuators allows elimination of the antivacuum valve configuration and therefore more compact actuator designs with the power piston within the nominal envelope of the blood pump. Electrical converters with fixed pulse power formats such as the solenoid or the piezoelectric devices require a variable-frequency mode of operation and use of the less compact Model 1 actuators. At the system level, the difference between the two actuators is 164 cm³ and 0.209 kg.
- Flexibility. Since the preset motor speed determines the output pulse rate, it also predetermines the maximum (and minimum) blood flow rates. The operating range of output flow rates can then be varied within a small range (up to the frequency limit set by the filling characteristics of the blood pump) very simply prior to installation by adjusting the preset motor speed. When blood pumps with better filling characteristics are developed, they can be incorporated without significant system redesign. A higher frequency of operation would achieve the same flow rate performance with a physically smaller pump unit, or increased flow performance from the same size of pump.

Another refinement that can be incorporated is to design the groove in the cam-drum (discussed later) to provide a blood pressure waveform which is a close match to the natural physiological wave form.

- State-of-the-Art Technology. The chosen design approach is based on existing and proven techniques. There is no question that the device can be developed within the projected size and weight.

The preferred mechanical design of the motor/reciprocator unit, with physical dimensions, is shown in Figure 3.1-12. The brushless dc electric motor is designed to run at a constant speed of about 9,000 rpm. It runs in an hermetically sealed housing on self-energizing compliant multi-foil type gas bearings. Rotary motion is coupled through the aluminum end face by means of a double plate magnetic coupling about 1 inch in diameter. The motor bearing surfaces are offset to compensate for the axial thrust resulting from this coupling. As noted in the section on turbogenerator design (Section 3.2.3), the bearing design is considered state-of-the-art. The bearing is designed to become aerodynamically supported at about 1,000 rpm and at full operating speed there is no physical contact whatsoever between the surfaces. Since the motor runs continuously at a

Figure 3.1-12. Motor/Reciprocator Unit

constant speed (within 2 or 3%), a minimum wear situation will exist. Exposing the motor to mechanical shocks greater than about 3g will cause a momentary "brushing" between the rotating Teflon-coated foils and the stationary surface before the rotor restores itself to the stable no-contact position. However this bearing design is commonly used for applications involving many start-stop cycles and it is felt that the motor will easily withstand a 10-year mechanical shock environment such as will be encountered in the body.

To achieve high conversion efficiency, the motor uses samarium-cobalt permanent magnets on the rotor and an ironless stator. It is electronically commutated by switching signals originated by a position encoder built into the motor. Calculations indicate that a motor efficiency well in excess of 70% can be achieved. This figure includes the commutation losses and the magnetic coupling

Under the projected operational load conditions, the stored momentum of the rotor will ensure virtually constant speed operation.

In the worst-case situation, when the actuator presents a no-backpressure situation for the first part of the ejection phase, the change in speed is calculated to be 2.3%.

The 75:1 rotary speed reduction from 9000 to 120 rpm is achieved using a very compact compound planetary gear train (Figure 3.1-14). The appropriate mechanical stress levels for each of the components are calculated to be well within the allowable design loads for regular high-quality gear and bearing materials. The moving parts are splash-lubricated with lightweight oil. The power transfer efficiency including projected bearing and windage losses is 85%.

Figure 3.1-13. Motor/Reciprocator at (a) Start-of-Stroke
(b) Mid-Stroke and (c) End-of-Stroke

The cam-drum runs in a set of combined radial/thrust film-lubricated porous bronze bearings. The 120 rpm rotary motion is converted to a reciprocating mechanical displacer with a fixed amplitude of 1.33 inches by means of a continuous guide or slot machined in

Figure 3.1-14. Electrical PCCS Gear Box Details

the inner surface. A shuttle (Figure 3.1-13) which is free to move back and forth along a set of stationary guides (which are designed to absorb the radial forces) is provided with a hardened guide pin which engages in the machined slot on the rotating drum. The continuous elliptical configuration of the machined slot is such that the shuttle is moved smoothly through one back-and-forward reciprocating cycle for each revolution of the drum. A specially designed long-life bellows is used to seal the lubricant within the reciprocator unit. This bellows is attached to the shuttle unit and is located concentric with the long axis of the unit within the stationary guides. A mechanical output is obtained by connecting the cable of the Model 2M actuator to a mechanical connecting post attached to the inside surface of the bellows.

The overall length of the unit is 3.5 inches and it is 2 inches in diameter and 10.9 in^3 (179 cm^3) in volume. The calculated weight is

INPUT = 9000 RPM
POWER = .013 H.P.
INPUT TORQUE = .091 IN-LBS

$$\text{GEAR RATIO } \frac{n_d}{n_e} = \frac{1 - \frac{ac}{bd}}{1} = \frac{1}{75}$$

$$\text{WHERE } \frac{ac}{bd} = 1 - \frac{1}{75} = \frac{74}{75}$$

LET B + C GEARS = 20 TEETH ON 64 D.P.

A - 74 TEETH ON 64 D.P.

D - 75 TEETH ON 64 D.P.

OUTPUT TORQUE = 75 (.091) = 6.8 IN-LBS

MAX. ALLOWABLE TOOTH LOADING = 1000 LBS/ 1" TOOTH WIDTH

$$\text{INPUT TOOTH LOADING } \frac{\text{TORQUE} \times L}{\text{TOOTH WIDTH}} = \frac{\frac{.091}{1.17} \times 2}{.03} = \frac{.155}{.03} = \frac{5.2 \#}{1" \text{ TOOTH WIDTH}}$$

$$\text{OUTPUT TOOTH LOADING} = \frac{\frac{6.8}{1.17} \times 2}{.08} = \frac{11.6}{.08} = \frac{145 \#}{1" \text{ TOOTH WIDTH}}$$

0.88 lb (0.4 kg). The overall power transfer efficiency from electrical input to mechanical output is 59%. The total surface area of the unit is about 180 cm^2 and, therefore, the mean flux through the surface is less than 0.04 watt/cm^2 .

3.1.4.3 Electrical Control for the Reciprocating Vapor Candidate Systems

The liner vapor/TESM system and the liner vapor system are both controlled electrically. The solenoid pump and the small thermoelectric module are considered to be part of the engine subsystems. The engine rate control for the liner vapor/TESM system is a non-contacting type of electrical switch which physically senses when the mechanical actuator drive reaches the pump-full and the end-of-stroke positions. The weight of this switch together with the necessary electronic circuitry is 0.041 kg. This figure must be added to the basic weight of the 1M model actuator (0.367 kg) to obtain the total PCCS weight (0.408 kg). These additional small parts can be easily accommodated within the existing package volumes.

For the liner vapor system, which runs at a fixed stroke rate of 120 cpm, the electrical stroke initiation and termination signals are generated by a preset fixed-rate electronic oscillator. The additional weight is estimated to be 0.035 kg, and it is again easily accommodated within the engine subsystem package envelope. The total PCCS weight in this case is the weight of the Model 2M actuator (0.160 kg) plus the additional electronic oscillator module (0.035 kg) which is equal to 0.195 kg.

3.1.4.4 Alternate Mechanical Configurations

The physical character of the arrangements used to illustrate the mechanical interface between the different actuators and the modified Kwan-Gett blood pump must be considered only as nominal designs at this time. The mechanical configurations will be optimized when a more detailed specification is available.

Two alternate general configurations have been considered for the motor/reciprocator unit (the preferred design approach). One is the transmission of mechanical power via a rotating rather than a push-pull linkage. The second is the impact of changing the direction of the mechanical drive with respect to the direction of displacement of the blood

pump membrane.

In order to use a rotary cable connection, the rotary-to-linear motion converter (the reciprocator part of the unit) must be physically located on the actuator. A specific detailed mechanical design with the drum-cam physically located within the actuator was not attempted since it is clear that this alternate configuration would have a minor impact on the overall size, weight, and performance of the unit. The lubrication fluids would be retained in the gear box in this case by a screw type seal rather than the bellows used in the nominal design. This would permit a slightly more compact design but this option was not preferred because it will increase the size and weight of the intrathoracic portion of the converter system. However, with this arrangement, the coupling cable is no longer necessarily parallel to the pump membrane displacement direction. According to the data supplied by one cable manufacturer, the power transmission efficiency is constant at about 95% for total bend angles up to 90° .

An alternate type of reciprocator unit uses an epicyclic gear principle rather than the drum-cam approach. In this case, the shape of the modified intrathoracic package is less symmetrical, but the cable drive runs at right angles to the direction of the pump membrane displacement. Since, again the cable can be flexed up to 90° without significant wear or efficiency penalty, these two basic configurations permit complete freedom in selecting the cable connection angle. One drawback with the epicyclic reciprocator is the need to lubricate the gears when it is mounted on the actuator, whereas the drum-cam unit can be run with dry lubricant. The need for a seal such as a bellows at the reciprocating mechanical output makes this option less attractive than the drum-cam.

3.1.5 Performance Summary

The major design features of the power conditioning and control subsystems for each of the candidates that were finally evaluated are briefly summarized below.

3.1.5.1 Linear Vapor/TESM System

Engine Output:	Reciprocating mechanical drive, 1.33-inch stroke at variable frequency (60-120 cpm). Power delivered on push stroke. Preload spring in engine required to provide force for filling action.
Modulation:	By variation in stroke frequency. Constant stroke work.
Engine Control:	Electrical control system requires signal at end of filling stroke to initiate the power stroke. Non-contact switch is installed within engine subsystem package.
Actuator:	Model 1M. Antisuction valve version with flexible mechanical push rod in plastic-coated guide.
Packaging:	Single package, 267 cm ³ , 0.408 kgm
Power Transfer Efficiency:	76%.

3.1.5.2 Linear Vapor System

Engine Output:	Reciprocating mechanical drive, 1.33-inch stroke at constant frequency (120 cpm). Power delivered on push stroke. Preloaded spring in engine required to provide filling force.
Modulation and Engine Control:	None, preset electrical timer. Excess stroke power dissipated in actuator/blood pump.
Actuator:	Model 2M. Make-up-reservoir version with flexible mechanical push rod in plastic-coated guide. No spring.
Packaging:	Single package, 103 cm ³ , 0.195 kg.
Power Transfer Efficiency:	76%.

3.1.5.3 Gas Reciprocating/TESM System

Engine Output:	Pumped high pressure (180 psi) hydraulic fluid delivered at a variable flow rate determined by the engine speed (nominally 750 cpm).
Modulation:	By variation of engine speed.

Engine Control:	Mechanical signal from accumulator varying engine dead space.
Actuator:	Model 1F. Antisuction valve version with piston reciprocator located behind the output power piston. Power transmission through a pair of high-pressure lines. Size 277 cm ³ , weight 0.381 kg efficiency 76%.
Power Conditioning:	Combined high pressure/low pressure accumulator with fluid lines and rotary valve located within the engine subsystem package. Size 285 cm ³ , weight 0.508 kg, efficiency 83%.
Packaging:	Actuator permanently connected to power-conditioning unit which is located within the engine subsystem package. Size 569 cm ³ , weight 0.889 kg.
Power Transfer Efficiency:	63%.

3.1.5.4 Gas Reciprocating System

Engine Output:	Pumped high pressure (180 psi) hydraulic fluid delivered at fixed (maximum) flow rate.
Modulation and Engine Control:	None. Engine speed preset at 750 cpm.
Actuator:	Model 2F. Make-up-reservoir version with piston reciprocator located behind the output power piston. Power transmission through a pair of high pressure lines. Size 236 cm ³ , weight 0.335 kg, efficiency 76%.
Power Conditioning:	Same as modulated version except that the modulator output is replaced with a check-valve controlled dissipative bypass. Size 261 cm ³ , weight 0.463 kg, efficiency 83%.
Packaging:	Same as modulated version. Size 497 cm ³ , weight 0.798 kg.
Power Transfer Efficiency:	63%.

3.1.5.5 Modulated and Nonmodulated Electrical Systems

Engine Subsystem Output:	DC electrical power at 15 volts.
Modulation:	Passive; by load variation.
Engine Control:	None. Preset to generate 6.2 watts (9.9 watts for system without battery).
Actuator:	Model 2M. Make-up-reservoir version with spring biased cable transmission, mechanically connected to the mechanical output of the motor/reciprocator unit. Size: 103 cm ³ , weight 0.160 kg, efficiency 76%.
Power Conditioning:	Electrical to reciprocating-mechanical output with 1.33-inch stroke at 120 cpm. Constant speed dc permanent magnet gas bearing motor driving a drum-cam reciprocator through compound planetary gear speed reducer. Nominally packaged separate from the main power unit (engine subsystem). The batteryless system has additional electronic circuitry to control the load sharing between the two generators. Size: 180 cm ³ , weight 0.4 kg, efficiency 59%.
Packaging:	Nominally separate packages for the actuator and motor/reciprocator unit with mechanical connection. Armored cable to carry electrical power between the main power unit and the motor/reciprocator. Except in the case of the thermoelectric/battery system design, where the motor/reciprocator is physically integrated with the static generator in the main power unit. Size 283 cm ³ , weight 0.560 kg.
Power Transfer Efficiency:	45%.

3.2 ENGINE SUBSYSTEMS

The thermal converter engine subsystems convert heat from the radioisotope into hydraulic, pneumatic, electrical or mechanical energy for the power conditioning and control subsystem or for direct activation of the blood pump. The most promising classes of engine subsystems selected for use in the thermal converter are:

- Gas reciprocating
- Linear vapor
- Rotary vapor
- Thermoelectrics
- Hybrid (consisting of thermoelectrics and rotary vapor devices)

Thermoelectric elements are also used with the reciprocating devices on several of the candidate systems for electrical control power. As used in this study, the term hybrid refers to subsystems in which the function of primary power generation is shared between two different types of engines.

The engine subsystems examined were drawn from three basic sources:

- Engines being developed by others specifically for artificial heart devices
- Engines that have been developed for other applications and whose characteristics make the designs attractive for this application
- Engines utilizing new approaches, but based on proven technology

Each type of engine examined in the gas reciprocating, linear vapor, rotary vapor, thermoelectric and hybrid class are discussed in Sections 3.2.1 to 3.2.5, respectively.

Although many engine designs were evaluated and are described in detail, most were not considered legitimate candidates for scoring because they were deficient in one or more of the following respects:

- 10-year-life design There must be a reasonable level of confidence that the engine design being considered will be capable of operating for a 10-year period.
- Proven technology design Information must be available to show that the design is feasible and that the requisite technology is currently available.
- Design must meet AEC groundrules
 - 60-watt heat source maximum
 - Volume less than 1.5 liters
 - Weight less than 3 kg
 - Specific gravity less than 2.0
- Design must meet TRW groundrules
 - Fibrous insulation (Min-K with xenon fill gas) throughout rather than vacuum foil insulation (see Section 2.7).
 - Heat rejection to body tissues with container designed to limit heat flux to 0.07 watt/cm² or less (see Section 2.5).
 - Energy storage must be from either TESM or high-energy-density batteries (see Section 2.8).
 - The engine output must be compatible with a power conditioning and control subsystem as presented in Section 3.1.

3.2.1 Gas Reciprocating Engines

3.2.1.1 Historical Background

The basic principles of regenerative reciprocating engines date back many years. Until the last few years, however, progress has been slow and has been confined to sizes and power levels outside the practical range for artificial heart applications. Milestones marking engine innovations are listed in Table 3.2.1-1.

The first displacer-type engine, built by Stirling in 1817, has been revived in modern form in sizes of about 50 hp per cylinder. Stirling also built a double-acting machine with a displacer. A regenerative gas machine using pistons with no displacers was invented in 1870. Because the pistons were single-acting, the specific power was low.

The first liquid/vapor engine was built in 1930. This engine introduced the liquid regenerator, recently termed a "tidal" regenerator and descriptive of its changing liquid level. A pressure generator, with output in the form of pressurized gas to pump up a tank, was built in 1937. The first multi-cycle, multi-cylinder machine came in 1945. Instead of conventional mechanical linkages, it had a swash plate and connecting links for reciprocating motion.

Table 3.2.1-1. Regenerative Reciprocating Engine Development Milestones

Single-Cylinder Displacer	1817
Double-Acting Pistons with Displacer	1840
Single-Acting Pistons without Displacer	1870
Liquid Regenerator	1930
Gas Pressure Generator	1937
Multi-Cylinder, Double-Acting Machine	1945
Resonant Free-Piston Multi-Cycle Variable Volume	1964
Resonant Free-Piston Constant Volume	1966
Resonant Free-Piston Single-Cycle Variable Volume	1967
Electrically-Controlled Free-Piston	1969

Up to that time, engine mechanical output had always been delivered by a crankshaft. The first free-piston engine, proposed and constructed in 1964, was a multi-cycle variable-volume machine. In 1966, a free-piston constant-volume pressure generator was built. This was proposed for artificial heart applications and has received further engineering development since then.

The resonant free-piston single-cycle machine with variable volume was first mechanized in 1967. Finally, an electrically-controlled free-piston engine was built in 1969. This was a single-cycle engine with a liquid regenerator.

3.2.1.2 Cycle and Working Fluid Selection

Regeneration

Regeneration is an essential feature of gas reciprocating machines. It is accomplished by a duct connecting high- and low-temperature working spaces. Typically, the duct is filled with a matrix of fine wire to provide large heat transfer surface area. As gas flows through the regenerator from hot to cold volumes, energy is deposited in the matrix. This energy is returned to the gas when the flow reverses. In small engines, the matrix may be omitted if the narrow passage walls are designed to fulfill the thermal regeneration function.

Carnot efficiency can be achieved only if all heat supply takes place at the peak cycle temperature, and all heat rejection at the lowest temperature. If external heat exchange occurs at intermediate temperatures, engine efficiency is degraded. It is the function of the regenerator to eliminate or minimize heat exchange at intermediate temperatures.

Engine Volume Variation

A simple way to accomplish regeneration in a reciprocating gas engine is to cycle a regenerative displacer back and forth in a volume of gas heated at one end and cooled at the other. The reciprocation alone will cause a fluctuation in pressure, with a pressure ratio of approximately 1.2 per stage. The pressurized gas can be used to charge an accumulator or to actuate a pumping mechanism directly. This operating mode will be referred to as "constant volume".

A second operating mode combines reciprocating flow through a regenerator with volumetric changes in the main volume of gas. Compression and expansion are phased to supplement pressure variations due to reciprocation of the regenerator alone, achieving a much higher pressure ratio (typically 2.2). This mode of operation will be called "variable volume".

Both "constant volume" and "variable volume" refer not to the individual component working spaces, which must always be varying in all mechanizations of Stirling engines, but rather to the total volume. Varying the total volume of the working fluid automatically subjects it to a periodic rise and fall in pressure.

The variable volume operating mode was selected for this program because, as shown in Table 3.2.1-2, its work output is an order of magnitude higher than for a constant volume device of the same physical dimensions and cycle conditions. Expressions for power output used to calculate specific power are as follows:

$$\frac{\text{Constant}}{\text{Volume}} P_{\text{out}} = \frac{\gamma}{\gamma - 1} (X_3 - X_2) (1 - \tau) P_{\text{max}} V \left[1 - \left(\frac{P_{\text{min}}}{P_{\text{max}}} \right)^{\frac{\gamma}{\gamma - 1}} \right]$$

$$\frac{\text{Variable}}{\text{Volume}} P_{\text{out}} = \frac{\pi K \sin \alpha}{\tau + K + \left(\frac{4K_r \tau}{1 + \tau} \right)} (1 - \tau) P_{\text{max}} V \frac{\sqrt{1 - \rho}}{\sqrt{1 + \rho} (1 + \sqrt{1 - \rho^2})}$$

where

$$\rho = \frac{\sqrt{\tau^2 + K^2 + 2\tau K \cos \alpha}}{\tau + K + \left(\frac{4K_r \tau}{1 + \tau} \right)}$$

X_3 = piston position when outlet valve closes

X_2 = piston position when outlet valve opens

V = swept volume

γ = ratio of specific heats

See Table 3.2.1-2 for other symbols

Table 3.2.1-2. Specific Power Comparison

Parameter	Units	Constant Volume	Variable Volume
Max. Temp. (T_{hot})	$^{\circ}\text{F}$	1200	1200
Min. Temp. (T_{cold})	$^{\circ}\text{F}$	100	100
Maximum Pressure (P_{max})	psi	500	500
Cylinder Diameter	inches	1.0	1.0
Displacer Extension Diameter	inches	0.5	0.5
Stroke (S)	inches	1.0	1.0
Fractional Stroke $(X_3 - X_2)/S$		0.25	—
Clearance Volume ($K_r V$)	cubic inches	0.5	0.5
Phase Angle (α)	degrees	—	90
Volume Ratio (K)	—	—	1.0
Temperature Ratio (τ)	—	0.337	0.337
Pressure Ratio	—	1.16	2.2
Minimum Pressure	psi	430	227
Power	lb-in/cycle/cu. in.	12	174
Specific Power	joule/cycle/cu. in.	1.36	19.5

Working Fluid Selection

A reciprocating gas engine can be operated not only by using single-phase gases, but also with a two-phase (liquid/vapor) working fluid. The advantage of using a gas is that 100 percent regeneration would be possible if the gas were ideal. This may be seen from a temperature-entropy chart for an ideal gas showing all polytropic lines parallel to each other.

This means that isothermal processes at different temperatures can be connected by any pair of arbitrary polytropic processes with perfect regeneration. All of the heat given up during cooling can then be recovered during heating.

In contrast, the temperature-entropy diagram for a superheated vapor shows differences in specific heats for expansion and compression. A so-called "pinch temperature" exists which determines to what extent regeneration is possible. In the liquid/vapor region the differences are even greater. However, the vapor systems have the advantage that very large pressure differentials can be achieved. While gas regenerative cycles are limited to pressure differences of a few hundred pounds per square inch, vapor cycles under similar conditions may go as high as several thousand pounds per square inch.

Gas Working Fluids. Ideal gases are most desirable for regenerative machines because of symmetry in their temperature-enthalpy diagrams. The same amount of enthalpy is exchanged during transition from one temperature to another for equal pressure changes in an ideal gas. In non-ideal gases, heat reception and rejection in the regenerator are not balanced, so that additional heat must be supplied or rejected externally.

Engine speed is the most pertinent design variable affecting the choice of gaseous working fluid. At high speeds, hydrogen and helium appear to be the best choices. At low (synchronous) speeds, the choice is not as clear-cut. Gas selection factors influenced by speed include viscous losses, gas leakage past the piston, and heat transfer as limited by conduction. Factors insensitive to speed include loss of gas by diffusion, and regenerator heat storage capacity. The impact of these factors is summarized below:

- Diffusion Losses: Hydrogen diffuses appreciably through metals at typical engine temperatures (1200°F) and, therefore, must be eliminated from further consideration for this reason.
- Regenerator Capacity: The amount of heat which must alternately be stored and released in the regenerator is proportional to the working fluid heat capacity. The C_p/R ratio is 2.5 for monatomic gases and 3.5 for diatomic gases. Accordingly, monatomic gases should

require less regeneration. A treatment of this factor has not been found in the literature.

- Heat Transfer: In relatively high-speed machines, performance may be limited by heat transfer to and from the working fluid in the heat exchangers and regenerator. Accordingly, in these machines, high thermal conductivity in the working fluid is desired. A plot of thermal conductivity versus molecular weight (Figure 3.2.1-1) shows low molecular weight to be decidedly advantageous.
- Leakage: As engine speed is reduced, the importance of gas thermal conductance diminishes because there is more time per cycle to accomplish the required heat transfer. There is also more time, however, for gas to leak around pistons, and at low speeds leakage can become the limiting factor. This effect is apparent in Figure 3.2.1-2 in which engine torque is plotted versus speed for three working fluids. (Reference 18) At low speed the torque with nitrogen is higher than with hydrogen or helium.
- Viscous Losses: For synchronous machines, the working fluid gas may be selected without regard to viscous losses, since these losses are negligible at synchronous speed. At higher speeds the gas selected will not profoundly affect the viscous losses since the viscosities of candidate gases do not differ markedly (Figure 3.2.1-3). Only hydrogen, which is not a candidate, has an exceptionally low value.

Vapor Working Fluids. Regenerative reciprocating machines using vapor working fluids are designed with small strokes, because vapor volume greatly exceeds liquid volume. This is an advantage since small strokes can be achieved by means of a bellows rather than a piston. It is a disadvantage in that dead volumes are disproportionately large compared with swept volumes, and engine operation is fixed within a fairly narrow range with little flexibility. It has been shown that in regenerative machines with vapor cycles the liquid interface in the regenerator can move through a long distance when the piston is displaced through a small distance. The location of this interface is critical, since the engine might stall completely if the interface drifts very far toward the hot or cold side.

Figure 3.2.1-1. Thermal Conductivity of Gases

Figure 3.2.1-2. Measured Engine Torque Versus Speed for Three Working Fluids (Reference 18)

Figure 3.2.1-3. Viscosity of Gases

Thermodynamic heat balances are illustrated in a temperature-enthalpy diagram (Figure 3.2.1-4) for a hypothetical liquid/vapor regenerative engine. A thermodynamic cycle is performed in the sequence ABCDEFGH with water/steam operating between 212°F and 1000°F. Two constant-volume lines at specific volumes of 5 cubic feet per pound and 0.015 cubic foot per pound are connected to the isotherms. At point H, heat is added at constant volume until evaporation is complete, with pressure rising from 14.7 to 2,300 psi. Additional heat is added up to a pressure of 4,500 psi. The pressure drops to 175 psi in the isothermal process between C and D and to 14.7 psi in the ensuing constant volume process. Finally, an isothermal process restores initial conditions.

Thermodynamic calculations show the enthalpy available during heat rejection at constant volume to be out of balance with the enthalpy needed to bring the working fluid up to the higher temperature. Most of the heat is required during evaporation which occurs only between H and B, while there is inadequate capability to recover this amount of heat during condensation which occurs only between F and G.

It can be shown that the overall efficiency of this cycle is 36 percent, compared with a Carnot efficiency of 54 percent. Thus, a relative efficiency of only 67 percent could be achieved for this cycle even if it could be mechanized.

If the desirable property of vapor (large change in specific volume upon evaporation) is to be exploited, the chief disadvantage (imbalance between heat rejection during expansion and heat reception during compression) must be overcome. It is obvious that, where evaporation and condensation take place, an unsymmetrical process diagram must be found in order to realize a higher theoretical efficiency.

3.3.1.3 Configuration Selection

Configurations for reciprocating regenerative machines are classified and schematically represented in Figure 2.2.1-5. The design variants are grouped according to the number of reciprocating elements.

Figure 3.2.1-4. Temperature-Entropy Diagram for Hypothetical Vapor Cycle

Figure 3.2.1-5. Design Options for Regenerative Machines

The two simplest mechanical arrangements (No's. 1 and 2) use a single free piston. Diameters are equal on both sides of the piston, so that these engines operate on the constant-volume principle at a pressure ratio of about 1.25. The only constant-volume machines are in the first group (one reciprocator).

If diameters are unequal on the two sides of the piston, the machine is a variable-volume device, and energy for actuating the piston can be obtained from the expansion of the working fluid itself. In this case, the space on the larger diameter side functions as an expansion cylinder and the other space as an equivalent feed pump. Pressure ratios above 2 are typical, higher than for the constant volume device, because compression and expansion effects are added to heating and cooling.

In the second group (two reciprocators) there are two basic single-cycle variants: conventional link-operated engines (No. 4) and free-piston engines (No's. 5 and 6). Until recently, all closed-cycle reciprocating gas engines were mechanically phased, with each piston connected to a crankshaft by connecting rods or other links. This was necessary not only to maintain the proper phase difference between pistons in gas engines with more than one reciprocator, but also to smooth out power fluctuations in single-cycle machines.

In single-cycle machines, power output is negative during a portion of the cycle. The engine must be kept running with a flywheel to store and deliver energy. The flywheel may be eliminated or reduced in size if more than one cycle operates on a single crankshaft. Power flow is always positive if more than two cycles are used in the same machine. Two-cycle (No. 7) and four-cycle (No. 8) versions of the two-reciprocator group are feasible, as well as the four-cycle, four-reciprocator engines (No. 9).

Mechanically-Linked Versus Free Piston Configurations

Stirling engines perfected for larger power levels are all based on crank mechanisms connected to flywheels. For a single-cycle machine, positive power is available for little more than half cycle time. To keep the engine running smoothly, it is necessary to store about ten times the average energy developed during one cycle.

The power range of single-cycle Stirling engines with mechanically-linked pistons has been relatively limited. One reason for this is that flywheel energy storage capability decreases with the square of the dimension, unless speed is correspondingly increased. Therefore, required flywheel speeds and/or weights become impractical for small engines. In addition, if a mechanical linkage such as a rhombic drive is employed, then the crankcase must be lubricated and sealed from the working volume of the engine. This becomes a major design problem and a major source of engine unreliability. For these reasons we chose to consider only free-piston configurations of Stirling-type engines. Forcé-displacement investigations for a number of possible arrangements are described in the following paragraphs. The analyses are based on assumed sinusoidal displacements and isothermal operation.

Nomenclature. Conventional nomenclature for examining the various mechanizations of regenerative machines is given in Figure 3.2.1-6. The sketch shows the three basic engine volumes: the heated expansion volume, the regenerator, and the cooled compression volume. In the figures, expansion and compression volumes are connected to imaginary cranks rotating at ω radians/sec with a relative phase displacement of α and with $\phi = \omega t$ the reference line. Equations for instantaneous pressure (p , or, in nondimensional form, ψ) and for power (P) are as follows:

$$P = \frac{2WRT}{V(\tau + K + \nu) [1 + \rho \cos (\phi - \theta)]} \quad (1)$$

$$\psi = \frac{pV}{2WRT} = \frac{1}{(\tau + K + \nu) [1 + \rho \cos (\phi - \theta)]} \quad (2)$$

$$P = \frac{\omega WRT [\sin \omega t + K \sin (\omega t - \alpha)]}{(\tau + K + \nu) [1 + \rho \cos (\omega t - \theta)]} \text{ lb-in/sec} \quad (3)$$

W = weight of gas in engine, lbs

T = temperature of heat rejection, °R

ω = rotational speed, rad/sec

Figure 3.2.1-6. Basic Regenerative Cycle with Nomenclature

t = time, sec

τ = temperature ratio = $T_{\text{cold}}/T_{\text{hot}}, ^\circ\text{R}$

K = volume ratio (compression/expansion)

K_r = clearance ratio = dead volume/expansion volume

$$\rho = \sqrt{\tau^2 + K^2 + 2\tau K \cos \alpha} / (\tau + K + \nu)$$

ϕ = crank angle = ωt

α = phase angle (between expansion and compression volume vectors)

$$\nu = 4K\tau / (1 + \tau)$$

$$\theta = \tan^{-1} [K \sin \alpha / (\tau + K \cos \alpha)]$$

Pressure (ψ) from Equation (2) is plotted in Figure 3.2.1-7 for values of the parameters indicated. Pressure is plotted against volume for the expansion space, compression space, and total engine in Figure 3.2.1-8. These plots are all closed curves for one complete cycle. The enclosed area is a measure of the net work per cycle. The work is positive if the area is circumscribed clockwise, negative if counterclockwise.

It is seen from the pressure-volume plots that, over one cycle, work is done by the expansion space, work is done on the compression space, and the difference is the net engine work. Because the compression space net work per cycle is negative in this instance, the compressor piston must be driven using a flywheel with link mechanism.

Figure 3.2.1-7. Pressure Versus Crank Angle

Single Cycle Machine. The most common single-cycle machine is the displacer design shown schematically in Figure 3.2.1-9. The design on the top is the conventional configuration (Stirling engine), that on the bottom, the modification for free piston operation. The force-displacement diagram for the conventional Stirling engine (Figure 3.2.1-8), showing positive net cycle power, applies for the piston of this machine. However, since displacer power flow is theoretically zero, it must be driven by the crankshaft.

$K = .75, \tau = .3373, K_r = .5, \alpha = 90^\circ F$

Figure 3.2.1-8. Basic Indicator Diagrams for Variable Volume Machines

Free Piston Operation

In free-piston machines, a self-actuated displacer is desired. In the design modification to achieve this (Figure 3.2.1-9, lower sketch),

Figure 3.2.1-9. Stirling Engine With and Without Piston Extension

the displacer has a smaller diameter extension rod protruding through the piston. This causes cyclic work to be shared between the piston and displacer extension rod. In this configuration, the correct stroke vectors for displacer ($\bar{S}_e$) and piston ($\bar{S}_p$) are

$$\bar{S}_e = \frac{\bar{v}_e}{A_c}$$

$$\bar{S}_p = \frac{\bar{v}_p}{(A_c - A_r)}$$

where A_c and A_r denote cylinder and rod areas, and $\bar{v}_e$ and $\bar{v}_p$ denote displacer and piston swept volume vectors, respectively. Indicator diagrams for the piston and the displacer extension rod are given in Figure 3.2.1-10. The net work is positive for each component and sums exactly to the piston work in Figure 3.2.1-8. Introduction of the extension rod thus enables free-piston operation.

Single-Cylinder, Two-Cycle Machine. A single-cylinder, two-cycle machine is shown schematically in Figure 3.2.1-11. Since the two cycles are 180° out of phase, one equation with a sign change in the trigonometric term applies for the pressure in both cycles. The force acting on the inner piston is derived from the pressure difference between ψ_1 and ψ_2 . Cycle work in expansion and compression spaces is determined from pressure-displacement plots for two cases. In the first, the cylinder is taken to be stationary and all three pistons move. In the second, the outer two pistons are stationary and the inner piston and cylinder move.

a) Stationary Cylinder

For the stationary cylinder, the inner piston displacement is represented by V_{e1} . The differential pressure ($\psi_1 - \psi_2$) acting on the inner piston is plotted against V_{e1} on the left in Figure 3.2.1-12, showing work in the expansion spaces to be positive and twice the net magnitude of Figure 3.2.1-8. Although the force is in the right direction for only 95 percent of the stroke, a spring force along the dashed line would make the piston self-acting in all positions. A similar force-displacement diagram for the outer piston pair is given on the right in Figure 3.2.1-12. Net work in the compression space is negative and its magnitude double that of Figure 3.2.1-8. Again, such a design would require a link mechanism to the outer pistons since external work is needed to drive them.

b) Moving Inner Piston and Cylinder

For the design with a movable cylinder and stationary outer pistons, the power developed by the piston and cylinder is

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Figure 3.2.1-10. Indicator Diagram for Stirling Engine With Displacer Extension

$$P_{\text{Piston}} = (\psi_{\phi} - \psi_{\phi-\pi})(\dot{V}_e + \dot{V}_c) \frac{A_e}{A_c}$$

$$P_{\text{Cylinder}} = (\psi_{\phi} - \psi_{\phi-\pi}) \dot{V}_c \left(1 - \frac{A_e}{A_c}\right)$$

where A_e and A_c represent expansion and compression space piston areas. Net work per cycle is obtained by integrating these power expressions over one cycle. If the cylinder net work per cycle is to be positive, A_e must be greater than A_c . This follows because energy is absorbed in the compression space and therefore $\oint (\psi_{\phi} - \psi_{\phi-\pi}) dV_c$ is negative. The pressure-volume diagrams in Figure 3.2.1-13 show that equal amounts of positive net work are done by the piston and cylinder.

Figure 3.2.1-11. Two-Cycle, Single-Cylinder Machine

Multicylinder Machines

Multicylinder configurations analyzed include a four-cylinder, four-cycle machine (Figure 3.2.1-14) and a two-cylinder, four-cycle machine (Figure 3.2.1-15). In each figure, only one of the cylinders is shown, the others being equivalent by symmetry. Pressure-displacement (indicator) diagrams for one cylinder in each of these configurations are given in Figures 3.2.1-16 and 3.2.1-17.

Figure 3.2.1-12. Indicator Diagrams for Single-Cylinder, Two-Cycle Machine (Stationary Cylinders)

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$$K = .75 \quad \tau = .337 \quad K_r = .5 \quad \alpha = 90^\circ \\ A_b/A_c = 2$$

Figure 3.2.1-13. Indicator Diagrams for Single-Cylinder, Two-Cycle Machine (Moving Cylinder—Stationary Outer Pistons)

$$v_{e1} = V(1 + \cos \phi)/2$$

$$P_1 = \frac{2WRT}{(K + \tau + \nu) [1 + \rho \cos(\phi - \theta)]}$$

Figure 3.2.1-14. Piston/Cylinder for Four-Cycle, Four-Cylinder Machine

$$v_{e2} = \frac{1}{2} V (1 - \cos \phi)$$

$$P_2 = 2WRT / [1 - \rho \cos(\phi - \theta)] (\tau + K + \nu)$$

Figure 3.2.1-15. Piston/Cylinder for Four-Cycle, Two-Cylinder Machine

Figure 3.2.1-16
Indicator Diagram for Four-Cycle,
Four-Cylinder Machine

Figure 3.2.1-17
Indicator Diagram for Four-Cycle
Two-Cylinder Machine

The indicator diagram for the four-cylinder, four-cycle machine (Figure 3.2.1-16) shows positive power conversion, but with forces in the wrong direction for a substantial portion of the stroke. Hence, a spring-inertia return and high-speed operation are required. With simple spring control, positive energy flow can be effected for the whole path except over the small inverted loop, representing about 15 percent of the stroke. Since the indicator diagram for the two-cylinder, four-cycle machine (Figure 3.2.1-17) is symmetrical, a spring return for this machine can provide positive energy flow over the entire cycle.

Performance Comparisons

In a synchronous power system there should be a positive power output from the mechanism at each instant of time. In high speed non-synchronous engines this is not essential because the periods of negative power can be accommodated by stored kinetic energy. In low speed synchronous machines the only practical energy storage is in the form of potential energy, such as spring forces or the pneumatic compression of a bounce chamber.

Efficiencies of four engines were compared. Two configurations were the single-cycle, single-cylinder constant volume and variable volume designs operating in the nonsynchronous (high-speed) mode. The other two configurations were operating in the synchronous mode with one being a two-cycle, one-cylinder design and the other being a four-cycle, two-cylinder design. Table 3.2.1-3 identifies the four engines.

Configuration A is the simplest mechanical arrangement and utilizes only one single free piston with nearly equal diameters on both ends. This is a constant volume device and as mentioned previously can produce pressure ratios of about 1.2 to 1.

The low pressure ratio is a result of using only the relative movement of working fluid between the hot and cold regions to produce pressure variations. This type of engine is currently being developed for the artificial heart power supply application by Aerojet and McDonnell Douglas. There are two known ways of energizing the piston movement: by a plug/orifice drive or by a piston extension. The latter operates in conjunction with fluid-operated valves and secondary low and high pressure storage.

Table 3.2.1-3. Comparative Regenerative Machine

Configuration	Cycles	Cylinders	Operating Mode	Volume	Figure 3.2.1-5 Sketch No.
A	1	1	Non-synchronous	Constant	2
B	1	1	Non-synchronous	Variable	5
C	2	1	Synchronous	Variable	7
D	4	2	Synchronous	Variable	8

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When two free pistons are used as in Configuration B, higher pressure ratios of the order of 2:1 are typical because compression and expansion of the working fluid are added to the heating and cooling effects. Single-cycle free-piston versions have been constructed which depend mainly on reciprocator mass and spring stiffness to maintain oscillation and to provide piston coordination. This design has to be operated at a high frequency, since no obvious way of maintaining piston movement at heart beat frequency is known at present.

Operation at heart frequency has the potential for higher overall efficiency by eliminating the requirement for a hydraulic or pneumatic frequency conversion mechanism. While this has not yet been achieved in practice, feasible approaches (Configuration C and D) are outlined here for evaluation.

From the viewpoint of mechanical efficiency, it is more desirable to use the principle of piston operation only (the Rider design) rather than of displacer operation (the Stirling design) to achieve the same cyclic sequence of events. With double-acting pistons, a fully self-actuating slow speed machine can then be designed. (Configuration C or Figure 3.2.1-18).

In this configuration the radioisotope heat source is embedded in the inner piston. This piston oscillates in a cylinder which carries two regenerators at the ends and is surrounded by insulation. This cylinder is connected by bellows to the stationary end pieces, so it is also able to reciprocate in response to the changing pressure. The space between the outer casing and the inner assembly is filled with hydraulic fluid and the whole assembly reciprocates so that the liquid can be directed to the PCCS.

An alternative configuration is shown as Configuration D in Figure 3.2.1-19. The two stepped pistons with interconnections are combined in such a way that four cycles are performed simultaneously in the mechanism. The radioisotope is in two separate internal capsules which provide heat for the four expansion spaces. These expansion spaces are connected by four regenerative ducts to the outer, small-diameter compression spaces where heat is rejected. The two pistons are coordinated so that they

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Figure 3.2.1-18. Two-Cycle, Single-Cylinder Machine

Figure 3.2.1-19. Four-Cycle, Two-Cylinder Machine

operate with a relative phase displacement of ninety degrees. The power take-off to the PCCS consists of alternating pneumatic pressure pulses supplied to the opposite faces of an actuating piston.

Heat Loss Breakdown

A breakdown of the contributing losses is given below and results are summarized in Table 3.2.1-4.

- Heat Loss from Isotope Through Insulation. A 60-watt heat source is assumed. In Configurations A and B, with the heat source oriented so that heat flows in only one direction, the insulation loss is assumed to be 10 watts. In Configuration C, since heat flow is permitted in two directions, the assumed loss is reduced to 6 watts. In Configuration D, losses occur only along the partial circumference near the center, further reducing the estimated insulation loss to 4 watts.
- Heat Loss from Isotope Along Engine. The heat loss along the engine is directly proportional to surface area and inversely proportional to path length. Therefore, it varies directly as the linear dimension L. Taking 8 watts for Configuration A and B, the corresponding losses for Configuration C are

$$2 \times 8 \times \sqrt[3]{0.5} = 12.7$$

and for Configuration D

$$4 \times 8 \times \sqrt[3]{0.25} = 20.1.$$

Table 3.2.1-4. Engine Efficiency Comparison

	A Constant Volume Thermocompressor	B Variable Volume Thermocompressor	C Two-Cycle Opposed Piston Machine	D Four-Cycle Two- Cylinder Machine
1. Direct Heat Loss From Isotope to Body Through Insulation	0.83	0.83	0.90	0.93
2. Heat Loss From Isotope Along Thermal Converter to Cooled Portions of the Engine	0.87	0.87	0.79	0.67
3. Enthalpy Losses by Leakage of Working Fluid Along Non-Power Producing Paths	1.0	0.90	0.87	0.89
4. Carnot Heat Rejection (Ideal)	0.66	0.66	0.66	0.66
5. Lack of Correspondence Between Actual and Carnot Cycle	0.70	0.80	0.80	0.80
6. Aerodynamic Flow Losses in Thermal Converter	0.90	1.0	1.0	1.0
7. Mechanical Friction Losses	0.90	0.80	0.90	0.95
8. Miscellaneous Losses - Regenerator Inefficiency	0.81	0.86	0.85	0.86
9. Allowance for Hydrostatic Bearings	0.80	0.80	0.80	0.80
Engine Efficiency	0.18	0.19	0.20	0.19

- Heat Loss by Enthalpy Transport. Losses due to enthalpy transport depend on gas leakage. For Configuration A, leakage is zero, since there are no internal pressure differences in the engine. Configuration B is used as the baseline and a loss of 3 watts is assumed for it. Losses in the other designs are scaled from the proportionality of leakage to (Pressure Difference) x (Perimeter of Gap) ÷ (Length of Flow Path). Perimeters in Configurations C and D are scaled from Configuration B by the ratio of linear dimensions. Because effective pressure difference is greater with cycles out of phase by 180 deg, an additional loss factor of 1.25 is applied for Configurations C and D. Configuration B losses are then multiplied by the following factors:

$$\text{For Configuration C: } 2 \times \sqrt[3]{0.5} \times 1.25 = 1.98$$

$$\text{For Configuration D: } 4 \times \sqrt[3]{0.25} \times 1.25 = 3.25$$

- Carnot Efficiency Limit. The same cycle temperature limits (1200°F and 100°F) are assumed for all designs. The "Carnot loss" is then $60 \times 560/1660 = 30.5$ watts.
- Deviations from Ideal Cycle. Correspondence between actual and ideal cycles is assumed to be in the range of 0.7 to 0.8. With leakage losses accounted for elsewhere, the main cause of inefficiency is heat transfer across the cylinder walls. Hence the valve-operated machine (Configuration A) is placed at the lower end of the range and the others at the high end.
- Aerodynamic Losses. A 10% aerodynamic loss (6 watts) is assumed for the gas flow through valves (Configuration A) and zero loss for the low speed devices.
- Mechanical Friction Losses. Mechanical efficiency may be independent of speed, with better hydrodynamic bearing properties at high speed compensating for increased $v \times P$ factors. Hence, the main consideration is the number of moving parts per cycle. These are 1, 2, 1, and 0.5 for Configurations A, B, C, and D, respectively. With constant loss per component, the corresponding mechanical efficiencies are taken to be 0.90, 0.80, 0.90, and 0.95.
- Regenerator Inefficiency. Regenerator effectiveness will be very high (0.96) for low speed devices and somewhat lower (0.90) for high speed devices. For practical machines these effectiveness values are further reduced by 10 percent.
- Hydrostatic Bearings. Hydrostatic bearings are to be used as detailed in a later section. The associated power loss is allowed for by an efficiency factor of 0.80 for all designs.

All four engine configurations have comparable efficiencies. All other things being equal, the potentially higher power conditioning and control subsystem efficiency of the synchronous engines should favor their selection. However, as mentioned previously, the synchronous designs (Configurations C and D) have not been reduced to practice and involve many development problems; one of which is synchronizing the engine output to match the requirements of the blood pump. One approach is to modulate the heat conducted to the engine by providing two parallel heat paths from the heat source; one by conduction directly to the TESM, the other by a heat pipe interposed between the heat source and the engine. The external surfaces of the engine expansion space form the condensing chambers for sodium in the heat pipe. The heat pipe is backfilled with argon, the pressure of which establishes the effective length of the condenser. By using a pneumatic signal from the blood pump, it would be possible to regulate the heat flow through the heat pipe as required. Another approach is to operate the engine at constant speed and have a mechanical connection between the piston and blood pump. Both rate and stroke of the blood pump are constant. In this case, no energy storage is provided and engine power equals the maximum demand of the blood pump with the excess power being dissipated at the blood pump.

Since neither of the synchronous engines nor the means of controlling them have been reduced to practice, they were not considered as legitimate candidate systems for evaluation.

3.2.1.4 Component Design

Pistons Versus Bellows

Reciprocating gas engines depend upon volume changes in a quantity of gas. These changes can be effected either by piston-cylinder assemblies or by bellows. Both mechanizations are entirely practical and the relative performance of these components was therefore investigated.

Bellows have no leakage and theoretically no friction; however, they are subject to some mechanical hysteresis. They are limited primarily by life. At relatively high speeds (above 500 strokes per minute) their use may be ruled out by fatigue. Another disadvantage of bellows is that the gas between the convolutions adds to system dead volume. This is

particularly deleterious in Stirling engines, degrading the pressure ratio and specific power.

Bellows are more suitable for short strokes with large diameters. They are most effective with two-phase working fluids, where small volume changes occur with large pressure variations.

Piston-cylinder assemblies may be distinguished by the type of reciprocating member, which may be either a displacer or a piston. Displacers are characterized by equal pressures on opposite ends and therefore need little power to reciprocate. The only force opposing their motion is aerodynamic friction due to gas displacement. With displacers, it is possible to mount regenerators either integrally within the displacer or in an external duct through which the working fluid is made to flow.

Bearings

The use of piston-cylinder assemblies depends mainly on their capability for ten-year operation without undue wear. In conventional applications, a hydrodynamic lubricating film, such as an oil film, is normally used to reduce wear. Such bearings function mainly during the high velocity periods of the stroke. With reciprocating elements, a complete stop is experienced at each end. Operation then either reverts to a hydrostatic mode or the bearing film breaks down. If there is a metallic contact at these points, wear cannot be avoided. Accordingly, hydrostatic gas bearings are preferable, since their functioning does not depend on the maintenance of high relative motion between the reciprocating parts. However, the operation of a hydrostatic bearing does require a steady leakage from the bearing pad to both ends of the reciprocating member in order to provide a constant flotation action and eliminate metal-to-metal contact. We have assumed that it will be practicable to design these bearings so that except during brief intervals, the pressure in this space is always above the pressure in the engine or in the sump.

Regenerator

Calculations indicate that the wire matrix regenerator customarily used in Stirling engines is not necessarily most efficient in miniature Stirling engines. Displacer and cylinder wall or piston and cylinder wall

surfaces may provide sufficient heat transfer area to regenerate the gas flowing in the annular gap between them.

In the very small engines under consideration here, heat transfer from the walls of a moving displacer or piston, and those of the cylinder is deemed to be sufficient. Most of the heat exchangers are made as simple tubular passages, or as annular elements between cylinder walls and piston surfaces.

3.2.1.5 System Design Summary

Two gas reciprocating engine concepts were selected as candidates for evaluation. The mechanical compression free piston nonsynchronous engine was the basic engine design. The first design operates at constant speed and constant power output, supplying a high pressure (180 psia) hydraulic fluid to the power conditioning unit. The second design is similar to the first but utilizes TESM in order to decrease the heat source size. This design is modulated by controlling the working fluid volume in the engine. Both candidate designs are discussed below.

Gas Reciprocating Engine

The design of the gas reciprocating nonsynchronous, nonmodulated gas cycle engine is shown in Figure 3.2.1-20 and the system design characteristics in Table 3.2.1-5.

The gas engine contains a 54.4-watt spherical heat source (1.4 inch diameter) within the upper end of the spherical engine housing. A gap between the heat source and engine housing allows helium to circulate and provides thermal contact between heat source and housing with minimal clearance volume. Xenon-filled Min-K insulation surrounds the heat source and the heated portions of the engine cylinder. Fins are used to minimize the temperature differential between the spherical part of the engine housing and the heated length of the engine cylinder.

To reduce heat conduction between hot and cold ends of the cylinder, the cylinder inner wall over the one-inch working length of the regenerator is reduced by special construction to a thickness of 0.003 inch. In this region the wall is reinforced with radial stiffeners spaced along the axis. The 0.010-inch outer wall and the stiffeners act as the main load carrying

Figure 3.2.1-20. Nonsynchronous, Nonmodulated Thermal Converter

Table 3.2.1-5. Design Summary Nonsynchronous, Nonmodulated Single-Cylinder, Single-Cycle Gas Engine

Component	Parameter	Units	Value
Heat Source	Thermal	watts	54.4
	Temperature	° F	1300
	Diameter	inches	1.4
Engine	Power Density	joules/in ³	14.45
	Relative Specific Power	%	60
	Efficiency	%	16
	Piston Displacement	in ³	0.08
	Bore	inches	0.5
	Stroke	inches	0.54
	Maximum Pressure	psi	500
	Minimum Pressure	psi	190
	Maximum Temperature	° F	1200
	Minimum Temperature	° F	160
	Power Output (into PCCS)	watts	7.05
	Cycle Rate	cycle/min	750
	Insulation Efficiency	%	90
	Total Weight	grams	772
	Total Volume	cc	870
	Dimensions	inches	3.04 x 3.04 x 5.74

members and circumferential slots cut in the outer wall act as heat dams. Beyond the regenerator region the inner wall thickness increases to 0.025 inch. Calculations for this construction indicate heat conduction losses to be 2.8 watts, approximately five percent of the heat source output.

The displacer piston is of hollow construction to limit heat conduction between expansion and compression ends of the engine. The hot end of the displacer is contoured to match the shape of the heat source. The cylindrical walls of the displacer are thin (0.003 inch) while the ends and internal radiation shields are 0.010 inch. Holes in the displacer and shields reduce the differential pressure. The gap between the displacer and engine cylinder walls acts as a clearance regenerator.

The hollow-core displacer is approximately 1.5 inches long by 0.5 inch diameter. The diameter of the displacer extension rod passing through the power piston is 0.25 inch. One spring attached to the displacer rod is recessed internally and connects the piston to the displacer rod. Another spring is internally recessed within the power piston and connected to the hydraulic bellows. These springs assist in maintaining the proper phase relationship and rebound energy for the piston and displacer.

The power conditioning system is a separate package in which the pneumatic pressure of the engine is converted to hydraulic power at the bellows. Relatively low pressure isotonic fluid is introduced through the inlet check valve from the PCCS and compressed to 180 psi by the engine. The fluid flows out of the engine through the outlet check valve into the annular cold-end heat exchanger where it picks up waste heat. It is then discharged into a tubular heat exchanger within an annular gap formed by an inner and outer container. This gap is also filled with an isotonic solution which absorbs the waste heat and distributes it over the entire outer surface of the package at a maximum heat flux of 0.06 watt/cm^2 .

Gas bearing support is provided to minimize power piston wear by bleeding gas from the compression space through the end of the displacer rod and engine cylinder wall. A series of differential flow orifices act as unidirectional valves to control the flow of gas. Developing a satisfactory

gas bearing support and minimizing its effect on the engine performance are considered major design uncertainties.

Gas Reciprocating/TESM Engine

Characteristics of the gas reciprocating/TESM, modulated, non-synchronous engine are given in Table 3.2.1-6. Its design (Figure 3.2.1-21) resembles the nonmodulated system but differs as follows:

- The engine is modulated by controlling dead space volume in the compression end by means of a pressure signal from the PCU.
- Part of the PCU is an integral part of the engine package and also acts as a distributor for waste heat rejection.
- Energy storage has been added. TESM capacity is 53.3 watt-hours. Internal TESM fins maintain heat transfer rates during solidification.

The use of TESM lowers the required heat source thermal power. However, the reduction in heat source size is offset by the need for engine speed modulation. Consequently, efficiency is variable and slightly lower than for a nonmodulated engine. Maximum efficiency occurs at the average power level and is less at the lower and higher levels.

The signal to modulate engine power and frequency is proportional to the displacement of the diaphragm between high and low pressure accumulators. An alternate mode of modulation was considered using heat pipes and varying their conductance by means of an argon bellows at the condenser end. This scheme was rejected, however, because of its complexity.

The engine has five moving parts: displacer, power piston, bellows and two liquid check valves. Gas bearings are used to reduce wear and are again considered to be a major design uncertainty. The displacer piston is centered within the cylinder by the power piston and aerodynamic forces. Metal-to-metal contact may occur at the end of the stroke when velocity is zero. Since the two pistons are never simultaneously at top or bottom dead center, however, one is always moving to provide aerodynamic forces.

Wearout should be the major reliability consideration. The springs and bellows have been designed to maintain the fluctuating stresses within fatigue limits of the materials.

Table 3.2.1-6. Design Summary Gas Reciprocating/TESM Engine

Component	Parameter	Units	Value
Heat Source	Thermal Power	watts	48.9
	Temperature	° F	1300
	Diameter	inches	1.31
	TESM (LiF/NaF)	watt-hours	53.3
Engine	Power Density	joules/in ³	14.45
	Relative Specific Power	%	50
	Efficiency (Max/Min)	%	12/8
	Piston Displacement	in ³	0.07
	Bore	inches	0.5
	Stroke	inches	0.337
	Maximum Pressure	psi	500
	Minimum Pressure	psi	189
	Maximum Temperature	° F	1200
	Minimum Temperature	° F	160
	Power Output (into PCCS)	watts	4.1 - 6.7
	Cycle Rate	cycle/min	480 - 1000
	Insulation Efficiency	%	90
	Total Weight	grams	874
	Total Volume	cc	681
	Dimensions	inches	3.34 x 3.34 x 5.34

Figure 3.2.1-21. Nonsynchronous Modulated Thermal Converter

3.2.2 Linear Vapor

3.2.2.1 Historical Background

The technology for the linear vapor engine dates back over 200 years. This technology includes data for various engine types; expansion and non-expansion, condensing and noncondensing, and single and multiexpansion. Considerable data are also available* on the use of the vapor engine as an energy conversion device for an artificial heart assist system. These data include both analytical and experimental results for three distinct types of vapor engines all operating with different cycle conditions. These three engines are termed the reciprocating, ram, and tidal regenerator engine (TRE). An efficiency summary for these three engines is presented in Table 3.2.2-1.

Table 3.2.2-1. Artificial Heart Vapor Engine History

Engine Type	Working Fluid	rpm	Boiling Pressure and Temperature		Ideal Cycle (%)	Machine			Pump Volumetric (%)	Overall (Thermal to Hydraulic) (%)
			(psia)	(°F)		Mechanical (%)	Thermal (%)	Total (%)		
Reciprocating	Water	1500	250	900	28	44	71	31	79	~7
Reciprocating	Organic (CP-34)	1500	250	500	24	75	59	44	80	~8.5
Ram	Water	120	95	900	13	91	89	81	74	~8
Ram	Water	1500	95	900	13	86	84	72	74	~7
TRE	Water	50	160	900	14	NA*	NA*	32	NA*	~4.5

* NA - Not available

The reciprocating engine is representative of the conventional type piston engine utilizing a connecting rod, crankshaft, and flywheel. The engine design consists of a combined vapor and hydraulic piston connected by a universal crankshaft all housed within a single cylinder. A vapor exhaust valve is located within the vapor piston. The engine employs the Rankine cycle

* Data from Thermo Electron Corporation, Waltham, Mass.

which permits high ideal efficiencies. A flywheel is used to average out the pressure variations during expansion and to provide for piston return.

Using water as the working fluid, the measured overall efficiency (thermal to hydraulic) was ~7%. The low efficiency (25% of ideal) was caused by mechanical inefficiencies associated with the friction of water-lubricated bearings. Additional problems were also encountered because of fluid leakage and wear rates. The longest reported run, using water as the working fluid, was less than 4 days.

To reduce the friction on the bearing surfaces, an organic working fluid (Monsanto CP-34) was used. To enhance its lubricating properties and reduce wear rates, silicone oil was added, resulting in higher mechanical efficiencies as shown in Table 3.2.2-1. The thermal efficiency in this case was reduced because of pressure drops through inlet and exhaust valves caused by the higher mass flow required with the organic fluids.

In short, while high ideal efficiencies of 24 to 28% were possible with the reciprocating engine, efficiencies of only about one-quarter of ideal were attainable. Additional problems included flywheel weight, design complexity, and lifetime and reliability because of bearing and piston seal wear and number of moving parts.

To reduce the friction and wear of bearings, and to maintain long-term performance, a constant-pressure engine, termed the ram engine, was developed. This was accomplished by using a free-piston engine that eliminated the need for a crankshaft, connecting rod, flywheel and bearings. This engine operated with the full vapor pressure exerted on the piston at all times. Since there is no expansion work, the cycle efficiency is approximately one-half that of the expansion engine.

In the design of the ram engine a three-faced piston was used which combines the vapor engine, hydraulic pump and feed pump into a common assembly. A spring was used to return the piston. Measured efficiencies for the ram engine are shown in Table 3.2.2-1. Although the ideal cycle efficiency of the ram engine was approximately one-half that of the reciprocating engine, similar overall efficiencies were obtained. This was due to the increased machine (mechanical and thermal) efficiency

that resulted from the simpler design used in the ram engine. Even though the bearings, flywheel, crankshaft, and connecting rod were eliminated, piston seals caused wear and leakage problems. In addition, thermal losses were still present between the vapor and hydraulic fluid because of conduction losses in the piston and cylinder.

To eliminate the need for valves or sliding seals an electronically controlled vapor cycle engine, termed the "tidal regenerator engine" or TRE, was developed. The TRE is a vapor cycle analog of the regenerative gas cycle engine which combines some of the characteristics of the Rankine engine with those of the Stirling engine. The advantage of the TRE is its simple mechanical nature. It contains neither valves nor sliding seals and the only moving parts are the bellows, binary solenoid, and interface piston.

The vapor cycle is basically a sealed column filled with working fluid, having a displacer bellows at the cold end and a piston output bellows at the hot end. Located between the displacer and piston output bellows in the following order are a condenser, liquid (tidal) regenerator, boiler, vapor regenerator, and superheater.

The engine cycle begins with the liquid level at the interface of the condenser and liquid regenerator. As the displacer bellows, operated by a binary (double-acting) solenoid, moves, liquid is shifted from the condenser through the liquid regenerator to the boiler. The liquid is vaporized until the entire column reaches a new and higher pressure level. The piston output bellows then pushes on an interface piston (the piston is used to separate the high temperature vapor from the low temperature hydraulic fluid) which in turn pushes on the hydraulic fluid bellows, sending high pressure fluid to the blood pump. At the completion of the power stroke, the displacer bellows moves in the opposite direction lowering the liquid level in the sealed column. As the liquid level in the column recedes, the vapor is returned to the condenser at constant volume. As the vapor condenses, the column or system pressure decreases. When the system pressure is a minimum, the hydraulic fluid is returned by the spring in the blood pump. The hydraulic fluid in turn returns the hydraulic fluid bellows and interface piston to "top dead center" which then completes the cycle.

During the operation of the TRE, both the vapor and liquid regenerator regenerate much of the heat except for a large fraction of the heat of vaporization. Since the TRE operates with the absence of any expansion work, ideal cycle efficiencies comparable to the ram engine are obtained. By using a higher degree of superheat, improved TRE efficiencies, compared to the ram, are possible. Higher TRE efficiencies are also possible if a working fluid mixture (such as ammonia and water) is used that has a nonconstant-temperature vaporization curve, since more heat of vaporization is available for regeneration.

The efficiency of the TRE with water as the working fluid is shown in Table 3.2.2-1. The measured overall efficiency of the TRE was approximately one-half that of the reciprocating and ram engine. Data were also available for Fluorinol, which had an efficiency comparable to water. The efficiency degradation for the TRE is probably caused by regenerator inefficiencies, thermal losses, and bellows losses. Besides the low measured efficiencies, the TRE had a large volume (~2.35 liters excluding the blood pump) and appears sensitive to gravity effects, shock and vibration. If a mixed working fluid were used to increase the cycle efficiency, fluid breakdown or decomposition could present a problem because of the small volume of working fluid.

3.2.2.2 Working Fluid Selection

The choice of the working fluid and its related thermodynamic cycle characteristics for a heat engine is of fundamental importance. Factors which influence its choice are cycle efficiency, operating temperature and pressure, thermal stability, toxicity, and corrosiveness. The candidates that were investigated included: water, Monsanto CP-34, Fluorinol 51 and 85, and various other organic fluids.

The ideal cycle efficiencies for each of the working fluids were investigated as a function of inlet temperature, inlet pressure and condenser pressure for both an expansion and nonexpansion cycle. An example of a typical data set is shown in Figure 3.2.2-1. The ideal efficiency of water is plotted for an expansion and nonexpansion cycle as a function of inlet temperature and pressure. The results of these investigations showed that for a given inlet and condenser pressure, similar cycle efficiencies

would be obtained with both water and the organic fluids; but with the organic fluids, the same cycle efficiency could be obtained at a substantially lower temperature.

Hence, on the basis of the thermodynamic cycle characteristics, the organic working fluids appear potentially superior to water. Unfortunately, the use of an organic fluid in a linear vapor reciprocating engine does present other potential problems. A prime concern is avoiding decomposition resulting from overheating the working fluid. This type of problem is common to reciprocating engines that operate at heartbeat frequency. Since the engine operates at low speed, the working fluid flow is pulsatile and not continuous. This makes it difficult to design a boiler that will not overheat the working fluid. For this reason, water was chosen as the

preferred working fluid for the linear vapor engine. Other desirable characteristics of water are: good cycle efficiency, thermal stability, nontoxicity, and the availability of a large amount of prior operating experience.

3.2.2.3 Cycle Selection

Two cycles were investigated for use in the linear vapor engine. One was the Rankine or complete expansion cycle. The other was the nonexpansion cycle, sometimes termed the ram cycle. Both of these cycles have typically been used with a fluid that is alternatively vaporized and condensed. Their main advantage is the small fraction of the total engine work required for pump work. The primary components for both cycles are the same: boiler, condenser, regenerator, feed pump and engine.

Figure 3.2.2-1.
Comparison of Expansion and
Nonexpansion Cycle
Efficiency

A comparison of the ideal efficiencies for these two cycles is shown in Figure 3.2.2-1. In the ideal Rankine cycle, the working fluid undergoes a complete isentropic expansion between the boiler and condenser

pressure. In the nonexpansion cycle, a constant pressure is maintained during the entire stroke, as shown in Figure 3.2.2-2. During this process, no work is obtained from the internal energy of the working fluid as in the expansion cycle. Figure 3.2.2-1 shows that the ideal efficiency of the nonexpansion cycle is about one-half that of the expansion cycle.

To be able to fully utilize all the available cycle energy, the shape of the engine output pulse must be properly matched with the blood pump. Since the blood pump requires a close-to-constant pressure actuating pulse*, a similar engine output pulse is required. In the expansion cycle, the working fluid undergoes an isentropic expansion ($PV^\gamma = \text{constant}$), as shown in Figure 3.2.2-2, where the working fluid expands isentropically from 95 to 15 psia. This produces a variable force as shown in Figure 3.2.2-3. The force ranges from 129 to 20 pounds with a mean effective force of 47 pounds. For a 1-inch stroke, this is equivalent to

Figure 3.2.2-2. Pressure-Volume Relationship for Expansion and Nonexpansion Cycle

*As discussed in Section 2.3 this requirement is misleading and for some systems may result in an underestimation of the true power requirements by 20 to 80%.

a work output of 47 in-lb. Since the output of the engine will be used to either pressurize an accumulator or drive the blood pump directly, the maximum force the engine can exert over its entire stroke is 20 pounds. If a larger force were required, the engine would be unable to maintain this force during its entire stroke. The result is that less than one-half of the total available work output from the expansion engine can be utilized as useful work.

Two methods can be used to overcome this problem. One is to store the unavailable energy by the use of a flywheel. This method was ruled out based upon the complexity and additional weight such a design would introduce. A second method is to use a compound or multiexpansion engine.

Figure 3.2.2-3. Force Relationship for Expansion and Nonexpansion Cycle

In this type of engine, the working fluid is allowed to expand in various stages by passing it from one piston to another by the use of an interceding receiver. It is possible, through the use of multiple piston-receiver combinations, to reduce the output from an expansion cycle into a form that approximates a constant force. An analysis of this arrangement shows that the potential gains are offset by the pressure losses of transferring the fluid from one piston to another via the receiver.

It was concluded that the cycle from which the most work could be obtained, and still meet the blood pump requirements, was the non-expansion cycle. Figures 3.2.2-2 and 3.2.2-3 show that in the non-expansion cycle, a constant pressure or force is exerted during the entire stroke. Also, for the same operating conditions, the nonexpansion cycle produces 32 pounds of force for the entire stroke compared to 20 pounds for the expansion cycle. The nonexpansion engine has several other

desirable characteristics besides a constant-pressure output pulse. Two of the most important are that a constant engine temperature is maintained which greatly minimizes cylinder condensation, and, for a given cylinder size, the maximum possible work is obtained.

To select the operating temperature and pressures for the nonexpansion cycle, reference is made to Figure 3.2.2-1. This figure was constructed for a condenser temperature and pressure of 142°F and 3 psia. Since approximately 30°F of subcooling is required to prevent cavitation of the working fluid during pumping, it was felt that the condenser temperature is reasonable compared to the body temperature. Figure 3.2.2-1 shows that the efficiency of the nonexpansion cycle is sensitive to inlet temperature but is relatively insensitive to inlet pressure. Since it was desirable to obtain the maximum efficiency, an operating temperature and pressure of 900°F and 95 psia were selected. This results in an ideal cycle efficiency of 14.2%.

3.2.2.3 Configuration Selection

Two different methods exist for operating the nonexpansion engine. In one method the working fluid is pressurized, heated and cooled external to the engine. In the other this is accomplished within the engine itself. These two methods are termed the ram and tidal regenerator engine (TRE), respectively. A temperature-entropy diagram for each engine is shown in Figure 3.2.2-4. Both engines rely on regeneration to increase the cycle

Figure 3.2.2-4. Comparison of Two Nonexpansion Thermodynamic Cycles: Ram (a) and TRE (b)

efficiency. In the ram, regeneration occurs at constant pressure and in the TRE at constant specific volume. In the latter case a larger amount of heat is available for regeneration. In the ram a feed pump, regenerator, condenser, boiler, and engine intake and exhaust valves are required. For the TRE, a regenerator, condenser, boiler and solenoid-actuated displacer bellows are required. The advantage of the TRE is that the feed pump and engine intake and exhaust valve are eliminated in favor of the displacer bellows. The TRE is shown in Figure 3.2.2-5. Its operation is similar to that described in Section 3.2.2.1, except blood pump actuation is accomplished mechanically instead of hydraulically. To meet our power requirements the engine must be capable of delivering 5.84 watts. This power is based on an automatic actuator efficiency of 76% and a blood pump power of 4.44 watts. The ideal efficiency of the TRE is 14.8%. To meet the volume and power requirements for this application an overall cycle efficiency of 10.3% is required. This efficiency is twice the previously reported value. Even though the TRE has the advantage of eliminating the feed pump and engine exhaust and intake valves, it is believed that based on the currently available experimental data it does not have the capability of meeting our power requirements.

Figure 3.2.2-5. Tidal Regenerator Engine (TRE)

For the ram engine, the engine output can either be in the form of hydraulic or mechanical power. If the output is hydraulic, it can be used to either drive the blood pump directly or pressurize an accumulator. However, the latter type of design is unacceptable since an isotope inventory of 71 watts will be required. Figure 3.2.2-6 shows that a volumetric

Figure 3.2.2-6.
Hydraulic Pump Flow
Characteristics

efficiency of 76 to 78% can be obtained with a hydraulic pump. This type of inefficiency is eliminated by using a mechanical link that directly connects the output of the engine to the blood pump.

Since a mechanical, rather than a hydraulic link is used to drive the blood pump, electrical power is needed for the engine intake and exhaust valves. In addition, electrical power is used to operate the feed pump as the engine design precludes the use of either mechanical or hydraulic power. The electrical power is provided by a thermoelectric module.

Based on the actuator and engine efficiency, two designs of a linear vapor engine were selected.

One operates with a variable speed at heart beat rate, the other with a fixed output at 120 bpm. The configuration for both is the same, except one design uses TESM. By using TESM the isotope inventory is reduced, but a more complicated engine and actuator design are required since the engine operates at variable speed.

3.2.2.4 Component Selection and Design

As mentioned previously, the components for the two designs of the linear vapor engine are the same except one design uses TESM and the other does not. The major components in each design are: engine, electrical feed pump, regenerator, condenser, boiler, thermoelectric module, and electrically-actuated engine intake and exhaust valves. In both designs

the same maximum power output is required since the automatic actuator efficiency is the same. Thus, the same size components are used in both designs since the components must be designed for maximum engine output. Both designs have been optimized for the same operating conditions of 95 psia at 900°F engine inlet conditions and 3 psia and 142°F condenser conditions. The following presents a discussion of the selection and design of each component in the linear vapor engine.

Engine

A number of different engine configuration alternatives were investigated. They included piston versus bellows, linked versus free, and crankshaft versus spring return. Analysis of a linked piston with a crankshaft and flywheel return showed that besides excessive friction losses the overall weight of the design was prohibitive. The friction losses are particularly acute since the only available lubricant is water. In considering the use of a piston-cylinder combination or a bellows, the fluid leakage rates were examined.

Since such a small quantity of working fluid is present, it was found that even a minute leakage rate could result in a large degradation in cycle efficiency. Analysis of the leakage rates showed that rings must be used if a piston-cylinder combination is considered. It was found that without rings a clearance of 0.0001-inch was required to reduce the leakage to less than 3%. Clearances of this size would probably cause excessive friction and tend to cause engine seizure on heating because of differential thermal expansion. Since the use of piston rings is required, the problem of wear must be considered. Even though water-lubricated graphite rings would be acceptable, the problem of wear and eventual leakage would not be eliminated. To circumvent this problem, it was decided to use bellows instead of a piston-cylinder combination. The spring rate of the bellows is used to return the bellows to their original position. Data available on welded metallic bellows show that both the desired spring rates and lifetimes are feasible. Since the bellows act as a hermetically sealed device, fluid leakage losses are completely eliminated. Also the welded bellows act as a nonfriction device which avoids wear. Thus the use of bellows eliminates two of the major problems that were present in previous vapor engines of this type.

Since the engine is mechanically connected to the blood pump, their strokes must match. The size of the bellows is based on the maximum vapor flow rate, which is 41 in³/min. For a 1.3-inch stroke, the mean effective diameter of the bellows is 0.58-inch.

Condenser

The condenser and subcooler have been designed to transfer the heat from the condensing vapor to the package walls where it is then rejected to the body fluids and tissue. In the condenser the vapor is cooled from a saturated vapor to a saturated liquid at 3 psia, and is then subcooled approximately 30°F to prevent cavitation of the fluid during pumping. The condenser was designed based on a maximum allowable package temperature and heat rejection surface area of 107.6°F (42°C) and 0.07 watt/cm², respectively.

The maximum condenser heat load is 46 watts. The heat transfer coefficient from the vapor to the package wall was calculated using the equation (Reference 19)

$$h_m = \frac{0.065 \sqrt{\frac{\rho_l f}{\rho_v^2}} C_p G_m}{\sqrt{\frac{C_p u}{k}}}$$

where

h_m = mean heat transfer coefficient

C_p = specific heat

u = viscosity

k = thermal conductivity

ρ_l = density of liquid

ρ_v = density of vapor

f = friction factor

G_m = 0.58 of mass velocity (velocity ÷ area of tube)

and the liquid properties of the working fluid. The condenser size was determined by use of the convective heat transfer equation ($Q = h_m A_s \Delta T$) and a tube wall effectiveness factor of 0.7. The condenser is 30 inches long with a 0.0625-inch ID and a 0.020-inch wall. In order to reject the heat uniformly over the required package surface area, the condenser tube must be finned.

Boiler

Previous boiler designs for steam cycle engines in this size range have proven to be inadequate to achieve 100% steam quality or the desired degree of superheat. This deficiency may have been caused by heat transfer from the boiler tube to the vapor only, neglecting the liquid fraction. However, in a comparatively low speed reciprocating engine, the flow through the boiler may be in the slug flow regime and the hydrodynamic behavior of the liquid dispersion cannot be ignored.

The formulation for the heat transfer coefficient (Reference 20) for slug flow is as follows:

$$h = \frac{\left(\frac{48}{11} \frac{k_f}{D}\right) (1 - x)}{1 - x (1 - \rho_f / \rho_v)}$$

where

h = heat transfer coefficient

k_f = thermal conductivity of fluid

D = tube diameter

x = steam quality

ρ_f = density of the fluid

ρ_v = density of the vapor

The boiler tube size including the superheater, using the heat transfer coefficient and the convection heat transfer equation $Q = h A_s \Delta T$, was determined to be 18.7 feet long for a 0.0625-inch ID.

By using the Dittus-Boelter equation

$$h = 0.023(Re)^{0.8}(Pr)^{0.4}$$

where

h = heat transfer coefficient

Re = Reynolds number

Pr = Prandtl number

and assuming heat transfer to the vapor only, the length of the boiler and superheater for a 0.0625-inch ID tube would be 14.4 feet. As seen, the design for slug flow results in a longer boiler length. However, slug flow may only exist for steam qualities up to 30%; therefore, the required length will probably be between the two calculated results.

A different approach to the boiler configuration would be a packed bed boiler in which an annular housing packed with spheres made from a high conductivity material is located around the heat source. Heat flows from the heat source walls to the spheres and then to the fluid. The concept is attractive since the spheres in the packed bed provide a large heat transfer surface for the fluid. Data on packed bed boiling two-phase flow are meager. However, if the approach of considering heat transfer to the vapor only is taken, the graph in Reference 21, which pertains to gas flow through an infinite, randomly stacked, sphere matrix, may be used to determine the required boiler design. Again, this may be an inadequate approach for the same reasons as in the tube-type boiler design. A packed bed boiler design would appear to increase the weight and size of the engine system over the tube boiler design because of the weight of the material utilized in the packed bed (spheres) and the requirement of a sufficient annular volume to achieve the boiling and superheating of the fluid. However, if hot spots are a problem in the tube type boiler because of slug flow, a packed bed boiler approach may prove to be attractive.

Regenerator

Regeneration is used in most vapor cycles to increase the overall system efficiency. When a nonexpansion cycle is used, regeneration becomes quite important. Without it, the cycle efficiency could be reduced by almost one-sixth. The regenerator design selected for this engine is a conventional vapor-to-liquid concentric tube counterflow design. The liquid from the pump enters into the annulus of the outer tube where it picks up heat through the tube wall from the vapor which is flowing in the inner tube. In sizing the regenerator, consideration was given to the requirements of a low pressure drop on the vapor side to negate problems with the liquid pump. Because of the low flow rate, the pressure drop on the liquid side will be negligible.

The design consists of a 0.08-inch ID inner tube (vapor side) with a 0.020-inch wall and an outer tube with an ID of 0.16-inch and an OD of 0.20-inch. The length of the tube is 3 inches.

Thermoelectric Module

The electrical power for the engine feed pump, intake and exhaust valves, and control unit is provided by a thermoelectric module. The module is located in series thermally between the heat source and boiler. The pertinent characteristics of the thermoelectric module are:

Hot cap temperature	1500°F
Cold cap temperature	950°F
Thermoelectric material	SiGe
Element size	N 0.11-inch diameter x 0.2-inch long
	P 0.09-inch diameter x 0.2-inch long
Number of couples	14
Number of strings	2
Electrical power output	1.14 watts
Load voltage	0.7 volt
Conversion efficiency	2.0%

Feed Pump

An electrical feed pump is used to pump the liquid from the condenser to the boiler. This pressure range is from 3 to 95 psia. The feed pump was determined to have a 9% conversion efficiency from electrical to mechanical to hydraulic power. This requires a 0.21 watt electrical input. The feed pump consists of an electrically-driven solenoid with a stroke of 0.10-inch and diameter of 0.18-inch.

Engine Intake and Exhaust Valves

The engine intake and exhaust valves are located external to the engine and are actuated by an electrically powered solenoid. A detailed design for these valves was not undertaken, but a simple spool valve or similar device can be used to regulate the flow of vapor into and out of the engine. The valve would be operated by a solenoid which would have only two interlocking positions. The electrical power required to operate the solenoid will be less than 0.5 watt.

TESM

In the design of the synchronous vapor engine, a thermal energy storage material is used to reduce the isotope inventory. The material uses the latent heat of fusion to absorb excess thermal energy isothermally. The material chosen was LiF/LiCl since it has a melting point (930°F) slightly higher than the boiler temperature (900°F). This material has an energy density of 87.5 watt-hr/lb and a volumetric density of 6.9 watt-hr/in³. Since 69 watt-hours of thermal storage are required, the weight and volume of TESH are 0.79 lb and 10 in³, respectively.

3.2.2.5 System Design Summary

Two designs of a linear vapor engine were selected. One operates with a variable speed at heartbeat rate. This design uses TESH and delivers a power output equivalent to the average blood pump requirements. The other operates with a fixed speed and output at 120 bpm. This design delivers a constant power output equivalent to the maximum blood pump requirement. The components for both designs are the same except one uses TESH to reduce the isotope inventory.

The insulation used in both systems was Min-K 2020 filled with an inert gas (xenon). The package sizes were determined using the average isotope inventory and were based on a maximum allowable package and heat rejection surface area of 107.6°F (42°C) and 0.07 watt/cm^2 , respectively. A summary of the pertinent design characteristics for each system is given below.

Linear Vapor Engine/TESM

The design for this engine is shown in Figure 3.2.2-7. An electrical control system is used and its power is supplied by a thermoelectric module. The engine consists of a bellows with a mechanical output drive that is directly coupled to the blood pump. Bellows are used to eliminate fluid leakage and mechanical friction and wear.

The engine operates with the vapor from the boiler driving the engine bellows. Exhaust vapor from the engine passes through a regenerator into a condenser where it is condensed and subcooled. The condenser is designed to reject the heat to the package walls where it is then rejected to the body fluids and tissue. The fluid from the condenser passes through the feed pump where it is pumped through the regenerator into the boiler. The engine is controlled by regulating the vapor flow rate into and out of

Figure 3.2.2-7. Linear Vapor Engine/TESM

the engine. Since the engine operates with a constant pressure, the inlet valve is open during the entire engine stroke. At the end of the stroke the inlet valve is closed and the exhaust is opened. The exhaust valve remains open during the entire exhaust stroke. By regulating the inlet and exhaust rate, the engine speed can be varied over the desired range.

The engine power output and isotope heat input as a function of engine speed are shown in Figure 3.2.2.8. The heat source fuel inventory was based on the average blood pump power requirements. The heat source inventory is 48 watts. To accommodate periods of peak activity, 69 watt-hours of TESM are required. The overall engine efficiency versus engine power output is shown in Figure 3.2.2-9. The engine efficiency is seen to increase with power output. The average daily efficiency of the engine is 8.0%. Table 3.2.2-2 summarizes the pertinent engine design characteristics and a system weight breakdown is presented in Table 3.2.2-3. The volume of the engine subsystem is 1.152 liters and the weight is 1.36 kg.

Linear Vapor Engine

The vapor engine operates at constant frequency and delivers constant power at all times. The power output is equivalent to the maximum blood pump requirements. A schematic of the engine is shown in Figure 3.2.2-10 and its operation is similar to that of the previous engine. Since the engine delivers maximum power, a larger heat source inventory is required. The advantages to be gained by delivering a constant power output are that the need for thermal energy storage is eliminated and the engine controls are simplified since the engine operates at constant frequency. In addition, the problem of variation in the daily power profile on the heat rejection rate are eliminated.

Figure 3.2.2-8.
Linear Vapor Engine/TESM

Figure 3.2.2-9. Linear Vapor Engine/TESM

Table 3.2.2-2. Linear Vapor Engine/TESM

Operating Conditions	
Boiler	95 psia, 900°F
Condenser	3 psia, 142°F
Subcooler	3 psia, 115°F
Efficiency	
Ideal Cycle	14.2%
Machine	49 to 60%
Overall	7 to 8.6%
Engine Characteristics	
Speed	60 to 120 bpm
Stroke	1.3 inches
Diameter	0.58 inch
Power Output	2.7 to 5.9 watts
Feed Pump	
Stroke	0.10 inch
Diameter	0.18 inch
Power Requirement	0.21 watt

Table 3.2.2-2. Linear Vapor Engine/TESM (Continued)

Thermoelectric Module	
Material	SiGe
Number of Couples	14
Electrical Power Output	0.96 watt
Voltage	0.6 volt
Engine Subsystem	
Volume	1.152 liters
Weight	1.36 kg
Power Input	48 watts
TESM	69 watt-hours

Table 3.2.2-3. Weight Summary

	Linear Vapor/ TESM (lb)	Linear Vapor (lb)
Engine		
Heat Source	0.47	0.55
Thermoelectric Module	0.40	0.40
TESM	0.79	-
Boiler	0.13	0.13
Engine Inlet and Exhaust Valve	0.11	0.11
Regenerator	0.03	0.03
Feed Pump	0.04	0.04
Condenser	0.03	0.03
Engine (Bellows and Lines)	0.10	0.10
Electronic Package	0.17	0.17
Outer Housing	0.16	0.16
Insulation	<u>0.57</u>	<u>0.78</u>
Total	3.01	2.51
	(1.36 kg)	(1.14 kg)

Figure 3.2.2-10. Linear Vapor Engine

A summary of the engine design characteristics is presented in Table 3.2.2-4. The isotope inventory is 57 watts and the volume of the engine subsystem is 1.365 liters. Since the need for TESM was eliminated, the weight of the system was reduced as shown in Table 3.2.2-3.

Table 3.2.2-4. Linear Vapor Engine

Operating Conditions	
Boiler	95 psia, 900°F
Condenser	3 psia, 142°F
Subcooler	3 psia, 115°F
Efficiency	
Ideal Cycle	14.2%
Machine	72%
Overall	10.2%
Engine Characteristics	
Speed	120 bpm
Stroke	1.3 inches
Diameter	0.58 inch
Power Output	5.9 watts
Feed Pump	
Stroke	0.10 inch
Diameter	0.18 inch
Power Requirement	0.21 watt
Thermoelectric Module	
Material	SiGe
Number of Couples	14
Electrical Power Output	1.14 watts
Voltage	0.7 volt
Engine Subsystem	
Volume	1.365 liters
Weight	1.14 Kg
Power Input	57 watts

3.2.3 Rotary Vapor

3.2.3.1 Historical Background

One of the possible approaches to the conversion of thermal energy into electrical or mechanical energy is the use of the rotary Rankine cycle. This cycle is attractive for this application since there are various thermodynamic and heat transfer advantages in using a working fluid in both its liquid and vapor phases. An important advantage of the Rankine cycle is the greatly reduced ratio of compression work to expansion work which results from pumping a liquid rather than a gas. For example, in an ideal Rankine cycle, this ratio can be less than 1.0% when water is the working fluid. This compares with the 30% ratio in a typical Stirling cycle. Thus, a Rankine cycle can be essentially independent of the efficiency of the compression process (i. e. , pump work). In addition, in the cycle the condenser and boiler both involve either a condensing or boiling fluid, thus increasing the overall heat transfer coefficient. Both Stirling and Brayton cycles require gas heat exchangers, thus requiring greater heat transfer area.

In the past consideration was given to the use of a rotary expander driven hydraulic pump. It was felt that in order to achieve as high an efficiency as possible, a positive displacement expander would be required. Steam was selected as the working fluid because of its high energy content and its compatibility with heat source and body temperature constraints. Because of its continuous unidirectional motion, the elastic response of a curtain (slide type) valve and a continuously open exhaust, the potential problem of destructive action of trapped liquid could be eliminated, permitting the use of "wet vapor" in the system. A system of this type, as proposed by Westinghouse, had an estimated overall efficiency of 8%. This design was excessively heavy (17.5 pounds including heat source), mechanically complex (rotary and curtain valves, chain and sprocket wheels, etc.) and very sensitive to thermal conduction losses (low conductivity titanium alloy housing required). The latter condition limited the maximum cycle operating temperature to 389°F (218 psia). This coupled to a condensing temperature of 150°F (3.47 psia) to minimize the moisture content at the end of the expansion process resulted in a Carnot efficiency of only 36%. Thus, this system, at an estimated

overall efficiency of 8%, would achieve only 22% of the Carnot efficiency. The use of working fluids other than water was not considered. However, a so-called advanced engine concept operating at an increased peak cycle temperature of 500°F and with the titanium alloy engine casing replaced with a fiber-filled Teflon material was considered. This configuration was estimated to achieve an overall efficiency of 12.7%. However, the previously mentioned shortcomings (excessive weight and mechanical complexity) still existed. Work on this heat engine concept was terminated by the NHLI and no additional data on any future development efforts were available.

The second approach that can be considered for the rotary Rankine cycle system is turbomachinery. This concept had been looked at only in a cursory manner by previous investigators and summarily dismissed as not being applicable to this low power output application. It was felt that efficiencies would be too low. In addition, the high rotative speed required would necessitate the use of reduction gears in order to drive the blood pump at synchronous speed. A second factor which may have contributed to lack of interest in turbomachinery was that if a generator were utilized, the thermal-to-electrical converter had to be coupled to an electrical-to-mechanical/hydraulic converter to operate the blood pump. It was postulated that this system could not be competitive with a direct thermal-to-mechanical converter device. There are several factors, which when examined in a more comprehensive manner than has been done previously, result in arriving at a different conclusion. First, component efficiencies need not be low. For example, as will be discussed later in this section, turbine efficiencies of 55 to 60% and generator efficiencies of 90 to 95% can be achieved even at the low power levels involved. In addition, as was previously discussed, the pump efficiency is not critical in the Rankine cycle. However, even for this component, acceptable efficiencies in the range of 30 to 40% can be obtained using proper design techniques. Also overlooked in previous assessments was the considerable potential for achieving higher turbine and overall cycle efficiencies through the use of high molecular weight working fluids. Because the thermodynamic characteristics of many of these fluids permit the use of regeneration, a significant increase in overall cycle efficiency can be achieved. In addition, for those systems providing an electrical output, with a turbogenerator,

the use of a secondary battery for energy storage is also feasible. This permits sizing of the system for blood pump average power requirements and results in a further reduction in heat source thermal requirements. Finally, high speed turbomachinery components are capable of providing high power densities (watts/lb) and specific volumes (watts/cu in.) and by utilizing hydrodynamic type bearings, long life and low noise levels.

In selecting a candidate rotary Rankine cycle system both turbine and rotary expander driven systems were investigated. A discussion of the types of engine configurations considered follows.

3.2.3.2 Engine Configurations

There are several engine configurations that can be initially considered for this application. They are:

- Turbopump-alternator (ac electrical power output)
- Turbopump-generator (dc electrical power output)
- Turbopump (hydraulic power output)
- Turbopump (mechanical power output)
- Rotary expander-pump (hydraulic power output)
- Rotary expander (mechanical power output)

The turbopump configurations are characterized by operation at high speed (i. e., 96,000 rpm). In cases where an electrical output is provided they also permit a wide choice of electrical-to-mechanical or hydraulic energy conversion devices. In addition, they are compatible for use with secondary batteries for electrochemical energy storage. This latter capability permits sizing the system for the average rather than peak pump power requirements. The turbopump can also be employed to supply a hydraulic output to a hydraulic converter from the main cycle pump. This approach would be compatible with a high pressure hydraulic converter system capable of operating on the Rankine cycle working fluid. However, this system does not have the potential for energy storage and must be designed to provide peak power continuously. Finally, the turbopump can provide mechanical output power and through a suitable speed reducer be coupled to a mechanical or hydraulic converter. This configuration is also a peak

power output, dissipative type. Without energy storage capability, these latter two configurations result in requiring a significantly larger heat source than those providing an electrical output with a battery.

An alternate approach is to use a positive displacement rotary expander. This is characterized by low speed (i. e. , 390 to 6000 rpm) operation. This permits direct coupling to a hydraulic pump or a low gear ratio speed reducer capable of providing a mechanical output. These systems also are not capable of energy storage and have all the system shortcomings associated with peak power output, dissipative types.

A preliminary review of these six configurations was made. It became apparent that the turbopump-generator (or alternator) was superior to the turbopump alone in terms of minimum heat source size and system flexibility. Hence, emphasis was placed upon investigating various configurations of this type. The low speed, rotary expander configurations are inherently heavy, mechanically complex, and large in volume. However, because of their capability for direct coupling to a hydraulic or mechanical converter, these configurations were also retained for further analysis. A more detailed discussion of these systems is provided in Section 3.2.3.5.

3.2.3.3 Selection of Working Fluid

There is a wide variety of possible Rankine cycle working fluids that could be considered for this application. However, when one considers the low power level involved and the constraints imposed by body implantation, the number of choices narrows considerably. In the latter category, in addition to the obvious requirements of weight, volume, specific gravity, and heat rejection temperature limits, an additional overriding consideration is overall efficiency. Another important factor is that the working fluid should be impervious for the 10-year life with respect to pyrolytic and radiolytic decomposition.

The most commonly used fluid in the Rankine cycle is steam. However, except for use with the positive displacement rotary expander, steam

is not a very attractive fluid for use with small single-disc turbines. Its very high nozzle efflux velocity for a moderate temperature drop results in impractically high rotor speeds in order to achieve a reasonable level of efficiency. An additional shortcoming of steam is that unless a high degree of superheat is employed, the expansion process can result in entering the wet region of the working fluid and cause turbine erosion. A much higher degree of superheat (and hence high peak cycle temperatures) is also required if any significant cycle performance improvement using regeneration is to be achieved. Despite these shortcomings, steam was retained for use with the rotary expander and for comparison purposes with the turbopump-generator configurations.

The fluids most capable of meeting the essential prerequisites outlined above are broadly categorized as high molecular weight (HMW) working fluids. Fluids having a high molecular weight provide a low nozzle efflux velocity and a comparatively large mass flow rate through the turbine. This enables a satisfactory matching between the nozzle gas exit velocity and the rotor tip speed (velocity ratio) at moderate turbine rotational speeds and results in a higher single-disc vapor turbine efficiency. Another attractive feature of these fluids is that the slope of the saturated vapor line in the temperature-entropy diagram is either vertical or has a positive ds/dT slope. This not only assures that isentropic expansion does not result in "wet" vapor, but in an actual cycle permits the use of efficiency-improving regeneration without the need for excessive superheating in the boiler. Another important requirement is thermal stability at the peak cycle temperature for prolonged periods of time (10 years) and chemical inertness with respect to the materials used in the turbopump-generator. In addition, for this application where the heat source will be medical grade plutonium-238 dioxide, it is important that the threshold values for these working fluids be sufficiently high so that no adverse effects on fluid composition, viscosity, density, or heat transfer characteristics are experienced.

With the above criteria in mind, a review and analysis of the thermodynamic characteristics of approximately 25 of these fluids was conducted.

Based upon this investigation, four fluids were selected for consideration for use with various potential candidate rotary Rankine cycle systems. These, as well as steam, are listed in Table 3.2.3-1. The factors considered in selecting these organic fluids included:

- Reasonable isentropic work available at peak cycle temperatures in the 400 to 750°F range
- Condensing temperatures in the range of 115 to 350°F to be consistent with body heat rejection capabilities and practical heat exchanger designs
- Avoidance of fluids requiring extremely high condenser vacuum to assure adequate pump net positive suction head
- Thermal stability and sufficient imperviousness to 10 year integrated radiation dose levels from the heat source.

At this point, the final selection of a working fluid for this application involves an assessment of the cycle conditions to be utilized and their resultant impact on component and overall system efficiencies. This is discussed in Sections 3.2.3.4 and 3.2.3.5.

3.2.3.4 Cycle Conditions

The selection of cycle operating conditions is contingent upon many factors. These include thermodynamic characteristics of the working fluid, achievement of acceptable Carnot and Rankine cycle efficiencies, and compatibility with system component characteristics and body physiological constraints to insure maximum performance at minimum weight

Table 3.2.3-1. Candidate Rotary Vapor Cycle Working Fluids

Fluid	Molecular Weight
Steam	18
Monsanto CP-34 (Thiophene)	84
Biphenyl	154.2
Freon 11	137.4
Fluorinol 85	87.74

and volume. Since these factors are interrelated, it is usually necessary to iterate several times before arriving at acceptable cycle conditions. In addition to analytical assessments, experience and judgment also play an important role in arriving at a selection.

Based upon the foregoing, the cycle conditions selected for investigation for use with the turbopump-generator configurations are shown in Table 3.2.3-2. The peak cycle temperatures selected for steam are characterized by the fact that all include considerable superheat (from 110° to 300°F). This was done to insure that the steam remained in the superheated or saturated vapor condition after isentropic expansion so there would be no danger of turbine blade erosion caused by impingement of condensed liquid droplets. Condensing pressures were selected to provide acceptable heat rejection temperatures, avoid extremely high condenser vacuum, and result in reasonable Carnot cycle efficiencies. The isentropic work available for all the steam cycle conditions considered is fairly high (261 to 500 Btu/lb). This results in high nozzle spouting velocity ratios, even at high turbine speeds. In addition, at low power output levels, the required flow rates are extremely small. This has the effect of producing very low turbine and pump specific speeds, which in turn leads to low component efficiencies. When considered as a working fluid for rotary expanders, the maximum cycle operating temperatures and pressures must be considerably reduced to minimize thermal conduction losses and thermal differential expansion design complexity.

The cycle conditions for the HMW fluids impose minimal superheat requirements (i. e. , 5° to 30°F). The peak cycle operating temperatures are constrained primarily by the need to assure thermal stability over a period of 10 years. Except in the case of biphenyl, where the thermodynamically imposed high condensing temperature requires a high peak temperature to achieve a reasonable Carnot efficiency, this value was in all other cases kept at 500°F or less. By limiting the HMW fluid cycle temperature to this value, the total decomposition to be expected in 10 years will be less than 0.1% (Figure 3.2.3-1). The HMW working fluids are characterized by considerably lower available isentropic work (17.5 to 175 Btu/lb) and somewhat lower Carnot cycle efficiencies than that obtainable with steam. However, the lower isentropic work requires a higher

Table 3.2.3-2. Comparison of Cycle Conditions for Selected Working Fluids

Working Fluid	Peak Cycle Conditions		Condensing Conditions		Carnot Cycle Efficiency (%)	Isentropic Work Available (Btu/lb)
	Temperature (°F)	Pressure (psia)	Temperature (°F)	Pressure (psia)		
Steam	800	700	115	1.5	54.3	500
	800	700	281	50	41.2	261
	750	500	115	1.5	52.5	462
	750	500	212	14.7	44.2	321
	500	220	153	4	36.2	295
CP-34	500	450	150	9	36.5	70
	500	450	116	4	40.2	80
	430	250	116	4	35.0	69
	400	200	100	3	34.9	70
Biphenyl	750	116.5	330	1.5	34.7	175
Freon 11	390	550	165	67	26.5	19.0
	390	550	120	33	31.8	22.5
Fluorinol 85	470	700	165	15	32.8	57
	470	700	115	4.5	38.2	71

Figure 3.2.3-1. Thermal Decomposition of Typical HMW Rankine Cycle Working Fluids

cycle mass flow rate. For this low power level application, this results in improved component efficiencies since higher specific speeds are obtainable. In addition, lower nozzle exit velocities are obtained which permits achieving higher turbine velocity ratios at reduced rotative speeds. This also increases the turbine efficiency. Finally, since it is possible to use regeneration with these HMW fluids, the overall effect is to be able to achieve higher Rankine cycle efficiencies than with steam, despite the initially lower Carnot cycle efficiencies. For the turbopump-generator, the Rankine cycle efficiency is defined as the net electrical power out divided by the thermal heat input to the working fluid. In the vapor Rankine cycle, the degree of regeneration available, as well as the higher turbine efficiencies achievable, are two of the more significant factors in obtaining higher Rankine cycle efficiencies at lower peak cycle temperatures, than with nonregenerative steam Rankine cycles.

The determination of the actual efficiencies obtainable for the cycle conditions outlined in Table 3.2.3-2 require a detailed analysis of each

system and its components. This also enables one to make assessments regarding system weight, volume, and heat source thermal requirements. The results of these analyses are presented in Section 3.2.3.5.

3.2.3.5 Selection of Candidate Rotary Vapor Cycle System Configuration

In analyzing the various rotary Rankine cycle systems, the major emphasis was placed upon investigating the turbopump-generator configuration. These designs held the highest promise of achieving the required performance within the weight, volume, and heat source thermal inventory constraints. The rotary expander configuration was also analyzed, but consideration in these cases was limited to steam and Freon 11 as working fluids. The approach taken in each case is described below.

Turbopump-Generator Configurations

The classic method employed for estimating the performance of turbomachines involves four similarity parameters. These are specific speed, N_s , specific diameter, D_s , Reynolds number, R_e , and Mach number, M . These are defined respectively as:

$$N_s = \frac{N(V)^{0.5}}{(H_{ad})^{0.75}} \quad (1)$$

$$D_s = \frac{D(H_{ad})^{0.25}}{(V)^{0.5}} \quad (2)$$

$$R_e = \frac{Du\alpha}{\mu} \quad (3)$$

$$M = \frac{v}{C} \quad (4)$$

where

N = turbine rotative speed, rpm

D = rotor diameter, ft

V = volumetric flow rate at the turbine outlet, ft^3/sec

H_{ad} = isentropic head available for work in turbine, ft-lb/lb

u = rotor tip speed, ft/sec

v = velocity at nozzle exit, ft/sec

α = density of fluid at turbine inlet, lb/ft³

μ = viscosity of fluid at turbine inlet, lb/ft-sec

C = sonic velocity at turbine inlet conditions, ft/sec

For Reynolds numbers greater than 10^6 and Mach numbers less than 1 (subsonic flow) which exist for the designs considered, the significant performance parameters became primarily the specific speed and specific diameter. An interesting aspect of the specific diameter, D_s , parameter is that this value represents a critical dimension of the turbomachine, the turbine rotor diameter, and introduces a geometric value into the similarity concept. This is a distinct advantage over use of the turbine velocity ratio (u/v) only, since it offers an opportunity to recognize relations for the optimum turbine geometrical values in terms of D_s . This concept is used to present available turbine performance test data in convenient design diagrams which depict maximum obtainable efficiency together with optimum design geometry.

In utilizing these parameters, engineering judgment must also be exercised since various turbine configurations can be considered. These include:

- Axial Flow – Impulse

- Single stage – partial admission

- Single stage – full admission

- Curtis type – velocity compounded

- Radial Flow – Centripetal

- Impulse blading – approximately 15% reaction

- Radial blading – approximately 50% reaction

For the low power outputs and correspondingly low flow rates involved in this application, the centripetal (inward flow) radial turbine is preferable to the axial flow turbine. This configuration, because of its simple, rugged construction and potentially lower manufacturing cost, becomes

extremely attractive. In addition, as a result of considerable development effort, they are capable of matching or exceeding the performance of axial flow turbines, particularly in the partial admission operating regimes common to low flow rate applications of this type.

With the cycle conditions established in Table 3.2.3-2, it is possible to determine the available isentropic head, H_{ad} . However, in order to size the turbine rotor and confirm its overall performance, it is still necessary to determine the required cycle flow rate and turbine operating speed. The turbine flow rate is related to the work output required from the turbine and is established as follows:

$$W_t = W_p + W_g \quad (5)$$

where

W_t = turbine work output, Btu/hr

W_p = cycle pump work input, Btu/hr

W_g = total generator work input, Btu/hr

The cycle pump work input is determined by

$$W_p = \frac{0.000583 \times \Delta P \times Q}{\eta_p} \quad (6)$$

$$= \frac{0.000583 \times \Delta P \times \frac{\dot{m} \times 7.48}{\rho_l} \times K_1}{\eta_p}$$

where

Q = cycle flow rate, gals/min

$\dot{m}$ = cycle flow rate, lb/min

ΔP = pump pressure rise, lb/in²

ρ_l = density of working fluid at pump inlet, lb/ft³

η_p = pump overall efficiency

K_1 = constant to convert to Btu/hr

and the generator work input (assuming electrochemical energy storage) is expressed by,

$$W_g = \frac{\text{blood pump average power}}{\eta_{pc} \times \eta_{gen}} \quad (7)$$

where

η_{pc} = power conditioning and control system efficiency

η_{gen} = generator efficiency

In addition, since the turbine work output can also be expressed as,

$$W_t = \dot{m} \eta_t H_{ad}$$

where

$\dot{m}$ = cycle flow rate, lb/min

η_t = turbine overall efficiency

Now by substituting these values in Equation (5), and solving for the cycle flow rate, one obtains the following expression,

$$\dot{m} = \frac{W_g}{(\eta_t H_{ad}) - \left(\frac{\Delta P}{\rho_l} K_2 \right)} \quad (8)$$

where

$$K_2 = K_1 \times 0.000583 \times 7.48 = \text{constant}$$

In order to establish the flow rate, at this point it is necessary to initially assume values for the various system component efficiencies, i.e., η_g , η_p , η_t , and η_{pc} . These must later be confirmed by detailed component design analyses, design charts, and when available, test data. An iterative process must be performed until the assumed efficiency values and the detailed design analyses values coincide.

The selection of the turbine operating speed is contingent upon achieving maximum performance from the three combined rotating unit

(CRU) components at minimum weight and volume. However, as was previously discussed, the pump efficiency is less critical than the turbine efficiency in the Rankine cycle when endeavoring to achieve maximum overall system efficiency. In addition, permanent magnet type generator efficiency is more a function of the rotor and stator electromagnetic properties than of speed. Hence, as long as pump cavitation limits are not exceeded and generator rotor windage and bearing losses do not become excessive, the choice of CRU rotative speed is usually based upon maximizing turbine efficiency. By utilizing Equation (1), the speed can be assumed and with the cycle conditions known, the specific speed, N_s , of the turbine can be determined. This value can be used to obtain a first approximation to the turbine efficiency by employing a design chart of the type shown on Figure 3.2.3-2. The specific diameters, D_s , can also be obtained from this chart; using the chart and Equation (2), the turbine rotor diameter can be calculated. If the values determined for efficiency and diameter are acceptable, a further check on turbine efficiency can be obtained by calculating the turbine velocity ratio, U_1/C_o , where,

U_1 = turbine peripheral velocity, ft/sec

C_o = spouting velocity from nozzle, ft/sec

The spouting velocity from the nozzle is given by,

$$C_o = \psi_N \sqrt{2gH_{ad}} \quad (9)$$

where

ψ_N = nozzle flow coefficient

With the velocity ratio known, the turbine diagram efficiency can be obtained from a design chart as shown on Figure 3.4.3-3. Final confirmation of turbine efficiency must ultimately be accomplished by a detailed, rigorous analysis of thermodynamic, aerodynamic, and heat transfer characteristics of the turbine design including all pertinent losses. However, for preliminary design purposes, it is usually sufficient to rely upon

Figure 3.2.3-2. Design Chart for Partial Admission Radial-Impulse Turbines

Figure 3.2.3-3. Turbine Diagram Efficiency vs Velocity Ratio

the design charts of Figures 3.2.3-2 and 3.2.3-3 which have been based upon a compilation of test data for the particular turbine type selected, i. e. , partial admission radial impulse turbine.

Using the above approach and investigating all the cycle conditions outlined in Table 3.2.3-2, it was possible to determine and select the preferred cycle operating mode for each of the working fluids considered. The results of these analyses are shown on Table 3.2.3-3. In addition to the varying cycle conditions, it should be noted that the required turbine operating speed for maximum turbine efficiency varied from 72,000 to 144,000 rpm. By referring to Table 3.2.3-1, one can see that the turbine speed is related to the working fluid molecular weight, with the lower values requiring the higher operating speeds. Also, for the cycle conditions selected, only the thermodynamic characteristics of CP-34 and biphenyl permitted the use of regeneration. The overall thermal converter performance improvement obtainable with regeneration is best seen by a comparison between steam and CP-34 working fluids. The nonregenerative steam Rankine cycle is shown on Figure 3.2.3-4 and the regenerative CP-34 cycle is shown on Figure 3.2.3-5. From Table 3.2.3-3, it can be seen that the Carnot efficiencies for these two cycles are nearly equal (i. e. , steam = 0.362 versus CP-34 = 0.35). However, because of the use of regeneration and the higher turbine efficiency achievable with CP-34, the Rankine cycle efficiency (work out $\div$ heat input) for CP-34 is considerably higher than for steam (i. e. , CP-34 = 0.201 versus steam = 0.135).

A review of the results of the analyses summarized in Table 3.2.3-3 led to the selection of CP-34 (thiophene) as the preferred working fluid for the turbopump-generator rotary Rankine cycle system. It not only demonstrated the highest overall thermal converter efficiency (i. e. , 16.0%) but it also achieved this at moderate peak cycle temperatures and rotative speeds. The latter two characteristics are important factors in achieving long life.

Rotary-Expander Configurations

The similarity laws for positive displacement rotary machines differ somewhat from turbomachines. However, it has been demonstrated that their maximum obtainable efficiency is also a unique function of the specific speed and specific diameter as previously defined for turbomachines.

Table 3.2.3-3. Comparison of Turbopump-Generator Rankine Cycle Systems

Working Fluid	Steam	CP-34	Biphenyl	Freon 11	Fluorinol 85
Turbine Speed, rpm	144,000	96,000	72,000	72,000	96,000
Turbine Inlet Temperature, °F	500	430	750	390	470
Turbine Inlet Pressure, psia	220	250	166.5	550	700
Superheat at Turbine Inlet, °F	110	5	0	20	30
Condensing Pressure, psia	4	4	1.5	33	4.5
Condensing Temperature, °F	153	116	340	120	115
Adiabatic Head, Btu/lb	295	69	63	22.5	71
Adiabatic Head, ft	228,000	53,600	49,000	17,500	55,200
Flow Rate, lb/min	0.0033	0.0134	0.0149	0.0347	0.014
Specific Speed	0.95	1.7	2.75	1.65	1.89
Turbine Wheel Diameter, inch	1.25	1.125	1.60	1.00	1.00
Velocity Ratio, U_1/C_o	0.21	0.254	0.290	0.3	0.25
Turbine Efficiency, %, Diagram	62	70	75	76	70
Turbine Efficiency, Overall	45	58	63	64	58
Carnot Efficiency, %	36.2	35.0	34.5	31.8	38.2
Rankine Cycle Efficiency, %	13.5	20.7*	19.4*	14.8	18.8
Thermal Converter Heat Input, watts	60	60	60	60	60
Generator Electrical Output, watts	4.26	9.6	9.17	8.25	8.85
Overall Thermal Converter Efficiency, %	7.2	16.0	15.3	13.75	14.75

*With regeneration

COMPARISON OF STEAM AND CP34 THERMODYNAMIC CYCLE DIAGRAMS

Figure 3.2.3-4. Steam (Nonregenerative) Rankine Cycle

1-2 TURBINE EXPANSION	5-6 REGENERATION (LIQUID SIDE)
2-3 REGENERATION (VAPOR SIDE)	6-7 BOILER (LIQUID PHASE)
3-4 CONDENSING	7-1 BOILER (VAPORIZATION PHASE)
4-5 PUMP WORK	

Figure 3.2.3-5. CP-34 Regenerative Rankine Cycle

However, the design charts to be used for relating N_s , D_s and expander efficiency must be based upon performance data obtained on these types of machines. In addition, there is a wide variety of positive displacement type machines. These include:

- Roots type
- Eccentric rotor (Westinghouse type)
- Heli-rotor (Lysholm-Fairchild Hiller type)
- Modified Wankel type

Detailed information on the different loss coefficients in rotary displacement machines is rather limited. Hence, a comprehensive analysis of these machines for a number of working fluids over a wide range of cycle conditions, as was done with turbomachines, was not considered practical. In addition, since these are basically low speed machines operating in the 390 to 24,000 rpm regime, they are not compatible with the fairly high speeds required (50,000 to 150,000 rpm) to obtain lightweight, small volume, permanent magnet generators. Thus, their output power must be coupled to a hydraulic pump and they must be operated in a nonmodulated fashion without energy storage.

Because of the foregoing, only two configurations were investigated in detail. These were an eccentric rotor steam expander and a heli-rotor expander operating on Freon 11. The cycle conditions were selected to be compatible with the temperature and pressure ratio limitations of the expander. Only the mechanical output configuration was considered since it was established during this study that a low pressure hydraulic output would be inconsistent with achieving an acceptable minimum weight and volume hydraulic converter. A comparison of the performance of these two systems is shown in Table 3.2.3-4. As can be seen from this table, the necessity for these systems to provide peak pulsatile power continuously imposes a severe penalty on the heat source thermal input. This occurs despite the assumption of expander efficiencies as high as 65% and utilizing a mechanical actuator conversion efficiency of 55%. Therefore, it is concluded that these nonmodulated, rotary expander systems without energy storage are not capable of staying within maximum allowable heat source constraints.

Table 3.2.3-4. Comparison of Rotary Expander-Mechanical Output Rankine Cycle Systems

Expander Type	Eccentric Rotor	Heli-Rotor	Heli-Rotor
Working Fluid	Steam	Freon 11	Freon 11
Expander Speed, rpm	390	24,000	24,000
Expander Inlet Temperature, °F	500	390	280
Expander Inlet Pressure, psia	218	550	180
Superheat at Expander Inlet, °F	111	20	30
Condensing Pressure, psia	3.5	67	30
Condensing Temperature, °F	147	165	115
Adiabatic Head, Btu/lb	299	19	15.5
Adiabatic Head, ft	233,000	14,800	12,100
Flow Rate, lb/min	0.00475	0.0955	0.100
Specific Speed	0.0035	0.575	1.05
Expander Outside Diameter, inch	1.3	0.300	0.500
Expander Efficiency, %, (estimated)	0.65	0.65	0.65
Input Power to Mechanical Actuator, watts	16.2	16.2	16.2
Carnot Efficiency, %	36.8	26.5	22.3
Rankine Cycle Efficiency, %	16.8	14.15	10.9
Thermal Converter Heat Input, watts	105	162	164
Overall Thermal Converter Efficiency, %	15.4	10.0	9.9

Configuration Selection

Based upon the foregoing evaluation, the turbopump-generator was selected in preference to the rotary expander. The cycle, using CP-34 as a working fluid, resulted in the highest overall system efficiency. Hence, this configuration was selected as the candidate rotary vapor cycle system.

3.2.3.6 Component Design

The components required for implementation of the CP-34 rotary vapor cycle system include the following:

- Inward-flow radial impulse turbine
- Centrifugal pump
- Permanent magnet, brushless generator
- Boiler
- Condenser
- Regenerator
- Secondary battery
- Speed controls

These components were chosen on the basis of achieving maximum performance for the required cycle conditions. In addition, emphasis was placed on achieving long operating life. By selecting component elements such as noncontacting vapor bearings, operating at very low stress levels, and moderate peak cycle temperatures, adequate design margins can be achieved to meet the 10-year goal.

The main element comprising the rotary vapor cycle system is the turbopump-generator or simply turbogenerator unit (TGU). The TGU is shown in Figure 3.2.3-6. It consists of a radial, inward-flow, impulse-type turbine, a centrifugal pump, and a permanent magnet dc generator. They are all mounted on a single shaft rotating at 96,000 rpm. The maximum peripheral velocity, which occurs at the turbine outside diameter, is only 470 ft/sec. This represents an extremely conservative design since the typical rotor peripheral velocities for many turbomachinery applications are two to four times greater and occur at peak cycle temperatures

Figure 3.2.3-6. Turbogenerator-Pump Unit, Rotary Rankine Cycle

that are considerably higher. This is shown in Figure 3.2.3-7 for various terrestrial and aircraft applications. Since all the rotating components are mounted on a common shaft, this heat engine consists of only one moving part. This feature, when coupled with the use of self-energizing, compliant-foil, hydrodynamic gas bearings results in a design with high inherent reliability and long life. The characteristics of the components comprising the TGU, as well as the overall system components (excluding the heat source), are described below.

Figure 3.2.3-7. Comparison of Rotor Peripheral Velocities and Temperatures for Various Turbomachinery Applications (Shaded areas indicate range of rotor velocities)

Turbine

The turbine selected for this system was a radial, inward-flow, impulse type. This turbine will operate at 96,000 rpm and will be 1.125 inches in outside diameter. When operating at the cycle conditions given in Table 3.2.3-3 for CP-34, this will result in a velocity ratio of 0.254, and using a 41-watt thermal heat source, a power output of 6.25 watts electric at a specific speed of 1.6. Under these conditions, this turbine will have a diagram efficiency of 70% (Figure 3.2.3-3) and an overall efficiency of 55% (Figure 3.2.3-2). Because of the lower flow rate required at this lower output power level, the overall system efficiency will be 15.7%. The turbine will be a partial admission type (approximately 6%) and to assure achievement of maximum efficiency will employ approximately 10% reaction in the blading.

The achievement of this level of performance in the turbine is contingent upon an accurate assessment of the various system performance degradation factors. Analyses were performed to confirm the validity of the estimated loss factors and to determine if sufficient design margins exist for this low power level turbopump-generator. The three areas of concern were bearing losses, thermal losses, and leakage losses. These are treated separately in the following discussion.

Bearing Design and Losses. The vapor bearings under consideration for this design are a self-energizing, compliant, multifoil, hydrodynamic type. These bearings utilize full fluid film lubrication. In a closed, hermetically sealed system of the type envisioned for this application, there should be no difficulty in achieving the 10-year (87,500 hours) operating life. They are state of the art, having been used on the DC-10 Air Cycle System (53,000 rpm), the NASA/Lewis Brayton cycle turbo-alternator (48,000 to 64,000 rpm) and a high speed (180,000 to 220,000 rpm), 7.8 watt, reverse Brayton refrigeration cycle. Because of this, the use of these type bearings constitutes a low development risk. During operation there is no physical contact between the shaft and the bearings and the lubricating film is maintained hydrodynamically rather than hydrostatically. The general arrangement of these bearings is as shown in Figure 3.2.3-8. These are overlapping foil conical bearings which are capable of sustaining both radial and thrust loads. These conical bearings are composed of six thin Teflon-coated spring-metal blades or foils. The foils are equally spaced within a conical bushing, and each foil extends or wraps 135 degrees within the bushing. Pins are spot-welded to the end of the spring-steel blades and inserted into the slots by a press fit to retain these spring-steel foils. A 60-degree cone angle is normally used. The steel foils are fabricated from 0.0007-inch thick heat-treated "Havar", a spring material available from the Elgin Watch Company and coated with 0.002-inch thick Teflon using a DuPont process. The steel foils are formed with a punch and die.

In operation, the spring materials provide stiffness in the bearings by resisting the hydrodynamic pressure developed by the rotation of the shaft. The springs exert pressure on the shaft while it is stationary. At start-up, the Teflon coating functions to reduce friction. The shaft

Figure 3.2.3-8. Self-Energizing, Compliant, Multifoil, Hydrodynamic Type Vapor Bearing (Combined Radial and Thrust)

becomes hydrodynamically supported at approximately 40,000 rpm. Only a minute vapor film thickness (perhaps a few millionths of an inch) is required to support the shaft. Because the hydrodynamic film pressure creates the clearance between the bearing foils and the shaft, the necessity of machining to very close tolerances is eliminated. In addition, if an externally applied "g" load is experienced, this decreases the operating clearance on one side of the bearing and increases it on the opposite side. This increases the hydrodynamic pressure on the decreased clearance side thus providing a self-generated restoring force and stable bearing operation.

An additional consideration in the use of these bearings is to confirm their satisfactory operation, including the magnitude of their losses, utilizing CP-34 (thiophene) vapor. Initial estimates for these bearings were taken as 5%, which for a nominal turbogenerator output of 9.0 watts would equal 0.45 watt. This is the total loss for both the radial and thrust bearings. To verify this value, an analysis was conducted. The dimensions of the bearings are as shown in Figure 3.2.3-9. The analysis was based upon the data listed in Table 3.2.3-5.

Figure 3.2.3-9. Dimensions for Conical Combined Radial and Thrust Bearings

Table 3.2.3-5. Conical Gas Bearing Design Data

Total Weight of Rotating Elements	0.0432 lb
Bearing Loads:	
3g in thrust direction	0.1296 lb
2g in radial direction	0.0864 lb
Working Fluid	CP-34 (Thiophene)
● Temperature	300°F
● Pressure	15 psia
● Viscosity (at 300°F)	$0.4068 \times 10^{-8} \text{ lb-sec/in}^2$
Shaft Rotative Speed	96,000 rpm

For the above conditions, the load factor (Figure 3.2.3-10) was determined to be 0.296, and the corresponding compressibility factor is 460. The curve shown in Figure 3.2.3-10 is based upon test data obtained on four different size conical bearings over a speed range from 12,000 to 102,000 rpm. Under these conditions, the bearing film thickness, h , was

Figure 3.2.3-10. Normalized Load as a Function of Bearing Compressibility Number

calculated to be 2.38×10^{-5} inch. The torque, T , was now determined using the following analysis:

$$\begin{aligned}
 T &= \int_{r_1}^{r_o} \lambda 2\pi r^2 dr \\
 &= \int_{r_1}^{r_o} \frac{\mu\omega r}{h} 2\pi r^2 dr
 \end{aligned}$$

integrating and evaluating within the limits,

$$T = \frac{\pi \mu \omega}{2h} \left[r_o^4 - r_1^4 \right]$$

Now, by substituting the above data, the torque was determined to be 4.05×10^{-4} in-lb.

The power into the bearings under these conditions becomes 0.46 watt (6.16×10^{-4} hp) which compares very well with the value of 0.45 watt originally assumed. This value is on the conservative side since it is based upon a sustained 3g acceleration level. If the normal 1g load is taken as the steady-state bearing load, the design margin for this calculated bearing loss is 1.4.

Finally, it should be noted that this bearing is operating on superheated (approximately 125°F of superheat) vapor. The power loss in the bearing will maintain or slightly increase this superheat condition. Hence, no problem from two-phase fluid operation is anticipated.

In summary, the performance estimates made for the gas bearings in this application are conservative. In addition, because the technology upon which they are based has been proven on earlier applications, no difficulties should be experienced in their implementation.

Thermal Losses. An additional performance degradation factor which must be taken into account is the thermal losses in the turbine housing. Under steady-state conditions, the heat loss from the turbine inlet scroll, which is at 430°F, to the turbine outlet, which is at 240°F, is the most significant factor. At the design flow rate (for a 6.25 w_e output) of 0.01 lb/min, the isentropic work available is 41.5 Btu/hr. The thermal loss caused by conduction through the turbine housing was calculated to be 0.171 Btu/hr (i. e., 0.05 watt). This constitutes a loss of only 0.41%. The value used for this loss in confirming the overall turbine efficiency was assumed to be 2%. Hence, a conservative design margin of five exists for the proposed turbopump-generator configuration.

Leakage Losses. The turbine leakage losses for this design have been estimated at 2%. This is readily achieved by virtue of the selection of a shrouded, radial, impulse-type turbine. The static pressure differential between the turbine blading inlet and outlet passes is negligible since the overall pressure drop in an impulse type turbine occurs in the turbine inlet nozzle. A pressure differential of 1.125 lb/in^2 would be required to produce a flow rate equal to this leakage loss. An analysis

was conducted and the pressure differential across the turbine clearance space was calculated to be only 0.025 lb/in². Since the leakage flow is proportional to the square root of the pressure differential, this is equivalent to a leakage loss of only 0.33%. The additional loss is the amount of working fluid required for bearing heat removal. This value was calculated to be approximately 1.7% of the turbine flow. Thus, the 2% turbine leakage factor estimated for this design adequately accounts for the anticipated losses in this area.

Turbine Disc Stresses. Another important factor in the turbine design is the factor of safety associated with the disc stresses.

The currently envisioned turbogenerator for the rotary vapor cycle system operates at 96,000 rpm. It uses a turbine wheel with an outside diameter of 1.125 inches. This results in a tip speed of 470 ft/sec. If a uniform thickness disc at an average temperature of 300°F is conservatively assumed, the maximum allowable tip speed based upon the yield strength of the material is 1750 ft/sec (Figure 3.2.3-11). This represents a design margin of 3.7 and is a conservative first approximation.

Experiments have shown, however, that a disc will burst at speeds at which the centrifugal force of half the disc induces an equivalent constant stress to occur across the section equal to the ultimate tensile strength of the material. Thus, in Figure 3.2.3-12, if the position of the center of gravity of the half disc is obtained at some radius, r inches, and w is the weight in pounds of the half disc, A is the square inches of net cross-sectional area, and if σ_y , the yield stress, is again taken to be on the conservative side, then,

$$A\sigma_y = \frac{w}{12g} \omega^2 r$$

or the speed of rotation for bursting would be

$$N = 187.6 \sqrt{\frac{A\sigma_y}{wr}}, \text{ rpm}$$

Figure 3.2.3-11. Turbine Disc Stresses as a Function of Tip Speed and Temperature

from which an estimate of the maximum speed for any disc can be obtained. Using this equation and assuming the use of 7075-T6 aluminum at 325°F average temperature (see Table 3.2.3-6 for yield and ultimate tensile strength), the estimated speed of rotation at bursting would be approximately 139,000 rpm. This represents a factor of safety of 1.45 for the design speed of 96,000 rpm. If AISI 301 stainless steel is considered for the turbine disc, the estimated burst speed would be 165,000 rpm and the design margin increases to 1.72.

The above values are conservative since a more detailed stress analysis would show even higher factors of safety. The actual stress levels and distribution for the turbine wheel under consideration in this program are more closely represented by the values shown on Figure 3.2.3-13. The maximum average tangential stress for a wheel operating at 20% overspeed (120,000 rpm) is 11,250 psi. For 6061-T6,

Figure 3.2.3-12
Stress Distribution in Rotating
Disc

at an average turbine wheel temperature of 325°F, this represents a factor of safety of 2.2 based upon yield strength, and 2.75 based upon the ultimate tensile strength. In summary, the turbine wheel stresses at speeds up to 120,000 rpm are fairly modest. Hence, operation of the turbine wheel at 96,000 rpm over a 10-year period should pose no problem with respect to life or reliability.

Rotor Critical Speed and Dynamics. The critical speed of the rotor in bending may be conservatively estimated by treating the rotor as a uniform beam with rigid supports at the bearings (hinged-hinged beam). The equation for critical speed reduces to

$$f_n = \frac{\pi d}{8\ell^2} \sqrt{\frac{Eg}{\rho}}$$

where

f_n = critical frequency, rev/sec

d = shaft diameter, in.

e = shaft modulus of elasticity, lb/in² (30 x 10⁶ for stainless steel)

g = gravitational constant, 386 in/sec²

ρ = rotor specific weight, lb/in³ (0.283 for steel)

ℓ = length of beam or bearing span, in.

Table 3.2.3-6. Material Property Data

Type	Yield Strength, psi		Ultimate Tensile Strength, psi		Aging Temperature (°F)
	(RT)	(325°F)	(RT)	(325°F)	
<u>Aluminum</u>					
6061-T6	35,000	24,500	42,000	31,000	325
7075-T6	65,000	33,800	75,000	39,000	250
7075-T73	60,000	31,500	70,000	36,300	250
<u>Stainless Steel</u>					
AISI 304	30,000	—	85,000 to 150,000	—	—
AISI 301 (Sheet)	40,000 to 160,000	—	105,000 to 185,000	—	—
AISI 301 (Rod)					
(0.25 Hard)	75,000	—	125,000	—	—
(0.50 Hard)	110,000	—	150,000	—	—
AISI 303	35,000	—	90,000	—	—
AISI 302	35,000	—	125,000	—	—
AISI 431	135,000	—	175,000	—	—

Figure 3.2.3-13. Estimate of Tangential and Radial Stresses in Rotary Rankine System Radial Turbine

If a minimum shaft diameter of 0.156 inch is conservatively assumed and a bearing span of 0.75 inch is taken, the first critical speed of the rotor becomes 1,322,000 rpm. This represents a design margin of 13.8 on the nominal operating speed of 96,000 rpm. Consequently, no problem is anticipated with respect to operation near a shaft bending critical speed.

An analysis was also made of the turbopump-generator gyroscopic moment. If the shaft axis is mounted horizontally within the body which is assumed to rotate at the rate of 1 radian/sec (9.55 rpm), the gyroscopic moment is only 0.57 in-lb. If the shaft axis is mounted vertically, as is contemplated, the gyroscopic moment would become negligible. In either case, the gyroscopic moment will have negligible effect on the implanted heat engine.

Pump

The pump to be utilized for this application will be of the centrifugal type. In order to develop the required head of 236 psi (523 ft) a pump outside diameter of 0.625 inch is required. This is based upon the determination that the required peripheral velocity at the impeller outside diameter is given by

$$U_2 = \frac{2g\Delta H}{\psi}$$

where

U_2 = peripheral velocity at impeller outside diameter, ft/sec

ΔH = pump pressure rise, ft (523 ft)

ψ = head coefficient, (0.70 estimated)

g = gravitational constant, 32.2 ft/sec²

The impeller outside diameter can then be determined from the following relationships.

$$d_2 = \frac{720 \times U_2}{\pi N}$$

where

d_2 = impeller outside diameter, in.

N = pump rotative speed, rpm (96,000 rpm)

The specific speed of this pump can be determined by an equation similar to the turbine, namely,

$$N_s = \frac{N(V)^{0.5}}{(H)^{0.75}}$$

where H , represents the head actually produced by the pump, and the other parameters are as defined in Section 3.2.3.5. This gives rise to a specific speed of 1.4. The estimated performance for this pump is shown in

Figure 3.2.3-14. The overall pump efficiency is estimated to be 40%. The overall pump efficiency is the product of the hydraulic, volumetric (leakage loss) and mechanical (disc and seal friction) efficiencies. This is represented by:

$$\begin{aligned}\eta_{\text{pump (overall)}} &= \eta_{\text{hydraulic}} \times \eta_{\text{volumetric}} \times \eta_{\text{mechanical}} \\ &= 0.5 \times 0.9 \times 0.9 \\ &= 40.5\%\end{aligned}$$

These are readily achievable values for this size pump at its design specific speed. The use of labyrinth seals at the pump inlet and both a dynamic (noncontacting) seal and vented labyrinth seal at the back of the pump impeller are envisioned for this application.

The net positive suction head requirement for this pump is a function of the achievable suction specific speed. Using CP-34 and a pump inlet inducer, a suction specific speed of 20,000 (on a gpm basis) should be readily achievable. Under these circumstances less than 1 degree of sub-cooling is required to meet the net positive suction head requirements of this pump. The condenser design for this application provides in excess of

Figure 3.2.3-14
CP-34 Vapor Rankine Cycle Pump
Performance

this amount of subcooling. Hence, no problem from pump cavitation is anticipated in this design.

Generator

The generator proposed for this application is an ironless permanent magnet type. This design avoids the potential problems of earlier magnetic stator permanent-magnet designs, while providing an electrical efficiency of 98 percent. The stator windings of the ironless stator generator are similar to those of more conventional machines. After winding, the coils are hand-formed so that the windings are compactly located near the inner diameter of the stator. Mechanical support of the stator is provided by encapsulation of the finished windings in a suitable non-magnetic, nonconductive material. Epoxy compounds are excellent for these purposes.

The magnetic rotor material used in this machine is platinum-cobalt alloy. The magnet is used in a hollow cylindrical configuration. End pieces are installed at either end of the magnet cylinder to complete the rotor shaft. The end pieces may be of any adequate structural material with reasonable magnetic permeability. When compared with other materials capable of being strongly magnetized, this magnet material displays excellent mechanical characteristics. Yield strengths from 90,000 to 170,000 psi are obtainable with the platinum-cobalt alloy.

The major electrical losses in this generator are eddy current losses and I^2R losses. The ironless stator design virtually eliminates core loss. In fact, the electrical losses are so small, high efficiency can be maintained over a wide load range. For example, at a 7.8 watt output the total electrical loss is only 0.233 watt, resulting in an electrical efficiency of 0.971.

The generator incorporates a diode chip rectifier assembly, to convert from ac to dc power, whose design is primarily determined by its thermal environment. The temperature of the diode chip itself must be kept within acceptable limits. The dc voltage requirements affect the rectifier design. Should ripple requirements dictate a lower level than that obtainable from a simple, single-phase, full-wave bridge, the use of a filter or a three-phase system can be considered.

The overall electrical generator performance and design criteria are shown in Figure 3.2.3-15. At an output of 6.25 watts electric, the overall efficiency including electrical, windage, and bearing losses is 0.90. The generator has been designed for a nominal voltage of 24 volts but at no significant penalty in either efficiency or weight, can be modified to operate at any intermediate voltage level from 15 to 28 volts. This permits a high degree of flexibility in integrating the electrical power output with the requirements of the electromechanical actuator and its electronic controls.

Figure 3.2.3-15. Rotary Rankine Cycle System
Electrical Generator Performance
and Design Criteria

Heat Exchangers

As indicated in Figure 3.2.3-16, the rotary vapor cycle system will use three heat exchangers, i. e., boiler, condenser, and regenerator. Both the boiler and condenser involve two-phase flow. This increases the complexity of the analyses with regard to heat transfer and pressure drop. For example, the boiler includes a preheater section to increase the working fluid temperature to the saturated liquid level; a two-phase boiling section where such phenomena as slug flow and incipient, nucleate, annular, film, and spheroidal boiling must be taken into consideration; and a superheater section to raise the vapor temperature to 10° F above saturated conditions. Similarly, the condenser must be configured to handle the incoming vapor, the two-phase fluid during the conversion of the working fluid from a saturated vapor to a saturated liquid, and, finally, to further cool the liquid to 1° to 2° F of subcooling to assure providing adequate net positive suction head at the pump inlet. By comparison, the regenerator is fairly simple to analyze since the fluids on the liquid and vapor sides remain in one phase during their flow through this heat exchanger. The overall design criteria for these heat exchangers is given in Table 3.2.3-7. A brief description of these components for the rotary vapor cycle system follows.

Figure 3.2.3-16. Rotary Vapor Cycle System with Regeneration and Electrical Power Output (RBE)

Table 3.2.3-7. Heat Exchanger Design Criteria
(CP34 Working Fluid)

Component	Length (in.)	Diameter (in.)	Pressure Drop (psi)	Weight (lb)
Preheater	9.0	0.0625 ID	Negligible	0.01338
Boiler (10 deg superheat)	35.0	0.0625 ID	~ 1.0	0.0431
Regenerator	4.0	-	0.114	0.0149
● Vapor Side		0.105 ID 0.145 OD		
● Liquid Side		0.045 ID 0.085 OD		
Condenser	30.0	0.0625 ID		0.06
Total Weight				0.1314 lb. (0.0595 kg)

Boiler. A through-flow single-tube cylindrical boiler was selected as an attractive design combination of low weight and small volume. The tube boiler is wrapped around a cylindrical housing which is grooved in a spiral fashion. The tube is brazed into this groove to minimize thermal resistance. The isotope heat source and high temperature battery are located within this cylindrical housing as shown in Figure 3.2.3-17. This arrangement also permits the substitution of an electrical source during the earlier phases of bench model converter testing.

This once-through boiler design receives subcooled liquid and delivers superheated vapor (10 degrees) to the turbine with one pass through a continuous tube. In the selected boiler design the heat transfer mechanism from the isotope heat source to the working fluid (CP-34) is composed of conduction from the heat source to the tube wall and convection from the tube wall to the liquid. The assumption was made that the heat transfer efficiency from the heat source to the tube wall is 90% and since the pump speed is 96,000 rpm, the flow is continuous.

Figure 3.2.3-17. Rotary Vapor Cycle Heat Engine

Because the fluid enters the boiler in a subcooled state, the fluid must be preheated to the saturated liquid temperature before boiling will commence.

Since the flow rate is so small and tube sizes must be practical, the fluid flow in the preheater will be in a laminar regime. A 0.0625-inch ID tube was selected as the minimum practical size and the preheater length was calculated by use of the laminar flow equation

$$\frac{hD}{k} = 1.75 \left(\frac{W C_p}{kL} \right)^{1/3}$$

where

h = heat transfer coefficient

D = tube diameter

k = thermal conductivity

W = fluid flow rate

C_p = specific heat

L = tube length

and the convection heat transfer equation

$$Q = h A_s \Delta T$$

where

Q = heat transfer rate

A_s = surface area

ΔT = temperature difference between fluid and tube wall

All units in these and subsequent formulae unless otherwise noted are in pounds, feet, hours, Btu, and $^{\circ}\text{F}$.

The length of the tube was determined to be 9.0 inches which should prove to be conservative and allow sufficient length for the preheater since the use of other laminar flow equations would result in shorter lengths.

The required length for the boiling section, again assuming a 0.0625 inch ID was calculated by two different methods to gain insight into the degree of possible differences from two approaches.

The first approach assumes nucleate boiling followed by a dry wall boiling section that occurs at approximately 70 to 80% quality.

In the nucleate boiling regime for the working fluid, the required length can be determined from

$$\frac{Q}{A_s} = \frac{K k_l C_{Pl} \rho_l^2}{\sigma T_{SAT} (\rho_l - \rho_v)} (\Delta T_{SAT})^{1.3}$$

where

$$K = 0.218$$

T_{SAT} = Saturation temperature of fluid

σ = surface tension

$\Delta T_{SAT} = T_{Wall} - T_{Saturation}$ = Wall superheat

ρ = density of working fluid

subscripts

l = liquid

v = vapor

All other nomenclature are as previously described.

In the dry wall region the length is determined by use of the Dittus-Boelter equation using the vapor properties:

$$\frac{hD}{k} = 0.023 (Re)^{0.8} \left(\frac{C_p \mu}{k} \right)^{0.4}$$

where

Re = Reynolds number

μ = viscosity

and the normal convection heat transfer equation. All other nomenclature are as previously described.

Evaluation of these formulae resulted in a boiler length of 29.0 inches.

The second method was based on J. C. Chen's formulation of an additive mechanism of micro- and macroconvective heat transfer to represent boiling heat transfer in convective flow. Considering the macroconvective mechanism, a modification of the Dittus-Boelter equation was used to determine the heat transfer coefficient:

$$h_{mac} = 0.023 (Re_l)^{0.8} (Pr_l)^{0.4} (k_l/d) F$$

where

h_{mac} = macroconvective heat transfer coefficient

Pr = Prandtl number

with F as the ratio of the two-phase Reynolds number to the liquid Reynolds number and assumed to be a function of the Martinelli parameter X_{TT} which has been verified by experimental data. The microconvective coefficient was based on the Forster and Zuber analysis which results in the following equation for convective boiling:

$$h_{\text{mic}} = 0.00122 \frac{k_{\ell}^{0.79} C_{P\ell}^{0.45} \rho_{\ell}^{0.49} g^{0.25}}{\sigma^{0.5} \mu_{\ell}^{0.29} \lambda^{0.24} \rho_v^{0.24}} (\Delta T)^{0.24} (\Delta P)^{0.75} S$$

where

h_{mic} = microconvective heat transfer coefficient

g = acceleration caused by gravity

λ = latent heat of vaporization

where S is a suppression factor based on the liquid Reynolds number and F from the preceding equation. All other nomenclature are the same as previously described.

These two heat transfer coefficients are then additive to determine the total heat transfer coefficient,

$$h_T = h_{\text{mic}} + h_{\text{mac}}$$

Because this heat transfer equation was based on annular or annular-mist flow in the quality range of 1 to 70%, the length in this range was determined by the above heat transfer equation; in the 70 to 100% quality range, the Dittus-Boelter equation was used as in the previous approach.

The total length was then determined to be 23 inches. The 29-inch-length calculated from the first approach was therefore assumed for the boiler design and an additional 6 inches was added to insure achieving 10 degree superheat, since correlation of this type can vary as much as 20%.

The pressure drop was determined to be ~1.0 psi by use of the Martinelli and Nelson parameter.

Experimentation is normally required to gather data for final design verification. The low working fluid flow rate for the rotary vapor system may require additional design analyses since the normal boiling formulae may have to be modified in this range. The final detailed design must also consider the possibility of a slug flow phenomenon in the boiler because of this low flow. However, a design to account for this type of flow would only increase the weight by an insignificant amount and should not affect the overall design of the system.

Condenser

The condenser has been designed to reject heat to the system package walls where it is then rejected to the body fluids and tissue. The maximum allowable package wall temperature was set at 107.6° F (42° C) with a maximum heat rejection surface area of 0.07 watt/cm². The required heat rejection rate of the condenser is 29.5 watts and the required heat transfer coefficient from the vapor to the package wall was calculated by:

$$h_m = \frac{0.065 \sqrt{\frac{\rho_l}{\rho_v} \frac{f}{2} C_p G_m}}{\sqrt{\frac{C_p \mu}{k}}}$$

where

h_m = mean coefficient of heat transfer

ρ_l = liquid density

ρ_v = vapor density

f = friction factor

C_p = specific heat

G_m = 0.58 of working fluid flow rate/flow area

μ = viscosity

k = thermal conductivity

using the liquid properties of the working fluid. The condenser size was determined by use of the convective heat transfer equation,

$$Q = h_m A_s \Delta T$$

where

Q = heat transfer rate

A_s = surface area

ΔT = temperature difference between fluid and tube walls

and a tube wall effectiveness factor of 0.7, to be 30 inches long with a 0.065-inch ID and a 0.020-inch wall. In order to reject the heat uniformly over the required package surface area, the condenser tube must be finned.

Regenerator

The regenerator for this system is a conventional vapor-to-liquid concentric tube counterflow design. The liquid recovers heat from the turbine discharge vapor, that would otherwise be rejected to the package wall, thereby reducing the rejected heat load and increasing the overall system efficiency.

In this design, the liquid from the pump on the TGU is brought into a tube where it picks up heat through the tube wall from the vapor which is flowing in the annulus of the outer tube. The sizing of the regenerator considered the requirements of a low pressure drop on the vapor side to negate problems with the centrifugal pump. Because of the low flow rate, the pressure drop on the liquid side will be negligible.

The design consists of a 0.045-inch ID inner tube (fluid side) with a 0.020 inch wall and an outer tube with an ID of 0.105 and an OD of 0.145 inch. The length is 4.0 inches.

Battery

The amount of electrochemical energy storage that would be required to meet the current load profile is 12.5 watt-hours. This requirement is based upon meeting the maximum average blood pump requirements and is a function of the overall electromechanical actuator efficiency. Two additional requirements are that the battery must be rechargeable (secondary type) and have a life of at least 3650 cycles (assuming daily recharging over a 10-year period). Finally, the energy density (w-hr/lb) and the specific volume (w-hr/cc) must be high enough to remain within allowable weight and volume constraints.

Factors affecting secondary battery performance include discharge rate, charging rate, depth of discharge, operating temperature and charging efficiency. Other important factors are shelf life, material compatibility, electrode seal reliability and venting, if required.

Various cells have the potential for achieving the desired performance. However, only three candidate batteries appear promising for this application. They are listed in Table 3.2.3-8. The estimated weight and volume of these cells as a function of energy storage are shown on Figures 2.8-4 and 2.8-5. A review of these figures reveals that the nickel-cadmium battery would be too heavy and voluminous to be considered. The two high-temperature batteries, i.e., sodium-sulphur and lithium-selenium are the most suitable. Both these batteries are in the development stage; however, at the current rate of development, these batteries could be available in 2 to 3 years. Hence, for this rotary Rankine cycle system, the sodium-sulphur, solid electrolyte battery which operates at 570° F was selected. Because of its operating temperature, it can be interposed between the heat source and the 430° F peak cycle temperature boiler. At 12.5 watt-hours of energy storage, this battery will weigh approximately 0.125 pound and require a volume of 3.2 in.³. Both these values are consistent with maintaining the weight and volume constraints for the overall thermal converter.

Table 3.2.3-8. Candidate Secondary Batteries

- Safe Sealed Operation
- Cycle Life at Least 3650 Cycles
- High Energy Density
- Good Depth of Discharge Characteristic
- High Recharging Rate and Efficiency
- High Power Density (20 to 50 watts/lb)

	Energy Density (w-hr/lb)	Depth of Discharge (%)	Relative Weight	Recharging	Power Density (watts/lb)	Development
Nickel-Cadmium	8 (100° F)	25	33	83% eff. 5-8 hours	> 50	Available
Sodium Sulphur (570° F)	100	100	0.68	90% eff.	150	2-3 years
Lithium-Selenium (710° F)	50	100	1.36	100% eff. 1-3 hours	~ 90	

Working Fluid Stability

The two factors that are paramount in assessing working fluid stability are pyrolytic (thermal) and radiolytic (radiation) induced decomposition. In the case of pyrolytic decomposition, the selection of a comparatively low peak cycle temperature of 430° F assures that no problem exists in this area. As was indicated in Figure 3.2.3-1, the total decomposition to be expected in 100,000 hours (greater than 10 years) is 0.05%. This is further confirmed by the test data shown on Figure 3.2.3-18, which indicates that total thermal decomposition would be less than 0.5% at

Figure 3.2.3-18. Test Data on Thermal Decomposition of CP34 as a Function of Peak Cycle Temperature

the end of 20 years. Hence, it is evident that thermal decomposition will not be a problem in this application.

The second factor to be considered is that of radiolytic decomposition. The working fluid being utilized for the candidate rotary vapor cycle is CP-34 (Thiophene), a product of the Monsanto Co. Its major properties are:

Chemical formula	C_4H_4S
Molecular weight (M)	84.13
Gas constant (R)	18.4 ft/° R
Ratio of specific heats (k)	1.11
Specific heat at constant pressure	0.345 Btu/lb-° F (at 400° F) 0.236 Btu/lb-° F (at 150° F)

To estimate the design margin for radiolytic decomposition it is first necessary to determine the integrated neutron flux (total fluence) at the 1-Mev level and the total cumulative dose of gamma irradiation.

For this application, the heat source is medical grade plutonium-238 oxide. In addition, the thermal energy of the heat source may vary from 37 to 60 watts. For this evaluation, the higher value was assumed to be conservative. From layouts of the vapor Rankine cycle system, the distance from the center of the heat source to the fluid in the boiler tubes was determined to be 2.3 cm. Under these conditions and assuming no attenuation from the heat source encapsulant, the insulation, and the high temperature (570° F) battery, the following neutron and gamma, 10-year doses were determined:

$$\text{Gamma irradiation dose} = 2.75 \times 10^6 \frac{\text{ergs}}{\text{gm}} (\text{C}) = 2.75 \times 10^4 \text{ rads}$$

$$\text{Neutron irradiation dose} = 2.02 \times 10^{12} \text{ nvt}$$

The threshold values for organic fluids have been defined as those values where the density, viscosity, and carbon-hydrogen ratio increases are of such magnitude that the fluid becomes unsuitable as a heat transfer medium. While specific data on thiophene are currently being developed at Battelle Memorial Institute, a good comparison can be made at this time with four similar organic fluids for which data are available. These are shown in Table 3.2.3-9.

Table 3.2.3-9. Damage Thresholds for Organic Fluids

Organic Fluid	Neutron Irradiation (nvt)	Gamma Irradiation ($\frac{\text{ergs}}{\text{gm}} \text{ C}$)
Monoisopropylbiphenyl	1×10^{18}	2×10^{12}
Terphenyls	1×10^{18}	4×10^{12}
Biphenyl	1×10^{18}	1.4×10^{12}
Dowtherm "A"	1×10^{18}	$> 1 \times 10^{11}$

From these data, a design margin equivalent of six orders of magnitude exists for both neutron and gamma irradiation, so that radiolytic decomposition, as it would affect the thiophene viscosity, density, and carbon-hydrogen ratio, would be negligible.

The formation of noncondensable gas is another potential area of concern. However, at exposures as high as 1×10^{18} nvt, the G (gas) values (molecules of gas per 100 ev of absorbed energy) are quite low for monoisopropylbiphenyl and biphenyl. For example, the available G-values for biphenyl are 6.5×10^{-3} for hydrogen and 4×10^{-4} for acetylene. These are 1 to 2 orders of magnitude less than the values obtained for alkylidiphenyl ($6.2 - 9.5 \times 10^{-2}$) where gas evolution amounted to less than 1 cc/gram for an irradiation dose of $3.7 - 4.1 \times 10^{10}$ erg/gm (C). Since the estimated dose for this application is four orders of magnitude less as well, the gas evolution should also be negligible.

In summary, the neutron and gamma irradiation total doses are quite low. In addition, based upon the data available for threshold values and G-values for similar organic fluids, it appears that the design margins for the radiolytic decomposition of thiophene are more than adequate.

Turbogenerator Speed Control

The turbogenerator speed control circuit is simple. It is activated by the output voltage rising above a certain set point. Referring to Figure 3.2.3-19, the circuit is seen to consist of three major elements; a filter, a comparator with hysteresis, and a power switch to turn on an additional parasitic load.

The filter removes any line spikes which may inadvertently be mistaken by the comparator for an input voltage change. The input voltage is then compared with a reference voltage, by the comparator reference, and the state of the comparator changes when the input exceeds the reference voltage.

In order to reduce unwanted cycling, voltage hysteresis is provided in the comparator. The output of the comparator actuates a transistor switch, turning on an additional external load. When the load is added,

Figure 3.2.3-19. Turbogenerator Speed Control

the turbogenerator speed reduces which reduces the output voltage. When the output falls below the set point, minus hysteresis, the switch turns off. This produces a steady dc output voltage with a slight triangular wave on top of the dc. The amplitude of the triangular wave will be $\pm 2\%$ and the period will depend upon the load applied. This method of control is very reliable and simple; yet it provides the required performance.

The load will be configured and located so as to carry away the heat energy generated without excessive temperature rise. The control will be physically placed in the dc motor-actuator housing of the electromechanical actuator.

Rotary Vapor Engine Thermal Map

An important factor in the design of a small turbopump-generator is the resulting thermal gradients. With the turbine inlet at 430°F and the pump inlet at 115°F , adequate thermal barrier and cooling techniques must be provided to assure that the steady-state temperatures are compatible with the operating characteristics of each region of the TGU. Accordingly, a preliminary thermal map of the TGU was made. The results of this analysis are shown on Figure 3.2.3-20. In establishing this thermal map, it was assumed that the housing in the region of the generator would be maintained at 300°F by the insulation design. It was also established that the turbine end bearing would be lubricated and

Figure 3.2.3-20. Rotary Vapor Cycle Thermal Map

cooled by 300° F vapor, bled through a flow control nozzle from the turbine inlet. This bearing coolant then flows to the pump-end bearing and is finally directed through porting in the housing to the gas side of the regenerator. In this way, the heat absorbed in the bearings can be regeneratively returned to the pump discharge liquid, thus improving overall system efficiency. The generator stator windings and housing are maintained at 300° F by utilizing the pump discharge flow as a coolant. After removing heat from the generator, this flow is then directed to the liquid side of the regenerator.

From the temperatures indicated in Figure 3.2.3-20, there are no regimes where the resulting temperature gradients will create any heat transfer or differential expansion problems. In the final design configuration, additional thermal barriers can be incorporated (i. e., a central hole in the shaft and grooves in the stator housing) if they are found necessary.

3.2.3.7 Candidate Rotary Rankine System

As a result of the analyses conducted, the rotary vapor system using CP-34 as a working fluid has been selected as the candidate system for this class of heat engines. The cycle conditions are as shown in Table 3.2.3-3. The main elements comprising this system are shown on Figure 3.2.3-21. The system includes the combined rotating unit (turbine, pump, dc generator), three heat exchangers (boiler, condenser, regenerator), a solid electrolyte, elevated temperature (570° F) battery and charge control unit, an isotope heat source ($\text{Pu}^{238}_{94}\text{O}_2$), the necessary insulation (Min-K/xenon), and solid state electronics for TGU speed control and battery charge control. The overall system is capable of producing a 9.6 watts electrical output when combined with a 60-watt thermal heat source for an overall system efficiency of 16.0%. However, minimization of the heat source thermal requirements was an important objective of the program. Hence, it was determined that by providing a 12.5 watt-hr battery and directing this system design toward providing 2.81 watts (average) to the blood pump through a 45% efficient dc motor-driven mechanical actuator, that the output from the turbogenerator need be only 6.25 watts_e. This permitted a reduction in the heat source size to 41 watts thermal (which still includes a 10% allowance for heat losses) and is also compatible with the use of Min-K insulation.

The overall arrangement of the heat engine for this system is shown in Figure 3.2.3-22. The components have been arranged in a manner to result in minimum volume and insulation requirements. In addition, thermal interfaces have been arranged to reduce overall heat losses. For example, the high temperature battery has been designed as a cylindrical receptacle to enclose a high percentage of the heat source surface area.

Furthermore, since the peak cycle temperature for the rotary vapor cycle working fluid is 430° F, this permitted a boiler design which could surround and be integrated with the 570° F battery. Another feature that has been employed is to locate the high temperature turbine and the regenerator adjacent to the heat source, further minimizing system heat losses. Finally, the condenser, which is operating in the

Figure 3.2.3-21. Rotary Vapor Cycle System with Regeneration and Electrical Power Output

Figure 3.2.3-22. Rotary Vapor Cycle Heat Engine

115° to 117°F temperature range, has been mounted on the inner surface of the engine container. This permits heat rejection directly to the body fluids and tissue, eliminating the need for a separate heat rejection fluid. The heat source is a vented capsule as described in Section 2.6.

The envelope dimensions of this system are 3.25 inches in diameter by 6 inches long. This results in a total volume of 49 cubic inches (0.804 liters). The total weight of this rotary Rankine heat engine is 1.91 pounds or 0.868 kg. (See Table 3.2.3-10 for Weight Summary.) The output from the generator will be 24 vdc power. This electrical energy will not only be used to provide power to a dc motor-driven mechanical actuator, but will also provide energy for recharging the battery and for the system electronic control circuitry. The overall performance for this system and a 60-w_t system are shown on Table 3.2.3-11. A detailed discussion of the characteristics of the components that comprise this candidate heat engine configuration was given in Section 3.2.3.6.

Table 3.2, 3-10. Rotary Vapor Cycle System - Weight Summary

Combined Rotating Unit		
Pump and Scroll	0.0362	
Turbine and Housing	0.1775	
Alternator and Housing	0.2385	
Shaft	0.0183	
Heat Exchangers		
Preheater	0.0134	
Boiler (10 deg superheat)	0.0431	
Regenerator	0.0149	
Condenser	<u>0.0600</u>	
Subtotal		0.6019 lb; 0.273 kg
Heat Source (41 watts _t)	0.2270	
Battery	0.1250	
Min-K Insulation	0.4260	
Electronic Controls	0.3600	
Miscellaneous Tubing	0.0400	
Container	<u>0.1270</u>	
Subtotal		<u>1.3050 lb; 0.595 kg</u>
Total		<u><u>1.9069 lb; 0.868 kg</u></u>

Table 3.2, 3-11. Rotary Vapor Cycle System Performance Summary

Isotope Thermal Power (watts _t)	41	60
Electrical Power Output (watts _e)	6.25	9.6
Overall Heat Engine Efficiency (%)	15.7	16.0
Peak Cycle Temperature (° F)	430	430
Peak Cycle Pressure (psia)	250	250
Condensing Temperature (° F)	116	116
Condensing Pressure (psia)	4	4
Maximum Heat Rejected to Body (watts _t)	34.75	49.4
Total Weight (lb)	1.91	2.25
(kg)	0.868	1.02
Total Volume (liters)	0.778	0.900
Specific Gravity	1.11	1.13

3.2.4 Thermoelectrics

3.2.4.1 Historical Background

The thermoelectric engine makes use of a thermoelectric converter to transform heat generated by the radioisotope heat source directly into electricity. The basic concept of the thermoelectric converter has been in use since the early 1800's. The fabrication of devices producing usable levels of power at reasonable efficiency became practicable with the availability of semiconductor materials and the development of solid state physics theory in the early 1950's. Since that period, extensive development by the AEC has resulted in the use of thermoelectric converters for a variety of applications in space, underwater, and cardiac pacemakers where long life and high reliability under unattended operating conditions is mandatory.

A partial list of units developed under the AEC's SNAP program is given in Table 3.2.4-1. The power level of these units ranges from milliwatts to several hundred watts. SNAP-3, a 3-watt unit, has been operating in space for over a decade.

Efficiency of the SNAP units has increased significantly. The earliest units using PbTe had efficiencies in the range of 5%. Later units, such as SNAP-21 and SNAP-23, use segmented couples and achieve an efficiency of 8 to 9%. The recent development of refractory metal heat sources with a temperature capability of $\sim 1000^{\circ}\text{C}$ allows the consideration of cascading which leads to efficiencies in excess of 12%.

3.2.4.2 Materials Selection

Thermodynamically, the thermoelectric converter may be considered as a heat engine with its maximum efficiency limited to that obtained with an ideal Carnot engine. Three basic properties of a thermoelectric material are used in determining the performance of a thermoelectric converter; these are the Seebeck coefficient (α), the electrical resistivity (ρ) and thermal conductivity (k). The figure of merit for a thermoelectric material (Z) is defined as $\alpha^2/\rho k$, and the theoretical efficiency of a thermoelectric material can be approximated by

$$Z T_H \times \frac{T_H - T_C}{T_H}$$

Table 3.2.4-1. RTG Development History

System	System Contractor	T/E Manufacturer	T/E Material	Power Level	Application
SNAP-3	Martin	3M	PbTe	3w	Space
SNAP-7(A, C, E) (B, D, F)	Martin Martin	Martin Martin	PbTe PbTe	10w 60w	Marine Marine
SNAP-9	Martin	3M	PbTe	25w	Space
SNAP-10A	AI	RCA	SiGe	500w	Space Reactor
SNAP-15A B	GGA NUMEC	GGA NUMEC	BiTe Metallics	1-10mw 1-10mw	Classified Classified
SNAP-19	Martin	3M/Martin	PbTe	40w	Nimbus/Pioneer
SNAP-21A B	Martin 3M	Martin 3M	PbTe PbTe	10w 10w	Marine Marine
SNAP-23	Westinghouse	3M	PbTe	50w	Marine
SNAP-27	GE	3M	PbTe	60w	ALSEP
TRANSIT	TRW	GGA	PbTe	30w	Space

where T_H is the hot side temperature, T_C is the cold side temperature and

$$\frac{T_H - T_C}{T_H}$$

is the Carnot efficiency. The overall efficiency of a thermoelectric converter can be defined as $\text{Eff}_{ov} = \eta_c \times \eta_M \times \eta_T$ where η_c is the Carnot efficiency, η_M is the theoretical material efficiency (a function of Z only) and η_T represents the thermal and electrical losses associated with the fabrication of a practical device. From the above equation it is clear that the highest efficiency is obtained by operating the device over a large temperature range to obtain the maximum available Carnot efficiency.

For the present application the hot side temperature is limited to approximately 1000°C by heat source considerations and the cold side temperature is fixed at approximately 40°C by physiological considerations. As shown in the curves of Figures 3.2.4-1 and 3.2.4-2, there is no single material which has a high figure of merit over the entire temperature range; therefore, to achieve a highly-efficient device, it is necessary to combine the materials in a manner which allows the material to be operated over

the temperature range in which it exhibits its maximum efficiency. Materials may be combined in two general ways, by cascading or by segmenting.

In the cascading method, the thermoelectric materials are arranged so the element size can be independent for each material used. Each element of one material is not necessarily thermally or electrically in series with a single element of the other material. The elements of each stage are electrically independent of those in the next stage and the number of elements of one stage may

Figure 3.2.4-1.
Figure of Merit Versus
Temperature, N-Materials

Figure 3.2.4-2.
Figure of Merit Versus
Temperature, P-Materials

differ from that of the other stage.

In the segmenting method the materials are joined in such a manner that a single element consists of both materials. Thus the number of elements is identical in both stages and each stage of one material is thermally and electrically in series with the corresponding stage of the other material.

The choice of segmenting versus cascading to obtain maximum efficiency depends on the properties of the materials. When the materials have similar properties, the segmenting method is

advantageous because it eliminates the need for interstage electrical connections and minimizes thermal and electrical losses. If the materials have widely differing properties, the requirements for maximum efficiency, optimum intermediate temperature, and optimum current cannot be met by segmenting because the only parameter which can be varied is the geometry of the element. The use of cascading allows an adjustment of both the geometry and number of couples to yield the optimum current for each stage.

Analyses using presently available materials have shown that the maximum efficiency over the available temperature range is obtained by cascading SiGe with a segmented second stage using the telluride family of materials. A schematic of this converter system is shown in Figure 3.2.4-3. The analysis indicates that the efficiency of the cascaded system is 10 to 15% greater than that obtained when the two stages are combined by segmenting. For the cold stage, because the properties of the materials are quite similar, cascading of the telluride materials offers no significant advantage in efficiency and segmenting minimizes fabrication problems.

Figure 3.2.4-3. Typical Cascaded/Segmented Thermoelectric Converter

In the present design, the SiGe hot junction is operated at 950°C , which is enough below its maximum operating temperature of 1100°C to assure low degradation rates. An interstage temperature of 500°C was chosen to be compatible with the temperature capability of the telluride materials.

Summary of Candidate Materials

When the figure of merit is plotted against temperature, the relative performance of various materials can be compared and intersegment temperatures can be chosen. Such plots are presented in Figures 3.2.4-1 and 3.2.4-2 where Z is plotted against temperature for the N-type and P-type thermoelectric materials presently under consideration. The inherently low material efficiency of SiGe, relative to the telluride materials, can be seen to be caused by its relatively low figure of merit. The advantage in cascading SiGe and telluride materials is also apparent since tellurides cannot operate above $\sim 600^{\circ}\text{C}$ and SiGe efficiencies are much lower than telluride efficiencies below about 600°C .

A promising high temperature thermoelectric material which possesses a greater figure of merit than SiGe, and is capable of operating

at equivalent temperatures, has been developed by Minnesota Mining and Manufacturing Co. for the AEC. This P-type material, TPM-217 (see Figure 3.2.4-2), possesses unusual thermoelectric properties which have made the development of an N-type material unsuccessful to date. Calculations have shown that if an N-type material can be developed, overall RTG systems efficiencies as high as 16% are possible for a cascaded TPM-217/telluride generator.

Segmented Couple Design (Cold Stage)

The design and optimization of segmented couples requires analytical techniques which will perform a heat balance at each material junction and optimize the relative couple sizes to yield peak efficiencies. A computer program was written for the TRW Timesharing System to perform the iterative calculations required to optimize the segmented couple sizes.

Six thermoelectric materials were considered for comparative evaluations, three N-type and three P-type. The materials considered were:

<u>P-Type</u>	<u>N-Type</u>
● Bismuth Antimony Telluride	● 2N Lead Telluride (TEGS)
● TAGS 85	● 3N Lead Telluride (TEGS)
● Silver Antimony Telluride	● Lead Telluride/Germanium Telluride

Junction temperatures were chosen by using Figures 3.2.4-1 and 3.2.4-2 and operating at the temperature at which the figures of merit of the segmented materials crossed. For instance, the temperature at the TAGS and AgSbTe junction would be 265°C.

Table 3.2.4-2 is a summary of the couple efficiencies obtained by segmenting various material combinations. The results of the evaluation indicate that

- PbTe-GeTe/2N - TAGS/BiSbTe is the most efficient couple combination considered.
- Adding a AgSbTe segment to the TAGS/BiSbTe p-leg reduces the couple efficiency.
- The unsegmented PbTe-GeTe/TAGS couple yields good efficiency.

Table 3.2.4-2. Material Efficiency for Optimized Segmented Couples
 $T_H = 500^\circ\text{C}$, $T_C = 50^\circ\text{C}$

		N - MATERIALS		
		PbTe-GeTe/2N	3N/2N	PbTe-GeTe
P M A T E R I A L S	TAGS/AgSbTe/BiSbTe	11.99	11.18	11.73
	TAGS/BiSbTe	12.30	11.41	12.02
	TAGS	11.98	11.09	11.70

The addition of a AgSbTe segment to the TAGS/BiSbTe P-leg reduces the couple efficiency because of the variation in thermoelectric properties between AgSbTe and TAGS or BiSbTe. TAGS and BiSbTe have α , ρ , and k properties relatively close to each other; however, AgSbTe properties are much different. This difference in thermoelectric properties results in a "loading down" effect when AgSbTe is added to the leg. This "loading down" effect overrides the figure of merit increase shown in Figure 3.2.4-2.

Based on the considerations discussed above a hot stage using SiGe N and P elements and a cold stage using segmented PbTe-GeTe/2N N elements and segmented TAGS/BiSbTe P elements were selected for the conceptual design.

The PbTe-GeTe material has been well characterized and a limited number of devices using the material have been built; however, little life data are available. Both the TAGS and BiSbTe materials have been used successfully and long-term data are available for these materials as individual elements. Although theoretical considerations indicate that there should be no problem in segmenting the material, no test data are available. Should difficulties be encountered with the cold stage materials, a more conservative approach using 3N/2N N material with TAGS P material can be taken; the absolute decrease in efficiency using this combination is approximately 1%.

Cascading Design Considerations

When two or more thermoelectric materials or cascaded stages are connected electrically in series the stages must be electrically "matched" so that one stage does not "load down" the other stage(s). Matching can generally be performed in a thermoelectric generator by varying the size and number of thermoelectric couples. Perfect matching is achieved in a series circuit by having the ratio (V_{oc}/R_i) equal for both stages, however analysis has shown that virtual peak power may be obtained when

$$\left(\frac{V_{oc}}{R_i}\right)_1 / \left(\frac{V_{oc}}{R_i}\right)_2 \approx 1.5$$

For the converter under consideration, design of the stages for individual loads results in

$$\left(\frac{V_{oc}}{R_i}\right)_{Seg} / \left(\frac{V_{oc}}{R_i}\right)_{SiGe} \approx 6$$

and only 70% of peak power when the stages are combined. When the couple sizes and number of couples were altered so that

$$\left(\frac{V_{oc}}{R_i}\right)_{Seg} / \left(\frac{V_{oc}}{R_i}\right)_{SiGe} \approx 1.5$$

it was found that peak power was achieved. The resulting couple sizes are summarized in Section 3.2.4.4.

Superinsulation Versus Fibrous Min-K Insulation

The choice of a thermal insulation system for a cascaded RTG for an artificial heart power source is not straightforward. Although the use of superinsulation reduces the thermal shunt losses, there are several potential disadvantages. A summary of the merits of each system is presented in Table 3.2.4-3.

One interesting tradeoff involves the use of superinsulation versus fibrous insulation in the telluride stage of the converter. Because of the increased sublimation rate of telluride thermoelectrics in vacuum, it is assumed that, if superinsulation were used in this stage, the interstage temperature of the converter must not exceed 400°C. However, the use

Table 3.2.4-3. Summary of Insulation Merits

<u>Superinsulation</u>	
<u>Advantages</u>	<u>Disadvantages</u>
<ul style="list-style-type: none"> ● Low Thermal Conductivity 	<ul style="list-style-type: none"> ● Must Operate Telluride Stage Below 400°C ● Potential Outgassing Problem ● Potential Electrical Shorting Problem ● Joint Losses ● High Density

<u>Fibrous Insulation</u>	
<u>Advantages</u>	<u>Disadvantages</u>
<ul style="list-style-type: none"> ● State-of-the-Art Material ● Can Operate Telluride Stage Up to 550°C ● No Electrical Shorting Problem ● Minimal Joint Losses ● Low Density 	<ul style="list-style-type: none"> ● Higher Thermal Conductivity ● Inert Gas Containment Required to Operate Telluride Stage Above 400°C

of inert gas filled fibrous insulation (Min-K in this case) allows one to operate the interstage as high as 550°C, (500°C was assumed for conservatism). Figure 3.2.4-4 is a plot of the individual stage thermoelectric efficiencies and the total generator thermoelectric efficiency as a function of the interstage temperature. The figure shows an increase in the total efficiency as the interstage temperature is increased, although the hot and cold junction temperatures are maintained at 950° and 50°C, respectively. Using superinsulation in the telluride stage ($T_I = 400^\circ\text{C}$) results in a total thermoelectric efficiency of 14.6% whereas using fibrous insulation ($T_I = 500^\circ\text{C}$) results in an efficiency of 15.8%. The efficiencies reported in Figure 3.2.4-4 include the material efficiency and electrical losses but do not include the shunt heat losses. When the shunt losses are included

Figure 3.2.4-4. T/E Efficiency Versus Interstage Temperature

in the analysis the comparison (assuming a 60-watt heat source) is as shown below:

	Superinsulation	Fibrous Insulation (Min-K)
Thermal Inventory (w)	60	60
End Losses (w)	6	6
Heat into SiGe Stage (w)	54	54
Shunt Heat in SiGe Stage (w)	1	13.9
Heat into SiGe Elements (w)	53	40.1
Power Output SiGe Stage (w)*	2.5	1.6
Heat into Segmented Stage (w)	50.5	52.4
Shunt Heat in Segmented Stage (w)	0.3	7.5
Heat into Segmented Elements (w)	50.2	44.9
Power Output Segmented Stage (w)**	5.1	5.5
Total Power Output (w)	7.6	7.1
System Efficiency (%)	12.6	11.8

*SiGe Stage Efficiencies: 4.8% S/I, 4.0% Fibrous } See Figure
 **Segmented Stage Efficiencies: 10.25% S/I, 12.3% Fibrous } 3.2.4-4

It should be noted that a superinsulation system offers only a marginal efficiency advantage over a Min-K system because of the temperature limitation in the segmented stage. Other potential problems such as outgassing, joint losses, and electrical shorting may further decrease the efficiency of a superinsulation system and should be considered when comparing insulation systems.

3.2.4.3 Configuration Selection

RTG Geometry

In order to achieve a maximum efficiency system, while minimizing fabrication and developmental costs, a close-packed, flat-plate converter has been selected for the conceptual design. This type of design is considered optimum for the following reasons:

- Fabrication simplicity
- Telluride stage structural integrity
- Easy-to-maintain compressive load on couples
- Minimum insulation penetrations

Several converter geometries were investigated prior to selection of the flat-plate geometry. Both spherical and cylindrical geometries with close packed and distributed elements were considered. However, in each case, there was little difference in overall system efficiency. Fabrication simplicity was the overriding consideration in choosing the flat-plate design.

An analysis to determine the heat source geometry resulted in the selection of a right circular cylinder with an L/D of 1. This shape yields the smallest surface area and therefore minimizes the ratio of insulation area to thermoelectric material area. A schematic of the RTG is shown in Figure 3.2.4-5.

Thermal Analyses

Thermal analyses were performed to determine the thermal adequacy of the RTG system and determine the effect of varying converter geometry on converter efficiency, temperature distribution, and heat source temperature. A simplified two-dimensional model of a cascaded

Figure 3.2.4-5. RTG Schematic

unit was developed for this analysis. Of chief interest was the optimization of the thermal insulation required to maintain the converter at the required temperatures. It is known that adding insulation will increase the generator efficiency; however, it must also be realized that a marginal gain in efficiency may be at the expense of a substantial weight penalty.

Table 3.2.4-4 presents the results of the analyses performed for insulation thicknesses of 0.5, 0.75 and 1.5 inches, respectively. The insulation was assumed to be Min-K 2020 filled with argon (Reference 14). Argon was chosen for this analysis since it is known to be an excellent cover gas capable of inhibiting PbTe sublimation; experimental thermal conductivity data were also readily available. Other gases such as xenon and krypton are possible candidates as fill gases. In any case, the direction chosen is believed to be conservative and the possible change to xenon, krypton or even vacuum operation would only enhance the generator thermal performance. For purposes of calculating heat source temperatures, the silicon germanium hot shoes were assumed to be separated from the heat source by a distance of 0.5 mil; conduction through the argon gas and radiation were the sole modes of heat transfer considered.

The results of the thermal analyses are shown in Figure 3.2.4-6. It should be noted that only a small gain in efficiency is obtained when the insulation thickness exceeds 1 inch, although the size and weight of the system increase significantly. In order to compare the three designs

Table 3.2.4-4. Summary of Thermal Analysis Results

	Insulation Thickness (In)		
	0.5	0.75	1.5
Heat Source Thermal Inventory (w)	60	60	60
Heat Entering Converter (w)	45.7	47.7	49.6
Heat Entering End Insulation (w)	14.3	12.3	10.4
Maximum Heat Source Surface Temperature ($^{\circ}\text{C}$)	1004	1054	1104
SiGe Hot Junction Temperature ($^{\circ}\text{C}$)	969	1018	1065
Interstage Temperature ($^{\circ}\text{C}$)	483	510	538
Sink Temperature ($^{\circ}\text{C}$)	50	50	50

Figure 3.2.4-6. Converter Characteristics Versus Insulation Thickness

in Table 3.2.4-4 at the same operating temperatures, the cross-sectional areas of the thermoelectric elements are adjusted as required. However, the effect of this change on converter weight is negligible.

3.2.4.4 Component Design

System Requirements

In order to deliver 2.81 watts to the blood pump, the following diagram shows how the heat source thermal requirement is determined.

The thermoelectric converter includes both the thermoelectric materials and the thermal insulation; the 11.6% efficiency represents the overall engine efficiency including both thermal and electrical losses. The PCCS efficiency is described in Section 3.1.

Parametric Design Curves and Selection

Figure 3.2.4-7 presents the graphical results of an analysis performed to determine the design characteristics of an RTG as a function of the heat source thermal inventory. Based on previous thermal analyses reported in Section 3.4.4.3, an insulation thickness of 1 inch was selected. Adding the design constraint that the RTG must produce 6.25 watts (e), the following RTG design characteristics may be read from Figure 3.2.4-7:

Figure 3.2.4-7

RTG Power Supply, RTG Characteristics Versus Thermal Inventory

Power Output: 6.25 watts (e)

Power Input: 54 watts (t)

Weight: 1.7 lb (including heat source)

Efficiency: 11.6%

Outer Diameter: 3.3 inches

Thermoelectric Couple Sizing

The size of the thermoelectric couples is dictated by the converter heat flux, desired junction temperatures, and structural considerations.

Theoretically the power output of a thermoelectric device is independent of the geometry of the element as long as the ratio of the area of the element to its length is maintained. In a practical device, however, electrical losses in the conductor between elements, losses caused by contact resistance, and thermal losses through the insulation, become increasingly important and cause a decrease in power as element length and area are reduced to minimize weight.

The couple size chosen for the conceptual design presented here does not represent a completely optimized system but was selected to give couples that are producible by available fabrication techniques and give assurance of having structural integrity.

Heat balances were performed for each stage to determine the number of couples required and the area to length of each thermoelectric element. An iterative method was employed to achieve an integral number of couples and eliminate couple-truncation power losses. The following couple characteristics result:

Silicon Germanium Stage (8 couples)

Couple Length:	0.3 in.
N-element Cross-Sectional Area:	0.0179 in^2
P-element Cross-Sectional Area:	0.0098 in^2

Segmented Stage (26 couples)

Couple Length:	0.6 in.
N-element Cross-Sectional Area:	0.0212 in^2
PbTe-GeTe Length:	0.495 in.
2N Length:	0.105 in.
P-element Cross-Sectional Area:	0.0170 in^2
TAGS Length:	0.432 in.
BiSbTe Length:	0.168 in.

Couple Mounting

The SiGe couples are metallurgically bonded to SiMo hot shoes which are in turn compressively loaded against the heat source. The use of a beryllium oxide coating is necessary to prevent reaction of the SiMo with the Pt-Rh clad. The SiGe couples themselves are evenly spaced over the flat ends of the heat source and metallurgically bonded to the interstage plate. The telluride stage is metallurgically bonded to the center of the containment vessel end cover in a close-packed array intended to enhance the structural integrity of the unit. The two converter modules are held in compression against the heat source by means of tension rods located between the end covers. The entire assembly is packed with Min-K, back-filled with argon, and hermetically sealed in a cylinder.

3.2.4.5 System Design Summary

The thermoelectric/battery engine is designed to minimize the heat source power requirement by sizing the heat source and thermoelectric converter to provide the average heart pump power requirement, 2.81 watts, with the peak power supplied by the battery.

Without the use of a battery, peak power (4.44 watts at the blood pump) must be provided continuously. Since the maximum practical efficiency of the thermoelectric converter is about 12% and the PCCS

efficiency is about 45%, a converter capable of producing 9.85 watts (e) would be required. The corresponding heat source requirement, 82 watts (t), is in excess of the maximum allowable. The use of a 12.5 watt hour sodium-sulfur battery, Section 2.8, results in a thermoelectric converter power requirement of 6.25 watts (e) and reduces the heat source to 54 watts (t).

Since a waste heat flux limitation of 0.07 watts/cm^2 is used, the container for the engine is large enough to also contain the motor/reciprocator.

Figure 3.2.4-8 is a layout of the system consisting of the RTG, the motor/reciprocator PCCS and a thermal battery. A summary of the pertinent design features is given below:

Heat Source Thermal Input	54 watts (t)
RTG Power Output	6.25 watts (e)
Load Current	3.28 amps
Load Voltage	1.91 volts
Blood Pump Power Input	2.81 watts (ave)
Volume	77.4 in^3
Surface Area	112 in^2
Weight	2.96 lbs
Specific Gravity	1.06
SiGe Operating Temperatures	$T_H = 950^\circ\text{C}, T_C = 525^\circ\text{C}$
Telluride Operating Temperatures	$T_H = 500^\circ\text{C}, T_C = 50^\circ\text{C}$
Maximum Heat Source Surface Temperature	1079°C

Figure 3.2.4-8. Thermoelectric/Battery Engine

The system component weight breakdown is given in Table 3.2.4-5. The sodium sulfur battery provided to meet peak power is contained within an annular space around the RTG and surrounded by insulation to maintain it at a mean temperature of 300°C (570°F).

Table 3.2.4-5. Thermoelectric/Battery Weight Summary

<u>Item</u>	<u>Weight (lb)</u>
Heat Source	0.551
Converter	0.828
SiGe Element	0.081
Telluride Elements	0.287
Interstage Plate	0.037
Electrical Connections	0.017
Insulation	0.283
Container	0.123
Battery	0.130
Motor Reciprocator	0.880
Outer Container	0.562
Total	2.951

3.2.5 Hybrid

The term 'hybrid' engine is used here to describe a system in which the power input to the blood pump is provided by two separate thermal energy converters. Since all the candidate concepts are, by definition, capable of operating for 10 years with high reliability, there is no advantage to operating two converters in parallel (thermally) solely for the sake of redundancy. At best, the efficiency of two parallel converters can approach that of a single, larger device, and then only at the expense of weight and volume.

In order for the hybrid concept to be attractive, the two thermal converters must be physically and thermodynamically suited to operating in series (thermally) at an overall efficiency in excess of that attainable with either converter alone.

The only two candidate concepts which qualify under these groundrules are the rotary vapor and thermoelectric systems, which we will hereafter refer to simply as the 'hybrid' system. Since both the rotary vapor and thermoelectric converters separately employ (require) a solid electrolyte battery for energy storage, an obvious configuration for the hybrid system would also include this component. This hybrid/battery system is discussed in Section 3.2.5.1. However, with the improved overall efficiency of the hybrid combination, it is now possible to consider a system which does not require a battery to remain within all of the design groundrules. This option is described in Section 3.2.5.2.

The hybrid concept is similar to a thermodynamic binary cycle in which the rejection temperature of one system corresponds to the peak cycle temperature of the second system. In this application the thermoelectric module hot junction temperature is essentially equal to the heat source wall temperature, while the second stage thermoelectric cold junction temperature is essentially the hot-side input to the rotary vapor cycle. In this manner, all the heat not converted to electricity in the thermoelectric stage is still fully available for conversion in the rotary vapor stage.

As explained in Section 3.2.4, for the optimized thermoelectric configuration the elements are placed on both ends of a right circular cylinder. Since the rotary vapor cycle boiler operates at a maximum temperature of 430°F , the thermoelectric cold junction temperature is established at 464°F (240°C).

3.2.5.1 Hybrid/Battery Engine System

This engine system utilizes the hybrid concept previously described, and a sodium-sulfur battery to provide energy storage. With this configuration, the electrical output from the engine can be designed to provide the average power required by the blood pump. As discussed in the electrochemical energy storage section, a 12.5 watt-hour battery is required for this application. Under these conditions the required heat source for the hybrid/battery system is only 37 watts thermal.

In this configuration, the thermoelectric converter rejects heat to the boiler of the rotary vapor cycle system. The peak boiler operating temperature is maintained at 430°F (221°C), which limits the minimum allowable cold junction temperature of the 2N/TAGS couples. The operating temperatures of the individual stages of the cascaded thermoelectric converter and the rotary vapor cycle are as follows:

<u>Silicon-Germanium Stage</u>	<u>2N/TAGS Stage</u>	<u>Rotary Vapor Cycle</u>
$T_H = 950^{\circ}\text{C}$	$T_H = 475^{\circ}\text{C}$	$T_{\text{Boiler}} = 221^{\circ}\text{C}$
$T_C = 500^{\circ}\text{C}$	$T_C = 240^{\circ}\text{C}$	$T_{\text{Cond}} = 46.7^{\circ}\text{C}$

As can be seen, this hybrid concept has been thermally integrated so that the low temperature side of each stage can provide the thermal input to the high temperature region of the next stage. In addition, the battery, which operates at 300°C , can also be packaged within the insulation so that the heat loss from the battery casing can be utilized as a heat input to the rotary vapor cycle system boiler. Figure 3.2.5-1 shows schematically the thermoelectric elements and battery arrangement with respect to the isotope heat source. The design approach taken is similar to that described for the thermoelectric converter system (Section 3.2.4). This design also employs a

cascaded converter with 63% Si - 37% Ge couples in the high temperature stage and 2N/TAGS couples in the low temperature stage.

The 2N/TAGS stage is not segmented because the temperature differential across the elements is not sufficiently large to make segmenting effective. For this engine configuration, most of the power is provided by the rotary vapor cycle turbogenerator. Hence, the most state-of-the-art thermoelectric materials were selected for both stages because the performances of these materials would be adequate. The stages are electrically connected in series and the couples within each stage are connected in a series-parallel arrangement. The couple sizes were designed to incorporate electrical matching between stages so as to provide peak power operation.

Figure 3.2.5-1. Hybrid-Battery Thermoelectric Converter

The overall hybrid/battery engine system, shown in Figure 3.2.5-2, is capable of meeting average daily power requirements. The 12.5 watt-hour sodium-sulfur battery has been sized to provide the supplementary peak power requirements. This battery has an annular configuration, occupies 3 cubic inches and is embedded in the thermal insulation to maintain itself at a mean temperature of 300°C. The weight breakdown for the thermoelectric converters and battery is shown in Table 3.2.5-1.

Combined with the thermoelectric elements and battery is a rotary vapor cycle turbogenerator operating as previously described in Section 3.2.3. With a peak cycle temperature of 430°F, the boiler for the rotary cycle can be placed around the thermoelectric modules. The vapor cycle operating conditions and performance are similar to those discussed in Section 3.2.3 except that the output power is 4.55 watts_e at 15 volts. The vapor cycle uses CP-34 (thiophene) as a working fluid and operates between a peak cycle temperature of 221°C (at 250 psia) and a condensing temperature of 46.6°C (at 4 psia). The thermoelectric converter power gross output is 2.11 watts at 3.64 volts, but after increasing the voltage in a dc/dc converter to 15 volts this output is reduced to 1.7 watts.

Thus, the total net electric power being generated by this hybrid/battery engine is 6.25 watts. The dc motor/reciprocator described in Section 3.1.4 combined with the blood pump actuator has an overall efficiency of 45%. Therefore, the net power available to the blood pump equals the daily average requirement of 2.81 watts. As mentioned previously, the 12.5 watt-hour battery makes up the difference between this daily average and the maximum blood pump requirements.

In order to keep the volume of this system to a minimum, the fibrous insulation (Min-K with xenon fill gas) thickness and heat source were optimized. With an estimated 10% heat loss, the required heat source would be 34 watts thermal. However, in order to minimize the engine volume, an additional 3 watts was added to the heat source which resulted in a 16% heat loss, but permitted a significant reduction in overall volume. The overall characteristics of the hybrid/battery system are summarized in Table 3.2.5-2.

Figure 3.2.5-2. Hybrid/Battery Engine System

Table 3.2.5-1. Hybrid/Battery Engine-Thermoelectric Converter Weight Summary

	<u>Weight (lb)</u>
Heat Source	0.405
Silicon Germanium Elements	0.004
Telluride Elements	0.037
SiGe Hot Shoes	0.008
SiGe Cold Stack	0.026
Interstage Plate	0.029
Insulation	0.094
Telluride Cold Stack	0.076
RTG Can	0.076
Electrical Straps	0.014
Battery	<u>0.880</u>
Total Weight	1.649 lb (0.750 kg)

Table 3.2.5-2. Hybrid/Battery Engine System Design Summary

Heat Source Thermal Input	37 watts _t
Net Electrical Power Output	6.25 watts _e
Overall System Efficiency	16.8%
Total Weight (lb)	2.83
(kg)	1.28
Total Volume (liters)	1.115
Specific Gravity	1.15

3.2.5.2 Hybrid Engine System

As mentioned earlier, the hybrid engine efficiency is adequate to consider the option of no energy storage. In this case, the electrical output must be sized to meet the peak power requirements with the excess power being dissipated in the power conditioning and control unit as required. Under these conditions, this system will require a 49-watt thermal heat source. The advantage to this approach is that this engine can be developed completely from state-of-the-art components. Furthermore, when the sodium-sulfur battery becomes available, it would be a simple matter to convert this engine to the smaller heat source, hybrid/battery configuration.

This engine operates at temperature levels and cycle conditions similar to the hybrid engine with a battery. The cascaded thermoelectric converter rejects heat to the rotary vapor cycle boiler at 221°C (430°F) and the condensing temperature remains at 46.6°C (116°F). The operating temperature levels of the individual converter stages remain as outlined in Section 3.2.5.1. The major change in this configuration is the different geometry associated with the larger heat source and the sizing of the thermoelectric couples and the insulation thicknesses to minimize the overall volume. The design characteristics

of this thermoelectric converter are shown in Figure 3.2.5-3. A weight breakdown of the elements comprising the thermoelectric converter, including the heat source, is shown in Table 3.2.5-3.

In this configuration the electrical output power from the rotary vapor cycle turbogenerator is 7.25 watts_e at 15 volts. The gross electrical output power from the thermoelectric converter is 3.21 watts at 3.64 volts. In boosting this output voltage to 15 volts through a dc-to-dc converter, the net electrical power output becomes 2.61 watts. Thus the total electrical power output from the hybrid engine system is 9.85 watts_e. When supplied to the dc motor-driven reciprocator and blood pump actuator, which have an overall combined efficiency of 45%, the net power input to the blood pump is 4.44 watts which corresponds to its peak power requirements.

HEAT SOURCE THERMAL INPUT, WATTS	49		
POWER OUTPUT WATTS	3 21		
LOAD CURRENT, AMPS	0.88		
LOAD VOLTAGE, VOLTS	3 64		
TOTAL WEIGHT, LBS	0.97		
<u>SILICON GERMANIUM STAGE</u>		<u>2N/TAGS STAGE</u>	
HOT JUNCTION TEMPERATURE, °C	950	HOT JUNCTION TEMPERATURE, °C	475
COLD JUNCTION TEMPERATURE, °C	500	COLD JUNCTION TEMPERATURE, °C	240
COUPLE LENGTH, IN	0.3	COUPLE LENGTH, IN	0 3
N-ELEMENT AREA, IN ²	0.00427	N-ELEMENT AREA, IN ²	0.00977
P-ELEMENT AREA, IN ²	0.00233	P-ELEMENT AREA, IN ²	0 00433
NUMBER OF COUPLES	24	NUMBER OF COUPLES	60

Figure 3.2.5-3. Hybrid Thermoelectric Converter

Table 3.2.5-3. Hybrid Engine Thermoelectric Converter Weight Summary

	<u>Weight (lb)</u>
Heat Source	0.508
Silicon Germanium Elements	0.006
Telluride Elements	0.061
SiGe Hot Shoes	0.011
SiGe Cold Stack	0.043
Interstage Plate	0.036
Insulation	0.101
Telluride Cold Stack	0.122
RTG Can	0.067
Electrical Straps	0.016
Total Weight	0.971 lb (0.440 kg)

The overall arrangement of the hybrid engine system is shown in Figure 3.2.5-4. A summary of its design and performance characteristics is given in Table 3.2.5-4. It is interesting to note that the weight and volume of this system and the hybrid/battery engine system are nearly equivalent. This enhances their interchangeability, and the potential for heat source size reduction when the sodium-sulfur battery becomes available.

Figure 3.2.5-4. Hybrid Engine System

Table 3.2.5-4. Design and Performance Characteristics of the Hybrid Engine System

Heat Source Thermal Input	49 watts _t
Electrical Power Output	9.85 watts _e
Overall System Efficiency	20%
Total System Weight (lb)	2.8
Total System Weight (kg)	1.27
Total System Volume (liters)	1.05
Specific Gravity	1.21

3.2.5.3 Load Sharing Electronics

The hybrid-type heat engines use the electrical output power from two separate in-parallel sources, i.e., the thermoelectric converter and the turbogenerator unit. For maximum performance, it is desirable to continuously use the entire output power from the thermoelectric source and to supplement this power with the required turbogenerator power to meet the load requirements. Since the turbogenerator is providing constant output power, at less than peak load conditions, this energy can be used to charge the battery, or dissipated through the power conditioning and control system. To avoid the necessity for load matching on an impedance basis, a fairly simple approach is to employ an electronic load-sharing circuit. This circuit switches the load (dc motor) from one power source to the other at a fairly rapid rate. Using this technique each power generator operates at an effectively constant load. A switching rate of four times the dc motor speed is used so that inertia of the rotor makes switching transients negligible. With the dc motor operating at 9000 rpm, the electronic commutation is accomplished at 600 Hz.

The turbogenerator unit is provided with an overspeed control circuit (see Figure 3.2.3-19 of Section 3.2.3) which automatically protects this unit from load changes and maintains the output voltage within $\pm 2\%$. Similarly, the load-sharing electronic circuit is designed to maintain the output voltage from the thermoelectric converter at the same tolerance. Figure 3.2.5-5 is a block diagram showing the main elements associated with the load sharing and motor commutation electronics. Also shown in this diagram is a dc-to-dc converter used to increase the output voltage from the thermoelectric converter to 15 volts. This voltage level is desirable in order to achieve the required efficiencies on the turbogenerator and the dc motor. The dc-to-dc converter operates at a switching frequency of 50 kHz and an efficiency of 81.5%. Synchronous rectification is also used to reduce losses.

Figure 3.2.5-5. Load Sharing and Motor Commutation Electronic Block Diagram

3.2.5.4 Alternate Thermal Insulation Systems

As discussed in Section 2.7, fibrous insulation with a xenon fill gas was selected as the baseline thermal insulation approach for all candidate systems. However, for the hybrid engine we did carry out additional thermal analyses to determine the effect of alternate thermal insulation systems, including superinsulation, on system performance. The results are discussed below.

The baseline insulation design (summarized in the first column of Table 3.2.5-5) consists of a fibrous insulation (Min-K 2020) with a combination inert gas fill of xenon and argon. Xenon is used as the fill gas throughout the engine except in the thermoelectric modules, since no long-term data were available. If no incompatibility exists, the use of xenon in the thermoelectric package will further reduce the heat source inventory by a watt or two. The case of using argon throughout the thermal converter was also examined, and the only change is an increase in the isotope inventory by 1.4 watts, as shown in the second column of Table 3.2.5-5.

Table 3.2.5-5. Summary of Hybrid System Performance
with Alternate Insulation Techniques

	Fibrous Insulation		Superinsulation	
	Min-K (Xenon/Argon)	Min-K (Argon)	Ideal	Most Probable
Heat Loss	6.0 watts	7.4 watts	0.2 watt	4.4 watts
Isotope Inventory	49.0 watts	50.4 watts	43.2 watts	47.4 watts
Converter				
Diameter	3.5 inches	3.5 inches	2.6 inches	2.6 inches
Length	6.9 inches	6.9 inches	6.65 inches	6.9 inches
System Volume	1.33 liters	1.33 liters	0.86 liter	0.88 liter
System Weight	4.03 lb (1.83 kg)	4.04 lb (1.83 kg)	3.61 lb (1.64 kg)	3.68 lb (1.67 kg)
Specific Gravity	1.37	1.37	1.91	1.90

Thermal analyses were also carried out to determine the implications of using Linde composite superinsulation, which consists of Mo, Ni, Cu, and Al foils depending upon the temperature regime. Two cases were examined. The first was the ideal case which assumes no edge, joint, or penetration losses, as well as no heat losses through structural supports. This reduces the heat loss to 0.2 watt, compared to the Min-K (xenon/argon) heat loss of 6.0 watts. However, based on past experience with superinsulation, the second case or most probable value for the heat losses will be about 4.4 watts (1.0 watt through the insulation and 3.4 watts through the structural supports for the engine components). Therefore, the savings in thermal inventory for a superinsulation system will be on the order of 2 watts. The chief advantage of using a superinsulation system lies in the volume and weight savings which amount to 34% and 9%, respectively. As might be expected, however, from the disparity between the volumetric and weight reductions, the specific gravity increases from 1.37 to 1.90.

3.2.5.5 Safety

The basic safety philosophy dictates that the isotope fuel must be contained under all normal operating environments as well as during system malfunctions which reduce or terminate the active removal of heat from the heat source. Under normal operation, the heat source temperature is 1970°F. If the thermoelectric module fails (open circuit) the heat source temperature rises to 2450°F because of the loss of Peltier cooling. Should both thermoelectric module and turbogenerator unit fail, the heat source temperature would theoretically rise to 3400°F.* To assure containment under this condition, a thermal fuse was incorporated to assure adequate heat removal after rotary vapor engine failure. The thermal fuse contributes negligible additional heat loss during normal operations.

*Because of the degradation of the thermal insulation properties at temperatures above 2500°F, the peak temperature even without some sort of over-temperature protection would probably be considerably below 3400°F.

As will be recalled, the heat source is surrounded by a thermoelectric module and a high temperature thermal insulation system of Min-K 2020, filled with an inert gas. The thermal insulation is designed to fit snugly around the heat source and the thermoelectric module and to fill the gap between the turbogenerator boiler unit and the surrounding heat engine container. The thermal fuse is located between the boiler and outer structure. A number of small-diameter, non-radial holes through the Min-K insulation are filled with a lead-tin compound (solder) and Kaowool, a type of rock-wool insulation. The solder is mounted at the end closest to the boiler so that if the temperature of the boiler exceeds the melting point of the solder (600°F), the plug will melt and flow outward through the Kaowool insulation to form a high-conductivity fin that conducts heat from the boiler to the outer structure. This type of radio-isotope overtemperature protection has been considered for similar applications employing Min-K insulation. (Reference 22)

Analyses have indicated that a steady-state heat source temperature of less than 2600°F can be maintained. To meet these thermal conditions, four plugs, 0.125 inch in diameter are required. However, twice this number (eight) are used to assure independence of orientation. The additional heat loss through these plugs under normal operating conditions is less than 0.2 watt. While this approach appears feasible, experimental verification will be necessary because of the lack of thermal conductivity data for the Min-K 2020 above 2000°F .

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4. RELIABILITY OF THE CANDIDATE COMPONENTS

There are two primary failure mechanisms to be considered in a reliability analysis. The first is premature failure which occurs prior to wear-out of the component, and the second is wear-out itself. Premature failures occur randomly in time from predictable causes. After the device has reached design maturity, the causes of failure are either associated with (1) statistical occurrence of stresses exceeding the strength capabilities of the device, or (2) device defects which escape the quality control check points. If the design constraints on the device weight and volume are sufficiently generous, the susceptibility to the statistical occurrence of applied stress exceeding the material strength capabilities can be made insignificantly small. However, there are few situations where there are effectively no limits on design weight and volume. Similarly, the probability of a device escaping quality control check points is related to the amount of money that can be allocated for this activity. Given sufficient money, a device can be subjected to repeated nondestructive test techniques, metallurgical examinations, physical property measurements, radiographic inspection, etc., and premature failures caused by undetected quality problems can be reduced to a negligible level. But again, the funds available for inspecting each and every component are finite and therefore premature failure rates will also be finite.

The reliability analysis effort on the candidate systems therefore requires assessment of the susceptibility of the designs to premature failure. One technique consists of obtaining data on the number of failures that occur as a function of cumulative test time on components similar in configuration to those being utilized in the design. Data on many generic components were published by Avco Corporation in their Reliability Engineering Data Series, in 1962. These are still useful but a more recent data source, which is updated approximately 2 to 4 times yearly, is the "FARADA" or Failure Rate Data Handbook, which is published by the U. S. Naval Fleet Missile Systems Analysis and Evaluation Group, Corona, California. This collection includes failure rate data on man-rated equipment or components similar in design concept, if not size, to many of the components used in the candidate thermal converters. Where size,

operating environments, and other application data for the components are similar, the FARADA data are preferred. However, comparable FARADA components are not always available and few FARADA components are subjected to 10 years of uninterrupted use without maintenance. Therefore considerable effort was expended to obtain appropriate component failure rate data.

In order to reflect continued state-of-the-art advances in design, manufacturing and quality control, the lowest failure rate value was chosen in the FARADA tables when multiple failure rate data were available or, in the case of the Avco Data Series, the lower limit failure rate was chosen. This choice reflects the expected reliability improvements that will occur over the next 3 to 5 years.

It should also be pointed out that the premature failure reliability model deals only with the probability of observing a catastrophic, or sudden thermal converter failure.

Since most of the data are based on premature failure rates for components which have been tested for less than 10 years, the accuracy of the reliability does have some uncertainty on an absolute basis. However we believe that the comparative reliability predictions are sufficiently accurate to permit a valid relative comparison of the candidate systems.

4.1 PREMATURE FAILURE RELIABILITY MODELING

The most universally accepted measure of a component part reliability is the failure rate. A review of the failure rates used by different companies on different projects reveals a wide range of variations for each component part. This variation in failure rate is related not only to the manner in which a component is designed, manufactured, and inspected but also to the manner in which a component is applied in a system and the environment in which it is operated. As might be expected, these factors result in the assignment of different failure rates by different users to the same part.

Several detailed analyses have been performed to normalize the differences in failure rates for a generic class of components by correlating installation environment and observed failure rate utilizing a large

observed population. The correlations do not permit an independent determination of the degree to which different factors make up the installation environment, i.e., shock, vibration, temperature, and humidity. However, they do permit the determination of characteristic factors (K factors), which, when multiplied by the generic failure rate, yield the predicted failure rates for the different installation environments. Typical K factors are presented in Table 4-1. (Reference 23)

When obtaining a failure rate from the FARADA tables, or other similar data sets, it is necessary to adjust the source data to fit the new environmental application and therefore an environmental K factor for the thermal converter application is required. Table 4-2 compares the temperature, vibration, and shock environments generally associated with the operating modes presented in the table below.

Table 4-1. Results of Correlation Study

Mechanical/Electromechanical Component Operating Mode	Average K Factor
Satellite in Orbit	1
Computer	1
Ground Equipment	8
Shipboard Equipment	15
Rail Mounted Equipment	22
Aircraft Equipment (bench test)	30
Missile Equipment (bench test)	40
Aircraft Equipment (in-flight)	50
Missile Equipment (in-flight)	900

Table 4-2. Comparison of Temperature, Vibration and Shock Environments

Operating Mode	Environments			
	K Factor	Vibration	Shock	Temperature and Variations
Satellite in Orbit	1	None	None	Controlled
Computer	1	None	None	Controlled
Ground Equipment	8	Mild	Moderate	Moderate/Uncontrolled
Shipboard Equipment	15	Moderate	Moderate	Moderate/Uncontrolled
Rail Mounted Equipment	22	Moderate	Heavy	Moderate/Uncontrolled
Aircraft (in-flight)	50	Moderate/Severe	Moderate/Severe	High/Uncontrolled

The thermal converters will be subjected to mild vibrations, moderate-to-high shock, and moderate, but controlled temperatures (depending on the component). This environment has therefore been assigned a K factor of 5, intermediate between the computer and ground equipment environments.

When using the FARADA failure rate or other source data in the reliability models developed for each of the candidate systems, the reliability prediction results are described as "reliability best estimates" and are assumed to be equivalent to 50% statistical confidence level predictions. This statistical confidence level is not to be confused with the engineering confidence that is ascribed to the reliability predictions (see Section 4.4) since for many of the components utilized in the various candidate systems, no directly comparable failure rate data exist in the appropriate size and weight range. Therefore, adjustments are made to the failure rate data to reflect the utilization of smaller-sized or lighter-weight components. A discussion of these parts, including failure rate data accumulated from other than the FARADA, Avco Data Series, or MIL-HDBK-217A data sources, is presented below.

4.1.1 Vapor/Gas Bearings

One of the best sources for failure rate data on vapor/gas bearings are the Minuteman weapons system guidance and control packages which includes several bearings that operate continuously in the launch silo. Of the several gyro applications in the Minuteman, the 16,000 rpm pendulus integration gyroscope spool bearing, with a shaft diameter of 0.25 inch and a length of 0.43 inch, is very close in size and application to the vapor bearings proposed for the rotary engine. Table 4-3 presents the use history on these bearings.

Table 4-3. Bearing Use History

Number of Units	Average Continuous Operating Time	Longest Continuous Operating Time	Failure Rate (after spin-up)	10 yr Reliability
160 plus	4 yr	6 yr +	0.00035/1000 hr	0.94

Approximately 90 to 95% of all bearing failures occur during the spin-up of the bearing. The remaining 5% of failures are believed to be caused by internally generated contaminants such as off-gassing of potting compounds. Detailed examination of failure history indicates the failures to be random in time with no indication of wear-out. This is consistent with the fact that there is no metal-to-metal contact in the bearing and therefore 10-year life appears achievable. Several other data sources were investigated and while failure rate data were not available, maximum operating times are shown below:

<u>Gas Bearing Application</u>	<u>RPM</u>	<u>Demonstrated Life</u>
Minuteman Helium Blower	24,000	30,900 hr
Brayton Cycle Alternators	48,000	8,000 + hr
Cooling Turbine	6,000 to 73,000	≈ 7,000 hr 6,000 hr minimum
Turboalternator	180,000 to 220,000	6,000 hr minimum

These data for gas bearings are for discontinuous operation and therefore represent a more stressful application than a continuous service application of the same total accumulated service time.

4.1.2 Expansion Turbine

Review of the FARADA data was made for the most applicable turbine failure rate data. Hot gas turbine data were considered inappropriate for analysis because the rotary vapor turbine will be subjected to a peak temperature of only 425° F using the CP- 34 working fluid. The failure rate data chosen for the rotary vapor turbine are based on commercial aircraft (DC- 9) experience but adjusted as shown below:

<u>Component</u>	<u>Operating Time Total</u>	<u>No. of Failures</u>	<u>Unadjusted Failure Rate</u>	<u>Adjusted Failure Rate</u>
DC- 9 Cooling Turbine	561,670 hr	24	44.5×10^{-6} F/hr	0.36×10^{-6} F/hr

The adjusted failure rate is based on a review of the failure modes experienced by turbine assemblies as summarized below:

<u>Failure Mode</u>	<u>Approximate Percentage of All Failures (%)</u>
Damage (ingestion of foreign matter)	40
Bearing failure	5
Lubrication	2
Overspeed	20
Structural failure	25
Leakage and impeller wear	8
Total	100

Since a thermal converter turbine will not be subject to the ingestion of foreign matter, operator-induced inadvertent overspeed conditions, or bearing/lubrication failures, roughly 67% of the above failures experienced on the DC- 9 turbines are not applicable. Of the remaining 33%, 25% of the

failures are structural in nature and therefore related to the turbine blade stresses which are proportional to blade speed. Since the rotary vapor turbine tip velocity is 470 ft/sec as compared to the 1200 to 1500 ft/sec velocity of aircraft turbines, the failure probability caused by excessive stress conditions becomes negligible as shown below:

Essentially then the only failure mode that the rotary vapor turbine might be subjected to is the remaining 8% of the aircraft turbine failures attributable to "leakage and impeller wear." With the environmental application factor of 50 for an aircraft turbine and an application factor of 5 for the AHD thermal converter, the failure rate becomes:

$$\begin{aligned}
 \lambda_{\text{turbine thermal converter}} &= \lambda_{\text{turbine aircraft}} \times 8\% \times 5/50 \\
 &= 44.5 \times 10^{-6} \text{ F/hr} \times 0.08 \times 1/10 \\
 &= 0.36 \times 10^{-6} \text{ F/hr}
 \end{aligned}$$

4.1.3 Bellow Seals and Pumps

FARADA data, in the case of bellows seals and pumps, are inadequate both in numbers of generic bellows tests and types of bellows (non-metallic versus hydro-formed versus welded convolutions). Welded bellows are considered the only acceptable design for 10-year life. Therefore bellows manufacturers were contacted to determine the amount of long cycle life data available. Table 4-4 summarizes the test results on four different welded bellows designs.

Table 4-4. Test Results of Four Different Welded Bellows Designs

Bellows Design	No. of Units	Total Test (hrs)	Total No. of Cycles (millions)	No. of Failures	Avg. No. of Cycles Per Unit* (millions)
MB-21	18	61,581	11,100	0	616
MB-41	2	11,419	2,100	0	1,050
MB-150	10	85,461	8,800	0	880
MB-151	5	24,504	2,500	0	500

*Synchronous engine designs require a minimum of 630 million cycles for a 10-year life.

To achieve the cycle life shown in Table 4-4, two important requirements, besides limiting maximum allowable cyclic stresses, must be met. First, the alignment of the longitudinal axis of the top and bottom bellows convolutions must be maintained within 0.005 inch and, second, the bellows must be subjected to a 1-million cycle quality control "burn-in" to detect incipient bellows defects.

4.1.4 Precision Ball/Hydrodynamic Sleeve Bearings

Precision ball bearing and hydrodynamic sleeve bearings are both inferior, from the reliability point of view, to the gas or vapor bearings discussed earlier. The predicted design fatigue life on ball bearings is determined by using AFBMA standard analysis techniques using the equivalent radial loads, dynamic loads, operating speeds, etc. While the design fatigue life is considered to be the estimated life which will be exceeded by 90% of a group of identical bearings operating under identical load conditions (assuming proper mounting, lubrication and protection), the average fatigue life is approximately five times this figure. Thus the variability or uncertainty in the prediction of bearing life is very high and in order to predict design fatigue life at a 98 or 99% reliability level rather than the 90% level, the predicted life drops rapidly. For the thermal converter application, if the predicted life is 100 years based on a 90%-survival, the predicted life drops to ≈ 20 years for a 98% probability of

survival. In order to keep the radial loading down to meet the 100-year life at 90% reliability, bearing sizes for a 20-pound load are quite large (0.75 inch OD).

For the premature failure of bearings the FARADA tables include data on several bearing applications as shown below:

Bearing Application	Environment	Cumulative Test Time (hrs)	No. of Failures	50% Confidence Level Failure Rate (F/hr)
Attitude Instrument, Miniature Precision	Aircraft	5.65×10^6	0	0.123×10^{-6}
Bombing Computer, Miniature Precision	Aircraft	2.56×10^6	9	3.52×10^{-6}
Spin-Motor, 2400 rpm	Aircraft	1.01×10^6	14	14.8×10^{-6}
Ball Bearing	Submarine	1073×10^6	9	0.0083×10^{-6}

The failure modes experienced by the bearings used in the Bombing Computer are given below:

Frozen (seized)	50%
Vibration excessive	21%
Overheated	7%
Preload lost	7%
Worn	7%
Brinelled	7%

It should be noted that the above failures did not occur during the operational use of the equipment but during predelivery flight test experience and therefore the above failures should not in the normal sense, constitute bearing wear-out but rather premature bearing failures.

The gas bearings discussed earlier, which have demonstrated a reliability greater than 94% and no wear-out after more than 6 years of continuous service, have a significantly better reliability record than precision ball bearings.

In conclusion, premature ball bearing failures as discussed above can be assessed for reliability, but life predictions are sensitive to the actual bearing installation details.

4.1.5 Electronic Components and Solid State Devices

Numerous sources of failure rate data exist for the application of electronic parts to different equipments. Temperature environments are not severe for these parts in the typical thermal converter application but to reflect the high shock loading requirements, an application factor of 5 has been applied to the basic laboratory failure rate data compiled by Martin-Marietta Corp. This probably represents some degree of conservatism in the reliability modeling of the candidate systems.

Generic life expectancy of electronic parts such as diodes and transistors is predicted as being 200,000 hours. (Reference 24) More recent experience on Minuteman Missile Systems using Hi-Rel electronic parts subjected to burn-in and parameter drift screening indicates that generic life expectancy of capacitors, resistors, etc., is in excess of 200,000 hours. Based on the above data, wear-out of electronic parts in the present application is not considered a problem.

4.1.6 Gear Boxes/Speed Reducers

A significant quantity of failure rate data exists on gear assemblies and can be used to assess the motor reciprocator reliability against premature failure. The data shown in Table 4-5 were extracted from the FARADA tables for those assemblies that had accrued significant amounts of test data on large unit sample sizes.

The estimated life of a gear train is a function of the number of load cycles, torque levels, tooth sliding speeds, lubrication, material, etc., and as with precision ball bearings, depends on the actual design details of the gear box. Figure 32.22 of Mechanical Design and Systems Handbook, page 38, (Reference 24) indicated that gear life of 10^{12} cycles is obtainable provided gear bending and contact stresses are controlled. A particularly

Table 4-5. Sample Assembly Test Data from FARADA Tables

Application	No. of Units	Total Test (hr)	No. of Failures	Application Factor	Adjusted Failure Rate (F/hr)	10-Year Reliability
Aircraft Radar Gear Box	414	2.33×10^6	1	1/5	0.085×10^{-6}	0.992
Air Conditioning Gears - Grd.	738	3.19×10^6	1	5/8	0.019×10^{-6}	0.998
Blower Fan Gears - Grd	1200	5.18×10^6	1	5/8	0.012×10^{-6}	0.999
Air Compressor Gears - Grd	612	2.64×10^6	0	5/8	0.16×10^{-6}	0.9986

important consideration for long gear box life is accuracy of manufacturing and meshing and, therefore, rigidity of the gears, gear shaft, bearings and gear box. Further subjective evidence of achieving a 10-year life for gear boxes is offered in the common wrist watch, which by using high-precision lightly-loaded gears, can achieve long life, even without the benefit of an hermetically sealed environment.

4.1.7 High Energy Density Batteries

The two candidate high energy density batteries are the lithium-selenium cell which operates at 710°F and the sodium-sulfur cell which operates at 570°F. The lithium-selenium cell has had the benefit of a longer development time, but several failure modes remain to be controlled for a 10-year lifetime goal. The first problem is associated with the small relative size of the lithium ion (as compared to sodium and sulfur ions) which aggravates seal and corrosion-control problems. Another problem is the use of the semi-fused salt separator and liquid transport of the reacting ions. With the semi-fused salt separator it is not known what degree of capacity loss may occur during numerous charge/discharge cycles and the resulting poisoning or chemical short circuiting of the reacting elements.

The sodium-sulfur cell, while less advanced than the lithium-selenium cell, would appear to have several advantages. The first advantage is associated with the reduced operating temperature and correspondingly reduced materials compatibility problems. Ceramic seals and materials

compatibility technology are well developed for today's sodium-vapor lamps. Additional applicable technology is being developed for thermionic applications at temperatures as high as 700°C. In addition, since no semi-fused salt separator is required, little if any battery capacity loss is expected in the sodium-sulfur cell as a function of the number of charge/discharge cycles. Additional development effort is required in the ceramic separator and the beta alumina seal areas at this stage of sodium-sulfur battery technology. In summary, while no reliability data on cell life and premature failure exist at this time on either concept, a review of the possible cell failure modes indicates the sodium-sulfur cell to be a potentially higher reliability battery design.

4.1.8 PCU Circular Drive Cam

Life predictions on the drum-cam and follower system will depend on specific design details but analogies available in the transportation industry indicate that the typical automobile engine cam lobe turns approximately:

$$1/2 \times 1500 \frac{\text{revs}}{\text{min}} \times \frac{60}{\text{hr}} \times \frac{1500 \text{ hr}}{60,000 \text{ mi}} = 0.7 \times 10^8 \text{ revs}/60,000 \text{ mi}$$

For the thermal converter application the total number of revolutions (at significantly lower cam lobe surfaces stresses and rubbing velocities) is:

$$\frac{50 \text{ revs}}{\text{min}} \times \frac{60 \text{ min}}{\text{hr}} \times \frac{87,400 \text{ hr}}{10 \text{ yr}} = 2.6 \times 10^8 \text{ revs}/10 \text{ yr}$$

Thus 10-year life for a drum-cam appears achievable.

The FARADA failure rate data for premature cam and follower failures on large part populations is summarized below:

<u>Application</u>	<u>Total Hr</u>	<u>No. of Units</u>	<u>No. of Failures</u>	<u>Failure Rate (F/hr)</u>
Truck, AF	5.05×10^6	1171	1	0.19×10^{-6}
Truck, AF	3.18×10^6	738	1	0.31×10^{-6}
Compressor Air Conditioning	1.48×10^6	343	0	0.47×10^{-6} @50% C. L.

Since the above engine applications have 12 cam lobes for a 6-cylinder engine using one cam shaft, the failure rate for the thermal converter cam is equal to 1/6 the engine cam failure rate; or $1/6 \times 0.19 \times 10^{-6} = 0.031 \times 10^{-6}$ F/hr.

4.2 SUMMARY OF THE PREMATURE FAILURE RELIABILITY ESTIMATES FOR THE CANDIDATE SYSTEMS

4.2.1 System Reliability Math Model Results

The final premature failure reliability modeling results for the 8 candidate systems together with a brief description of the subsystems is shown in Table 4-6. System reliabilities vary from a high of 0.69 for the hybrid without battery to a low of 0.49 for the gas reciprocating with TESM. The following paragraphs discuss the systems in more detail.

4.2.2 Hybrid Thermal Converter

Prior to initiating the hybrid system reliability math modeling it was necessary to select the optimum electrical network for the two power sources. A review of the failure modes for both the turbogenerator and thermoelectric converter was made and indicated that approximately 2/3 of all turbogenerator failures resulted in open circuit or equivalent open circuit failures and the remaining 1/3 of the failures were short circuit. Similarly, thermoelectric converters of the spring-loaded element design rarely fail short-circuit and, given a total system failure, fail open-circuit. Therefore the two power sources were connected electrically in parallel to protect against open circuit failures and blocking diodes were used to protect against the alternator failing short-circuit. With this configuration the optimum reliability configuration was achieved. With 30 SiGe and sixty 2N/TAGS couples the hybrid engine reliability against catastrophic failure is predicted as 0.94 and is shown together with the electronic components required for the thermoelectric dc-to-dc voltage stepup converter in Table 4-7. Reliability modeling of the electrical PCU system for the hybrid engines is shown in Table 4-8 with the mechanical actuator shown in Table 4-9.

Table 4-6. Premature Failure Reliability

Candidate System	Engine	PCU	Actuator	Battery	TESM	Total
T/E with Battery	Two Parallel Redundant Strings of Couples 0.905	DC Motor, Planetary, Cam 0.80	Mechanical (2M) 0.90	Na/S 0.90		0.58
Rotary Vapor with Battery	(Turbo Alternator) 0.85	DC Motor, Planetary, Cam 0.80	Mechanical (2M) 0.90	Na/S 0.90		0.55
Hybrid with Battery	Parallel Redundant Engines Fail-Open Protection 0.94	DC Motor, Planetary, Cam 0.80	Mechanical (2M) 0.90	Na/S 0.90		0.61
Hybrid	Parallel Redundant Engines Fail-Open Protection 0.94	DC Motor, Planetary, Cam 0.80	Mechanical (2M) 0.90			0.68
Gas Reciprocating with TESH	0.89	Fluid Timer, High and Low Accumulator 0.85	Hydraulic (1F) 0.66		LiF/NaF 0.99	0.49
Gas Reciprocating	0.89	Fluid Timer, High and Low Accumulator 0.85	Hydraulic (2F) 0.66			0.50
Linear Vapor with TESH	0.80	Electronic Oscillator 0.81	Mechanical (1M) and Fill Switch 0.88		LiF/NaF 0.99	0.57
Linear Vapor	0.80	Electronic Oscillator 0.81	Mechanical (2M) 0.90			0.58

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Table 4-7. Premature Failure Reliability Model for the Hybrid Engines

ADDITIONAL HYBRID COMPONENTS FOR VOLTAGE MATCHING

COMPONENT	NUMBER ITEMS	FAILURE RATE	PRIMARY FAILURE MODES		FAILURE RATE SOURCE & ENVIRONMENT	CUMULATIVE OPERATION TIME & FAILURES	ENVIRONMENTAL APPLICATION FACTOR RATIO	ADJUSTED FAILURE RATE
TRANSISTORS (DC PULSES)	2	0.010×10^{-6} F/HR	45% SHORT	55% OPEN	MARTIN-MARIETTA CORP.	--	5/1	0.10×10^{-6} F/HR
DIODE (OUTPUT)	2	0.006×10^{-6}	50%	50%	MARTIN-MARIETTA CORP.	--	5/1	0.06
TRANSFORMER, 4 WINDING	1	0.100×10^{-6}	--	--	MARTIN-MARIETTA CORP.	--	5/1	0.50
RESISTORS	3	0.004×10^{-6}	5%	95%	MARTIN-MARIETTA CORP.	--	5/1	0.06
CAPACITOR	3	0.004×10^{-6}	60%	40%	MARTIN-MARIETTA CORP.	--	5/1	0.09
INDICATORS	1	0.050×10^{-6} F/HR	--	--	MARTIN-MARIETTA CORP.	--	5/1	0.25

$$R_{DC-DC} = e^{-(0.10 + 0.06 + 0.50 + 0.06 + 0.090 + 0.25) \times 10^{-6} \times 8.76 \times 10^4} = e^{-1.06 \times 10^{-6} \times 8.76 \times 10^4} = e^{-9.3 \times 10^{-2}} = 0.911$$

$$R_{TE} = 1 - \left\{ 1 - e^{-(1.70 \times 10^{-6} \times 8.76 \times 10^4)} \right\}^2 \Bigg]^{12} \times \left[1 - \left\{ 1 - e^{-(0.48 \times 10^{-6} \times 8.76 \times 10^4)} \right\}^2 \right]^{30}$$

$$\begin{aligned} & \times e^{-(0.006 + 0.003 + 0.009) \times 10^{-6} \times 8.76 \times 10^4} \\ & = (0.981)^{12} \times (0.9984)^{30} \times 0.998 \\ & = 0.79 \times 0.974 \times 0.998 = 0.768 \end{aligned}$$

$$\begin{aligned} R_{TOTAL} &= 1 - (1 - R_{RE}) (1 - R_{TE} \times R_{DC-DC}) \\ &= 1 - (1 - 0.85) (1 - 0.768 \times 0.911) \\ &= 1 - (0.15) (1 - 0.70) = 1 - (0.15 \times 0.30) \\ &= 0.94 \end{aligned}$$

R_{DC-DC} = RELIABILITY OF VOLTAGE STEP-UP CONVERTER

R_{TE} = RELIABILITY OF T/E HYBRID CONVERTER

R_{RE} = RELIABILITY OF ROTARY VAPOR ENGINE

Table 4-8. Premature Failure Reliability Model for the T/E and Rotary Vapor Engine PCU

COMPONENT	NUMBER OF ITEMS	FAILURE RATE F/HR	PRIMARY FAILURE MODES	FAILURE RATE SOURCE & ENVIRONMENT	CUMULATIVE OPERATING TIMES & FAILURES	ENVIRONMENTAL APPLICATION FACTOR RATIO	ADJUSTED FAILURE RATE, F/HR
FRACTIONAL H.P. ELECTRIC MOTOR	1	0.31×10^{-6} F/HR	BURNED OPEN/SHORTED	FARADA 352F GRD	3.18×10^6 HRS, 1 F	5/8	0.19×10^{-6} F/HR
HYDRODYNAMIC BEARINGS	4	0.098×10^{-6} F/HR	SEIZURE	FARADA 352A GRD	10.1×10^6 HRS, 1 F	5/8	0.24×10^{-6} F/HR
SPUR GEAR SETS	4	0.19×10^{-6} F/HR	TOOTH FRACTURE	FARADA 384J GRD	5.18×10^6 HRS, 1 F	5/8	0.48×10^{-6} F/HR
CAM & FOLLOWER ASSY	1	0.19×10^{-6} F/HR	PREMATURE WEAROUT	FARADA 352A GRD	5.06×10^6 HRS, 1 F	5/8 X 1/12	0.01×10^{-6} F/HR
BELLOWS SEAL	1	0.0285×10^{-9} F/CY	LEAKAGE	METAL BELLOWS LAB DATA	24.5×10^9 CYCLES, 0 F	5/1	0.142×10^{-9} F/CYCLE @ 50% C.L.

$$\begin{aligned}
 R &= e^{-(0.19 + 0.24 + 0.48 + 0.01) \times 10^{-6} \times 8.76 \times 10^4} \times e^{-(0.142 \times 10^{-9} \times 630 \times 10^6)} \\
 &= e^{-0.92 \times 8.76 \times 10^{-2}} \times e^{-89.5 \times 10^{-3}} \\
 &= 0.922 \times 0.91 = 0.84
 \end{aligned}$$

ELECTRONIC COMMUTATION COMPONENTS								
TRANSISTORS, NPN, (OSCILLATION)	2	0.010×10^{-6} F/HR	45% SHORT	55% OPEN	MARTIN MARIETTA CORP.	NOT APPLICABLE	5/1	0.10×10^{-6}
RESISTORS (OSCILLATION)	3	0.004×10^{-6}	5%	95%	MARTIN MARIETTA CORP.		5/1	0.060×10^{-6}
CAPACITOR (OSCILLATION)	1	0.004×10^{-6}	60%	40%	MARTIN MARIETTA CORP.		5/1	0.004×10^{-6}
TRANSISTORS NPN SWITCHING LOW POWER	4	0.010×10^{-6}	60%	40%	MARTIN MARIETTA CORP.		5/1	0.20×10^{-6}
PC BOARD	1	--	--	--	--		--	--
LOW TUNING COILS	2	0.010						0.10×10^{-6}
IC DECODE COUNTER	1	0.30×10^{-6}	60%	40%	MARTIN MARIETTA CORP.		5/1	0.15×10^{-6}

$$\begin{aligned}
 R &= e^{-\epsilon \lambda t} = e^{-(0.614 \times 10^{-6} \times 8.76 \times 10^4)} = e^{-0.54 \times 10^{-2}} = 0.95 \\
 R_{\text{TOTAL PCU}} &= 0.848 \times 0.95 = 0.80
 \end{aligned}$$

Table 4-9. Premature Failure Reliability Model For Actuator
(Electrical Systems)

COMPONENT	NUMBER OF ITEMS	FAILURE RATE F/HR	PRIMARY FAILURE MODES	FAILURE RATE SOURCE & ENVIRONMENT	CUMULATIVE OPERATING TIMES & FAILURES	ENVIRONMENTAL APPLICATION FACTOR RATIO	ADJUSTED FAILURE RATE, FAILURES/HR
DIAPHRAGMS (REDUNDANT)	2	0.6×10^{-6}	TEARS - LEAKAGE	AVCO DATA SERIES	--	5/1	3.0×10^{-6} F/HR/DIA
COMPLIANCE BAG	1	0.1×10^{-6}	TEARS - LEAKAGE	AVCO (ADJUSTED)	--	5/1	0.5×10^{-6}
SPRING	1	0.03×10^{-6}	FATIGUE/BREAKAGE	FARADA 373 CACFT	62.1×10^6 HRS, 2F	5/50	0.003×10^{-6}
CABLE ROD, SHEATH & ASSY.	1	0.002×10^{-6}	FATIGUE/BREAKAGE	AVCO DATE SERIES	--	5/1	0.01×10^{-6}
TITANIM HOUSING	1	NEGLIGABLE	--	--	--	--	--

$$\begin{aligned}
 R &= \left[1 - \left(1 - e^{-3.0 \times 10^{-6} \times 8.76 \times 10^4} \right)^2 \right] \times e^{-(0.5 \times 0.003 + 0.01) \times 10^{-6} \times 8.76 \times 10^4} \\
 &= 1 - (0.2305)^2 \times e^{-0.0446} \\
 &= 0.9468 \times 0.956 = 0.905
 \end{aligned}$$

PREMATURE FAILURE RELIABILITY MODEL FOR THE
LINEAR VAPOR ENGINE AND TESM ACTUATOR

SAME AS ABOVE EXCEPT FOR ADDITION OF ANTI VACUUM VALVE AND FILL SWITCH

COMPONENT	NUMBER OF ITEMS	FAILURE RATE F/HR	PRIMARY FAILURE MODES	FAILURE RATE SOURCE & ENVIRONMENT	CUMULATIVE OPERATING TIMES & FAILURES	ENVIRONMENTAL APPLICATION FACTOR RATIO	ADJUSTED FAILURE RATE, FAILURES/HR
FILL SWITCH	1	0.36×10^{-6} F/HR	OPEN/SHORT	FARADA 354 GRL	5.60×10^6 2F	5/18	0.225×10^{-6} F/HR
ANTI VACUUM VALVE	1	0.85×10^{-6} F/HR	TEARS, RUPTURE	FARADA 313 ACFT (FLAPPER VALVE)	1.195×10^6 , 1F	5/50	0.085×10^{-6}

$$\begin{aligned}
 R &= 0.905 \times e^{-0.315 \times 10^{-6} \times 8.76 \times 10^4} \\
 &= 0.905 \times e^{-2.75 \times 10^{-2}} \\
 &= 0.905 \times 0.973 = 0.88
 \end{aligned}$$

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In summary, the thermoelectric converter and rotary vapor engines together present an advantageous reliability picture as compared to the other candidate systems.

4.2.3 Hybrid Thermal Converter with Battery

From the reliability point of view and having designed the heat exchangers to prevent body tissue temperatures from exceeding 42°C under worst-case conditions, the presence of a sodium-sulfur battery decreased the overall system reliability, irrespective of how reliable the battery is. Although the initial degradation mode may be a loss of capacity or even open-circuit conditions for individual cells, the most likely ultimate failure is a system-level short circuit which is not readily isolated without a significant performance penalty. As discussed in paragraph 4.1.7 above, the sodium-sulfur battery failure modes were analyzed and from this information the reliability of the battery for 10 years (when developed) was estimated as 0.90. The redundancy contribution of the battery is small since in the event of engine failure, the several hours of survival time that would be available from battery power would be marginal from the standpoint of permitting remedial action.

4.2.4 Thermoelectric / Battery Thermal Converter

To help eliminate catastrophic open circuit converter failures, the thermoelectric modules are electrically connected in two parallel strings of couples for a total of 8 SiGe couples and 28 TAGS, BiSbTe/PbTe-GeTe, 2N cascaded couples. RCA data on SiGe modules and Snap 19, -21, and -23 data on the other couples were used to obtain the couple failure rates. Since both the RCA and Snap 19 data revealed no open-couple failures to date, the reliability calculations were performed at the 50% confidence level. (This is a little more optimistic than the FARADA technique of assuming one failure for purposes of calculating a failure rate when in fact no failure has been observed. Using 50% confidence limits, this is equivalent to a 0.693 failure probability when in fact no failure has been observed.) As noted in the reliability math model shown in Table 4-10, limited test data exist for the PbTe-GeTe elements but based on the similarity of PbTe-GeTe failure modes to the 2N elements, the failure rate

Table 4-10. Premature Failure Reliability Model for Thermoelectric Cascaded Converter

COMPONENT	NUMBER ITEMS	FAILURE RATE F/HR	PRIMARY FAILURE MODES	FAILURE RATE SOURCE & ENVIRONMENT	CUMULATIVE OPERATING TIME & FAILURES	ENVIRONMENTAL APPLICATION FACTOR RATIO	ADJUSTED FAILURE RATE, F/HR
SiGe COUPLES	8	$\frac{0.693}{2.05 \times 10^6} @ 50\% C L$	OPEN CIRCUIT-SUBLIMATION	RCA LAB DATA	2 050 X 10 ⁶ HRS, 0 F	5/1	1 70 X 10 ⁻⁶ F/HR/ COUPLE
TAGS (ELEMENTS)	28	$\frac{0.693}{7.28 \times 10^6} @ 50\% C L$	OPEN CIRCUIT-SUBLIMATION RESISTANCE	SNAP 19-LAB	7 28 X 10 ⁶ HRS, 0 F	5/1	0 48 X 10 ⁻⁶ F/HR/ ELEMENT
BiSbTe (ELEMENTS)	28	0 10 X 10 ⁻⁶ F/HR	OPEN CIRCUIT-SUBLIMATION RESISTANCE	SNAP 21 & 23	20 5 X 10 ⁶ HRS, 2 F	5/1	0 504 X 10 ⁻⁶ F/HR/ ELEMENT
PbTe-GeTe (ELEMENTS)	28	N/A	OPEN CIRCUIT-SUBLIMATION RESISTANCE	RCA (LIMITED) DATA	--	5 1	*
2N (ELEMENTS)	28	$\frac{0.693}{7.28 \times 10^6} @ 50\% C L$	OPEN CIRCUIT-SUBLIMATION RESISTANCE	SNAP 19-LAB	7 28 X 10 ⁶ HRS, 0 F	5/1	0 48 X 10 ⁻⁶ F/HR/ ELEMENT
MIN-K INSULATION	-	0 0006 F/IN-WELD	ARGON LEAKAGE	TRW SPACE FLIGHTS	--	15' OF WELD	0 009 X 10 ⁻⁶ F/HR
HOT & COLD SHOES	72	--	INTERACTION WITH T/E ELEMENTS	--	--	--	0 009 X 10 ⁻⁶ F/HR
SPRING LOADING PLATE	2	0 03 X 10 ⁻⁶ F/HR	LOSS OF PRELOAD, CREEP	FARADA 373C ACFT	62 1 X 10 ⁶ , 2 FAILURES	5/50	0 006 X 10 ⁻⁶ F/HR
ELECTRICAL CONNECTIONS	72	0 00001 X 10 ⁻⁶ F/HR	OPEN CIRCUIT	MARTIN MARIETTA CORP	--	5/1	0 0036 X 10 ⁻⁶ F/HR

$$R = \left[1 - \left\{ 1 - e^{-(1.70 \times 10^{-6} \times 8.76 \times 10^4)} \right\}^{27} \right]^4 \times \left[1 - \left\{ 1 - e^{-(0.504 \times 10^{-6} \times 8.76 \times 10^4)} \right\}^{27} \right]^{14} \times e^{-(0.006 + 0.0036 + 0.009) \times 10^{-6} \times 8.76 \times 10^4}$$

$$= (0.981)^4 \times (0.9984)^{14} \times 0.998$$

$$0.926 \times 0.9778 \times 0.998 = 0.904$$

*OPEN CIRCUIT FAILURE RATE ON THE PbTe-GeTe ELEMENTS IS ASSUMED EQUAL TO THE SNAP 19, 2n MATERIAL SINCE THE RCA LAB DATA ON PbTe-GeTe ELEMENTS IS EXTREMELY LIMITED

for the 2N material was assumed. On this basis the failure rate for the cascaded portion of thermoelectric converter is assumed equal to the element portion exhibiting the highest failure rate, namely the BiSbTe elements, and the reliability modeling reflects this. While the above assumptions result in lack of complete engineering confidence in the reliability estimate, similar assumptions were required in the modeling of the other candidate systems. The degree of confidence in the assumptions is reflected in the Section 4.4 discussion.

4.2.5 Rotary Vapor Engine

The reliability model for this engine is as shown in Table 4-11. With the exception of the turbine failure rate data which are discussed in section 4.1.2, little adjustment of the component failure rates was required. This results from the fact that this engine is generically well developed and the confidence in the reliability estimates (excluding the thermal battery) is therefore almost as high as for the candidate thermoelectric system

4.2.6 Electrical PCU

Table 4-8 shows the reliability model for the electrical PCU, consisting of a pancake dc electronically commutated motor, magnetic coupling, compound planetary gear drive, and drum-cam reciprocator. The gas bearings are similar to those used in the turbogenerator except they will operate at lower temperatures and rotational speeds. The FARADA data for hydrodynamic bearings and the Minuteman Missile gyroscope bearing data can be used to generate the reliability estimates. As with the turbogenerator, the confidence in the reliability predictions for the electrical units is higher than for the hydraulic power conditioning unit. No life-limiting ball bearings are used. Specific design features such as the use of lubrication for the speed reducer and cam, conservatively stressed parts, and the hermetic sealing of the dc motor, assure long life.

4.2.7 Mechanical Actuator for Electrical Systems

The reliability modeling for the electrical system automatic blood pump actuator (as well as the actuator for the linear vapor engine with TESM) is shown in Table 4-8. Fortunately the actuator diaphragms are

Table 4-11. Premature Failure Reliability Model for Rotary Vapor Heat Engine

COMPONENT	NUMBER OF ITEMS	FAILURE RATE F/HR	PRIMARY FAILURE MODES	FAILURE RATE SOURCE & ENVIRONMENT	CUMULATIVE OPERATING TIMES & FAILURES	ENVIRONMENTAL APPLICATION FACTOR RATIO	ADJUSTED FAILURE RATE, F/HR
EXPANSION TURBINE RADIAL INFLOW IMPULSE TYPE	1	29.9×10^{-6}	STRUCTURAL/CAVITATION & EROSION	FARADA 294 ACFT	0.0669×10^6 HRS, 2 FAILURES	$5/50 \times 1/4^*$	$0.36 \times 10^{-6}^*$
HELICAL SCROLL PUMP	1	0.31×10^{-6}	STRUCTURAL/LEAKAGE	FARADA 352 GRD	3.188×10^6 HRS, 1 FAILURE	5/8	0.19×10^{-6}
DC ALTERNATOR, PERM. MAGNETIC	1	$\frac{0.693}{1.29 \times 10^6} @ 50\% \text{ C.L.}$	66% OPEN, 34% SHORTED	FARADA 384J GRD	1.29×10^6 HRS, 0 FAILURES	5/8	0.338×10^{-6}
ZENOR DIODE, RECTIFIER	4	0.054×10^{-6}	SHORTING 75%, OPEN 25%	FARADA 355 GRD	$110. \times 10^6$ HRS, 6 FAILURES	5/8	0.034×10^{-6}
VAPOR GAS BEARING	2	0.35×10^{-6}	SEIZURE VIA CONTAMINATION	MINUTEMAN II & III	5.7×10^6 HRS, 2 FAILURES	$1/1^{**}$	$0.70 \times 10^{-6}^{**}$
BOILER & PRE-HEATER (TUBING)	1	0.09×10^{-6}	RUPTURE/LEAKAGE	FARADA 352K GRD	10.78×10^6 HRS, 1 FAILURE	5/8	0.056×10^{-6}
REGENERATOR	1	$\frac{0.693}{0.7221 \times 10^6} @ 50\% \text{ C.L.}$	LEAKAGE	FARADA 292 ACFT	0.7221×10^6 HRS, 0 FAILURES	5/50	0.0965×10^{-6}
CONDENSER	1	0.065×10^{-6}	LEAKAGE	FARADA 352J GRD	15.2×10^6 HRS, 1 FAILURE	5/8	0.0406×10^{-6}
INSULATION SYSTEM	1	0.0006 F/IN-WELD	VACUUM LOSS	TRW SPACE FLIGHTS	--	15" OF WELD	0.009×10^{-6}

$$R = e^{-\sum \lambda t} = e^{-(0.36 + 0.19 + 0.338 + 0.034 + 0.70 + 0.056 + 0.0965 + 0.0406 + 0.009) \times 10^{-6} \times 87,600 \text{ HRS}} = e^{-1.82 \times 10^{-6} \times 8.76 \times 10^4}$$

$$= e^{-15.9 \times 10^{-2}} = 0.85$$

*SEE DISCUSSION IN SECTION 4.1.2

THE AHD TURBINE TIP SPEED IS APPROX. 450 FT/SEC COMPARED TO APPROX. 1200-1500 FT/SEC FOR AIRCRAFT ENGINE APPLICATIONS AND APPROX. 3/4 OF ALL TURBINE FAILURES ARE CAUSED BY OVERSPEED/INGESTION OF FOREIGN MATERIAL FOR AIRCRAFT APPLICATIONS.

**MINUTEMAN MISSILE II & III GUIDANCE AND CONTROL PIGA GYRO DATA - SILO TEMP. CONTROLLED CONDITIONS - MIN. TEMPERATURE VARIATION & VIBRATION EXPOSURE. THEREFORE APPLICATION FACTOR ASSUMED TO BE ~1 TO 1, SILO CONDITIONS TO HUMAN CONDITIONS.

redundant, for without this redundancy, the unreliability of the actuators would significantly degrade the total system reliability.

4.2.8 Gas Reciprocating Engine

The primary unreliability associated with the gas reciprocating engine rests with the reliability of the check valves and piston/displacer bearing surfaces. Since the helium working fluid provides no lubrication, it is necessary to use two hydrostatic bearings, one for the cylinder-to-piston surfaces and the other for the displacer/piston surfaces. Since the bellows pump is subjected to much smaller deflections (in the order of 0.10-inch) as compared to the bellows seal in the electrical PCU drive, the failure rate for the pump was reduced by the deflection ratio of 1/10. The reliability model for the gas reciprocating engine is shown in Table 4-12.

4.2.9 TESM

Since the LiF/NaF material is only subjected to a phase change in the process of releasing and storing thermal energy, the premature failure reliability contribution to the reliability math modeling of the system is considered negligible. Wearout reliability considerations dealing with materials compatibility problems are discussed in Section 4.3.

4.2.10 Gas Reciprocating Engine PCU

The reliability math model for this PCU is shown in Table 4-13. No particular reliability problems are envisioned for this PCU with the exception of the rotary fluid timer which must possess quite tight tolerances for the rotary plug valve in order to maintain reasonable system efficiency. These tolerances in turn adversely effect the PCU reliability.

4.2.11 Gas Reciprocating Engine Actuators

The reliability math models for these PCU's are shown in Table 4-14. The only difference between the two actuators is the presence or absence of the anti-vacuum valve. This valve is essentially a simple flapper valve and has a negligible reliability effect on the reliability math modeling. The one critical item in these actuators is the rolling diaphragm which must retain the fluid to actuate the shuttle valve piston. The rolling diaphragm, unlike the actuator diaphragm, is not redundant and the reliability modeling results reflects this lack of redundancy.

Table 4-12. Premature Failure Reliability Model for the Gas Reciprocating Engine

COMPONENT	NUMBER OF ITEMS	FAILURE RATE F/HR	PRIMARY FAILURE MODES	FAILURE RATE SOURCE & ENVIRONMENT	CUMULATIVE OPERATING TIME & FAILURES	ENVIRONMENTAL APPLICATION FACTOR RATIO	ADJUSTED FAILURE RATE, F/HR
DISPLACER & CYLINDER	1	0.16×10^{-6} F/HR	SEIZURE, EXCESS LEAKAGE	FARADA 352E GRD	5.93×10^6 HRS, 1 F	5/8	0.100×10^{-6}
PISTON, 2 BEARING SURFACES	1	0.16×10^{-6} F/HR	SEIZURE, EXCESS LEAKAGE	FARADA 352E GRD	5.93×10^6 HRS, 1 F	5/8	$2 (0.100 \times 10^{-6})$
DISPLACER SPRING	1	0.03×10^{-6}	FRACTURE	FARADA 373C ACFT	62.1×10^6 HRS, 2 FAILURES	5/50	0.003×10^{-6}
PISTON SPRINGS	2	0.03×10^{-6}	FRACTURE	FARADA 373C ACFT	62.1×10^6 HRS, 2 FAILURES	5/50	0.006×10^{-6}
CLEARANCE REGENERATOR	1	$\frac{0.693}{0.722 \times 10^6} @ 50\% C L$	LEAKAGE	FARADA 292 ACFT	10.78×10^6 HRS, 1 FAILURE	5/8	0.056×10^{-6}
CHECK VALVES	2	0.59×10^{-6}	LEAKAGE, STICTION	FARADA 292 ACFT	1.68×10^6 HRS, 1 F	5/50	0.118×10^{-6}
HEAT EXCHANGER	1	0.065×10^{-6}	LEAKAGE	FARADA 352J GRD	15.2×10^6 HRS, 1 F	5/8	0.041×10^{-6}
HYDROSTATIC BEARINGS & ORIFICES	2	0.010×10^{-6}	LEAKAGE, BACK FLOW	AVCO DATA SERIES	--	5/1	0.100×10^{-6}
MIN-K INSULATION, Xe Gas	1	0.0006×10^{-6} F/IN-WELD	SEAL LOSS	TRW SPACE FLIGHTS	--	15" OF WELD	0.009×10^{-6}
BELLOWS	1	0.0285×10^{-9} F/CY	LEAKAGE	METAL BELLOWS LAB DATA	24.5×10^9 CYCLES, 0 F	$5/1 \times 1/10^*$	0.0142×10^{-9} F/CYCLE @ 50% C L

$$\begin{aligned}
 R &= e^{-\epsilon \lambda t} = e^{-(0.100 + 0.200 + 0.003 + 0.006 + 0.056 + 0.118 + 0.041 + 0.100 + 0.009) \times 8.76 \times 10^4} \times e^{-0.0142 \times 10^{-9} (750 \times 60 \times 8.76 \times 10^4)} \\
 &= e^{-(0.633 \times 10^{-6}) 8.76 \times 10^4} \times e^{-0.0142 \times 10^{-9} \times 3.94 \times 10^9} \\
 &= 0.946 \times 0.945 \\
 &= 0.89
 \end{aligned}$$

*THE 5/1 APPLICATION FACTOR REPRESENTS THE DIFFERENCE IN ENVIRONMENTAL STRESSES BETWEEN THE AHD ENVIRONMENT AND THE BELLOWS TEST LABORATORY, WHILE THE 1/10 FACTOR REPRESENTS ADJUSTMENT FOR THE VERY LOW AHD BELLOWS STRESSES AS COMPARED TO HIGHER LABORATORY BELLOWS PUMP STRESSES

Table 4-13. Premature Failure Reliability Model for the Gas Reciprocating Engine PCU

COMPONENT	NUMBER OF ITEMS	FAILURE RATE	PRIMARY FAILURE MODES	FAILURE RATE SOURCE & ENVIRONMENT	CUMULATIVE OPERATING TIMES & FAILURES	ENVIRONMENTAL APPLICATION FACTOR RATIO	ADJUSTED FAILURE RATE
<u>FLUID TIMER</u>							
ROTARY PUMP	1	0.501×10^{-6} F/HR	SEIZURE/LEAKAGE	FARADA 201 ACFT	1.99×10^6 HRS, 1 F	5/50	0.0501×10^{-6} F/HR
GEAR SET	1	0.19×10^{-6} F/HR	TOOTH FRACTURE	FARADA 3845 GRD	5.18×10^6 HRS, 1 F	5/8	0.12×10^{-6} F/HR
HYDRODYNAMIC BEARINGS	4	0.098×10^{-6}	SEIZURE	FARADA 352A GRD	10.1×10^6 HRS, 1 F	5/8	0.24×10^{-6} F/HR
ROTARY PLUG VALVE	1	$\frac{0.693}{0.597 \times 10^6} @ 50\% C.L.$	SEIZURE/LEAKAGE	FARADA 313 ACFT	0.597×10^6 HRS, 0 F	5/50	0.117×10^{-6} F/HR
<u>ACCUMULATOR</u>							
LOW PRESSURE BELLOWS	1	0.0285×10^{-9} F/CY	LEAKAGE/RUPTURE	METAL BELLOWS LAB DATA	24.5×10^9 CYCLES, 0 F	5/1 X 1/10	0.0142×10^{-9} F/CY
HIGH PRESSURE BELLOWS	1	0.0285×10^{-9} F/CY	FRACTURE/FATIGUE	METAL BELLOWS LAB DATA	24.5×10^9 CYCLES, 0 F	5/1 X 1/10	0.0142×10^{-9} F/CY

$$\begin{aligned}
 R &= e^{-(0.050 + 0.12 + 0.24 + 0.117) \times 10^{-6} \times 8.76 \times 10^4} \times e^{-2(0.0142 \times 10^{-9}) \times 750 \times 60 \times 8.76 \times 10^4} \\
 &= e^{-0.527 \times 10^{-6} \times 8.76 \times 10^4} \times e^{-0.112} \\
 &= 0.955 \times 0.894 = 0.85
 \end{aligned}$$

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Table 4-14. Premature Failure Reliability Model for Actuator
(Gas Reciprocating Systems)

COMPONENT	NUMBER ITEMS	FAILURE RATE	PRIMARY FAILURE MODES	FAILURE RATE SOURCE	CUMULATIVE OPERATING TIME & FAILURES	ENVIRONMENTAL APPLICATION FACTOR RATIO	ADJUSTED FAILURE RATE, F/HR
SHUTTLE VALVE PISTON	1	$\frac{0.693}{0.24 \times 10^{-6}} @ 50\% C.L.$	STICTION/LEAKAGE	FARADA 292 ACFT	0.24×10^5 HR, OF	5/50	0.29×10^{-6} F/HR
ROLL DIAPHRAGM	1	0.6×10^{-6}	LEAKAGE/TEARS	AVCO DATA SERIES	--	5/1	3.0×10^{-6}
DIAPHRAGMS (REDUNDANT)	2	0.6×10^{-6}	LEAKAGE/TEARS	AVCO DATA SERIES	--	5/1	3.0×10^{-6} F/HR/DIA
COMPLIANCE BAG	1	0.1×10^{-6}	LEAKAGE/TEARS	AVCO (ADJUSTED)	--	5/1	0.5×10^{-6}
ANTI-VACUUM VALVE	1	0.85×10^{-6}	TEARS, RUPTURE	FARADA 313 ACFT	1.19×10^6 , 1F	5/50	0.085×10^{-6}
WET CONNEC- TORS (QUICK DISCONNECT)	2	$\frac{0.693}{1.399 \times 10^6} @ 50\% C.L.$	LEAKAGE	FARADA 250 ACFT	1.399×10^6 , OF	5/50	0.050×10^{-6} F/HR
		$\frac{0.693}{237 \text{ CYCLES}} @ 50\% C.L.$	LEAKAGE	FARADA 309A MSLE	237 CYCLES, OF	5/8	0.295×10^{-2} F/ CYCLE

$$\begin{aligned}
 R &= e^{-\{0.29 + 3.0 + 0.5 + 0.085 + (2 \times 0.05)\} \times 10^{-6} \times 8.76 \times 10^4} \times \left[1 - \left(1 - e^{-3.0 \times 10^{-6} \times 8.76 \times 10^4} \right)^2 \right] \times e^{-0.29 \times 10^{-2} \times 1} \\
 &= e^{-34.8 \times 10^{-2}} \times 1 - (0.2305)^2 \times 0.997 \\
 &= 0.706 \times 0.947 \times 0.997 \\
 &= 0.66
 \end{aligned}$$

*THE ANTI-VACUUM VALVE HAS AN INSIGNIFICANT EFFECT ON THE PREMATURE FAILURE RELIABILITY MODELS. THEREFORE THE ABOVE RELIABILITY ESTIMATE APPLIES TO BOTH THE GAS RECIPROCATING ACTUATOR USED WITH TESM AND THE GAS RECIPROCATING ACTUATOR USED WITHOUT TESM.

4.2.12 Linear Vapor Engine and PCU

The linear vapor engine with PCU is modeled in Table 4-15 and reflects the unreliability of using solenoid check valves and solenoid shuttle and feed water pump valves. Redundant solenoids and quad-redundant check valves would improve reliability significantly. However, system weight and volume, even with just the redundant solenoids, becomes prohibitive. No reliability estimation adjustments were required for the basically solid state electronics and SiGe thermoelectric module.

4.2.13 Linear Vapor Engine Actuator

The actuator for the linear vapor engine without TESM is the same as the actuator for the electrical systems and is modeled in Table 4-9. The actuator for the linear vapor engine with TESM is shown at the bottom of the same table and reflects the addition of an anti-vacuum valve, and fill switch sensor.

4.3 RELIABILITY AGAINST WEAROUT

Each of the candidate systems was reviewed to determine relative susceptibility of all their components to 14 different possible wearout failure modes. These 14 different modes were then grouped under 8 general categories shown in Table 4-16. Since the 8 categories do not have the same degree of susceptibility to wearout nor the same effect on system degradation and failure, the categories were given different ranges of possible reliability scores. For example, both long and short stroke springs have a small probability of causing system failure even if they do weaken. However, system seals or diaphragms have not only a higher probability of wearout, but a higher probability that wearout will cause system failure. Therefore the allocated range of reliability scores for springs is between 97% and 100%, while seals and diaphragms are allocated a range between 90% and 100%. The major subassemblies of a candidate system were reviewed for the numbers of components susceptible to the 8 wearout categories and given a relative score with respect to the other candidate systems. For example, the thermoelectric system with a battery has no dynamic seals and therefore receives a score of 100. In contrast, both the linear vapor and gas reciprocating engines have multiple dynamic seals which upon wearout can cause complete system

Table 4-15. Premature Failure Reliability Model for the Linear Vapor Engine and PCU

COMPONENT	NUMBER ITEMS	FAILURE RATE F/HR	PRIMARY FAILURE MODES	FAILURE RATE SOURCE & ENVIRONMENT	CUMULATIVE OPERATING TIMES & FAILURES	ENVIRONMENTAL APPLICATION FACTOR RATIO	ADJUSTED FAILURE RATE, F/HR
BOILER	1	0.09×10^{-6}	RUPTURE/LEAKAGE	FARADA 352K GRD	10.78×10^9 HRS, 1 F	5/8	0.056×10^{-6} F/HRS
SOLENOID SHUT-TLE SPOOL VALVE-2-POSITION	1	1.61×10^{-6}	JAMMING, LEAKAGE, BURNOUT	FARADA 407 GRD	2.488×10^6 HRS, 4 F	5/8	1.0×10^{-6} F/HRS
BELLOWS-VAPOR ENGINE	1	0.0285×10^{-9} F/CY	LEAKAGE	METAL BELLOWS LAB DATA	24.5×10^9 CYCLES, 0 F	5/1	0.142×10^{-9} F/CYCLE
BELLOWS-ACTUATOR ROD	1	0.0285×10^{-9} F/CY	LEAKAGE	METAL BELLOWS LAB DATA	24.5×10^9 CYCLES, 0 F	5/1	0.142×10^{-9} F/CYCLE
REGENERATOR	1	$\frac{0.693}{0.7221 \times 10^6} @ 50\% C L$	LEAKAGE	FARADA 292 ACFT	0.7221×10^6 HRS, 0 F	5/50	0.0965×10^{-6}
ELECTRIC FEED PUMP	1	1.61×10^{-6} F/HR	JAMMING, LEAKAGE, BURNOUT	FARADA 407 GRD	2.488×10^6 HRS, 1 F	5/8	1.0×10^{-6}
SOLENOID VALVE	1	0.03×10^{-6}	FATIGUE/BREAKAGE	FARADA 373C ACFT	62.1×10^6 HRS, 2 F	5/50	0.003×10^{-6}
CHECK VALVES	2	0.59×10^{-6} F/HR	STICTION/LEAKAGE	FARADA 292 ACFT	1.68×10^6 HRS, 1 F	5/50	0.118×10^{-6}
CONDENSER	1	0.065×10^{-6}	LEAKAGE	FARADA 352J GRD	15.2×10^6 HRS, 1 FAIL	5/8	0.0406×10^{-6}

$$R_{\text{Engine}} = e^{-(0.0564 + 1.0 + 0.142 + 0.096 + 1.04 + 0.0034 + 0.118 + 0.0406) \times 10^{-6} \times 8.76 \times 10^4} = e^{-2.6 \times 10^{-6} \times 8.76 \times 10^4} = e^{-22.8 \times 10^{-2}} = 0.796$$

VAPOR CYCLE ENGINE PCU

SiGe COUPLES (REDUNDANT)	14	$\frac{0.693}{2.05 \times 10^6} @ 50\% C L$	OPEN CIRCUIT/SUBLIMATION	RCA LAB DATA	2.05×10^6 HRS, 0 F	5/1	1.70×10^{-6} F/HR/COUPLE
ELECTRICAL CONNECTIONS	14	0.00001×10^{-6} F/HR	OPEN CIRCUIT	MARTIN MARIETTA CORP	--	5/1	0.0007×10^{-6} F/HR
OSCILLATOR - 120 CYCLE	1	2.33×10^{-6} F/HR	OPEN/SHORT	FARADA 364B SHIP	0.858×10^6 2 F	5/15	0.78×10^{-6} F/HR

$$R_{\text{PCU}} = \left[1 - \left\{ 1 - e^{-(1.70 \times 10^{-6} \times 8.76 \times 10^4)} \right\}^2 \right]^7 \times e^{-0.78 \times 10^{-6} \times 8.76 \times 10^4} = (0.981)^7 \times e^{-0.068} = 0.874 \times 0.93 = 0.813$$

$$R_{\text{TOTAL}} = 0.813 \times 0.796 = 0.65$$

Table 4-16. Reliability Against Wearout

Components Susceptible To Wear-Out	T/E with Battery	Rotary Vapor with Battery	Hybrid with Battery	Hybrid	Gas Reciprocating with TESM	Gas Reciprocating	Linear Vapor with TESM	Linear Vapor
Valves, Spool, Check Valve and Rotary Valves	100	100	100	100	95	95	90	90
Seals								
Dynamic	100	98	98	98	90	90	90	90
Static	95	95	95	100	100	100	95	100
Hermetic	98	98	98	98	98	98	98	98
Metal-to-Metal Contact								
Sliding Unlubricated	100	100	100	100	90	90	100	100
Sliding Lubricated	98	98	98	98	98	98	95	95
Rolling	99	99	99	99	99	99	100	100
Diaphragms								
Stressed via ΔP	90	90	90	90	90	90	90	90
Springs								
Long Stroke	98	98	98	98	97	97	98	98
Short Stroke	99	99	99	99	97	97	97	97
Bearing								
Hydrodynamic	99	98	98	98	100	100	100	100
Hydrostatic	100	100	100	100	98	98	100	100
Rotational Stresses								
Cycling Stresses	98	98	98	98	95	95	95	95
Battery Cycle and Seal Life	90	90	90	100	100	100	100	100
Total Reliability Against Wear-Out	67	67	66	78	57	65	58	61

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failure and therefore these engines are given a score of 90. The rotary vapor engine has seals between the turbine and the feedwater pump, which upon wearout, will degrade system performance but probably not cause system failure. On this basis the rotary vapor engine was given a score of 98. Similar scores were given to all the systems for the 8 categories and the comparative probability of the systems not experiencing wearout in 10 years' time obtained by series modeling the 8 categories of reliability scores as presented at the bottom of Table 4-16. The reliability against wearout of the hybrid system without battery was better than either the thermoelectric or rotary vapor systems alone because of the lack of the thermal battery in the hybrid system. The static battery seals are presently an unsolved materials compatibility problem and therefore the battery constitutes an additional source for system unreliability against wearout.

The redundancy advantages of the hybrid system are maintained in the wearout reliability modeling as well as in the premature failure reliability modeling. The advantage in wearout reliability occurs primarily because the redundancy consists of two different types of engine which have a small probability of wearout at the same time. For example, it is unlikely that the thermoelectric couples would suffer complete electrical power output degradation at the same time that the turbogenerator or dc electric motor failed. In contrast to the non-redundant candidate system, the recipient could obtain remedial assistance prior to loss of both of the engine power supplies.

4.4 CONFIDENCE IN THE RELIABILITY ESTIMATES

The amount of confidence which can be associated with the system reliability estimates for both wearout and premature failure probabilities has been reflected in a representative 'figure of merit'. The 'figure of merit' is proportional to the degree to which each of the candidate systems fulfills the 4 different criteria as shown at the headings of Table 4-17. Each of the candidate systems was reviewed on the basis of how well the engine, PCCS, blood pump actuator, TESM or battery met the four evaluation criteria. Thus, if a candidate system consisted of an engine, PCCS, and actuator with no TESM or thermal battery, the total of 20 points allocated to criteria #1, 'Experimental Data Availability' was apportioned to

Table 4-17. Confidence in Reliability Estimates

Candidate Systems	Criteria No. 1 Experimental Data Available on Actual Component Hardware in Size and Power Range (20/45)				Criteria No. 2 Data Available From Two or More Sources (5/45)				Criteria No. 3 Data Scaled From Similar Components (10/45)				Complexity Level (10/45)	Total Raw Score	Normalized Score
	Battery	Engine	PCU	Actuator	Battery	Engine	PCU	Actuator	Battery	Engine	PCU	Actuator			
T/E with Battery	0	5	5	2	0	5	5	0	0	5	5	1	3.85	23.85	5.3
Rotary Vapor with Battery	0	3	5	2	0	5	5	0	0	5	5	1	2.93	20.43	4.6
Hybrid with Battery	0	4	5	2	0	5	5	0	0	5	5	1	2.73	21.73	4.8
Hybrid	Engine & T/E		PCU	Actuator	Engine & T/E		PCU	Actuator	Engine & T/E		PCU	Actuator	2.73	29.36	6.5
	4		5	2	5		5	0	5		5	1			
Gas Reciprocating with TESM	1	1	3	1	1	1	3	0	0	1	2	1	3.03	12.27	2.8
Gas Reciprocating	Engine		PCU	Actuator	Engine		PCU	Actuator	Engine		PCU	Actuator	3.03	13.73	3.1
	1		3	1	1		3	0	1		2	1			
Linear Vapor with TESM	1	3	5	0	1	3	5	0	0	2	5	1	3.90	19.15	4.3
Linear Vapor	Engine		PCU	Actuator	Engine		PCU	Actuator	Engine		PCU	Actuator	3.90	25.2	4.3
	3		5	2	3		5	0	2		5	1			

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the three subsystems on a (5, 5, 5) x 20/15 maximum point count. If the candidate systems included a battery as well, the 20 points were apportioned on a 5, 5, 5, 5 maximum point count.

For the first criterion, 'Experimental Data Availability' the candidate engines were ranked in order of decreasing data availability as follows:

<u>Engine</u>	<u>Data Availability</u>	<u>'Figure of Merit' or Point Count</u>
Static Thermoelectric	High	5
Rotary Vapor	Moderately high	3
Linear Vapor	Moderately high	3
Gas Reciprocating	Low	1

And similarly for the PCU Subsystem:

<u>Engine</u>	<u>PCU</u>	<u>Data Availability</u>	<u>'Figure of Merit' or Point Count</u>
Static T/E & Rotary Rankine	Electrical	High	5
Vapor Engine	T/E & Solid State Devices	High	5
Gas Cycle	Fluid Timer & Accumulator	Moderate	3

For the systems incorporating the sodium-sulfur battery, a zero point score was given to the battery since there are no existing reliability data either for the ceramic-to-metal seal life or relative to premature failure and cycle life characteristics.

Similar rankings were performed on the second and third evaluation criteria 'Data Available from Two or More Sources', and 'Data Scaled from Similar Components'. Total (raw) score for the four evaluation criteria and the weighted score are shown for each candidate system in the last two columns of Table 4-17.

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5. CANDIDATE SYSTEM DESCRIPTION

The eight candidate thermal converter systems are summarized in the following figures:

- 5-1 Gas Reciprocating/TESM Converter System
- 5-2 Gas Reciprocating Converter System
- 5-3 Linear Vapor/TESM Converter System
- 5-4 Linear Vapor Converter System
- 5-5 Rotary Vapor/Battery Converter System
- 5-6 Thermoelectric/Battery Converter System
- 5-7 Hybrid/Battery Converter System
- 5-8 Hybrid Converter System

BLOOD PUMP ACTUATOR

GAS RECIPROCATING/TESM ENGINE

PCU

SYSTEM DESCRIPTION

The nonsynchronous, modulated, gas cycle engine is a free piston Stirling engine employing mechanical compression by means of a separate power piston and a hollow-core displacer piston. A clearance-type regenerator, consisting of the mid-cylinder portions of the annular space between the engine cylinder and displacer piston, regenerates the heat for the cycle. Power is produced in this engine by heating helium with the isotope and thermal energy storage material (TESM) at the spherical end of the engine cylinder and rejecting heat at the cold end. Variable heat flow input is provided by solidification of the TESM which is a eutectic of LiF/NaF . The 48.9-watt heat source incorporates a vent and capillary tube assembly for release of the decay-produced helium into the abdominal cavity. The displacer piston moves the gas between hot and cold ends of the cylinder causing pressure variations that produce useful work. The springs at the cold end of the engine cylinder maintain the proper phasing between the displacer and power piston. Gas bearings are used at the cold end to reduce wear between the cylinder, power piston, and displacer piston rod.

Pneumatic-to-hydraulic power is accomplished by means of a bellows pump and check valves operating at high speed. Low pressure fluid flows from the PCU into the bellows pump, and is ejected at approximately 180 psia. This rate is controlled by a pneumatic signal from the PCU into the compression (cold) space of the engine that varies the amount of working fluid causing a variation in the work per stroke and cycle rate within the engine. The engine is mounted within a hermetically sealed containment vessel which is sized to provide surface area so the heat flux to the body tissue does not exceed 0.07 watt/cm^2 .

Within the PCU are the fluidic gear and motor/rotary valve. These mechanisms adapt the variable engine output to a constant 120 bpm high pressure, pulsatile, flow for the actuator. The high pressure and the low pressure accumulator and the dead space controls are in the engine enclosure. The high pressure accumulator allows maximum storage of 4.44 watts PCU output power during artificial diastole. A modulative signal is provided to the engine to allow excess isotope heat to be stored in the TESM.

The actuator delivers power to the blood pump at a variable stroke and constant frequency. An antivacuum valve prevents the actuator from imposing negative pressure on the incoming blood flow and limits the stroke of the power piston.

System Data

Engine

Working Fluid	Helium
Maximum Temperature	1200°F
Minimum Temperature	160°F
Maximum Pressure	500 psia
Minimum Pressure	190 psia
Speed (Variable)	480 to 1000 rpm
Efficiency	8 to 12%
Power Output (Hydraulic)	4.1 to 6.7 watts
Output Pressure	180 psia

PCCS

Energy Storage (Hydraulic)	1.11 watt-sec
Frequency	120 bpm
Efficiency	55 to 63%
Volume	0.552 liter
Weight	0.876 Kg

System Summary

Weight	1.75 Kg (3.86 lb)
Volume	1.233 liters (75.2 in ³)
Specific Gravity	1.42
Heat Source Size	48.9 watts
TESM	53.3 watt-hours

Figure 5-1. Gas Reciprocating/TESM Converter System

BLOOD PUMP ACTUATOR

SYSTEM DESCRIPTION

The nonsynchronous, nonmodulated, gas cycle engine is a free piston Stirling engine employing mechanical compression by means of a separate power piston and hollow-core displacer piston. A clearance-type regenerator consisting of the mid-cylinder portions of the annular space between the engine cylinder and displacer piston regenerates the heat for this cycle. Power is produced in this machine by heating helium with the isotope at the spherical end of the engine cylinder and rejecting heat at the annular cold end heat exchanger. The 54.4-watt heat source incorporates a vent and capillary tube assembly for release of the decay-produced helium into the abdominal cavity. The displacer piston moves the gas between the hot and cold ends, and the heating and cooling cause pressure changes to produce work. The springs at the bottom end of the engine maintain the proper phasing between the displacer and power piston. Gas bearings are used at the upper end of the engine to reduce wear between the cylinder, power piston, and displacer piston rod. The engine is mounted within a hermetically sealed containment vessel which is sized to provide surface area so the heat flux to the tissue does not exceed 0.07 watt/cm².

Pneumatic-to-hydraulic power conversion is accomplished by means of the bellows pump and check valves operating at 750 cpm. Low pressure fluid from the power conditioning unit (PCU) flows into the bellows and is ejected at 180 psia into the annular cold end heat exchanger. The high pressure fluid then flows into the tubular heat exchanger which rejects heat to the surrounding isotonic fluid and tissues. The fluid then flows into the PCU at a power output of 7.0 watts and at a pulsatile rate of 750 cpm.

Within the PCU are the high and low pressure accumulators, fluidic gear motor and hydraulic timer switch, rotary valve, and energy dissipative loop. These mechanisms adapt the engine output to a 120 cpm high pressure, pulsatile flow for the actuator. The high pressure accumulator allows storage of 4.44 watts PCU output power during artificial diastole. The excess energy difference between the constant engine output of 7.0 watts and the variable actuator output power is dissipated in a check-valve-controlled bypass.

The actuator delivers power to the blood pump at a constant stroke and frequency. A make-up reservoir automatically delivers a partial volume, complementary to the varying physiological blood flow, for the constant actuator displacement.

System Data	
Engine	
Working Fluid	Helium
Maximum Temperature	1200°F
Minimum Temperature	160°F
Maximum Pressure	500 psia
Minimum Pressure	190 psia
Speed (Constant)	750 rpm
Efficiency	16%
Insulation Efficiency	90%
Power Output (Hydraulic)	7.0 watts
Outlet Pressure	180 psia
PCCS	
Energy Storage (Hydraulic)	1.11 watt-sec
Frequency	120 bpm
Efficiency (Maximum)	63%
Volume	0.496 liter
Weight	0.799 Kg
System Summary	
Weight	1.57 Kg (3.48 lb)
Volume	1.365 liter (83 in ³)
Specific Gravity	1.15
Heat Source Size	54.4 watts

Figure 5-2. Gas Reciprocating Converter System

BLOOD PUMP ACTUATOR

LINEAR VAPOR/TESM ENGINE

SYSTEM DESCRIPTION

The vapor engine uses water as a working fluid in conjunction with a regenerative nonexpansion cycle. The engine operates with a variable speed at heart rate. The unit delivers a power output equivalent to the average blood pump requirement and utilizes a thermal energy storage material (TESM). The engine consists of a bellows with a mechanical output drive that is directly coupled to the blood pump. An electrical control system is used and its power is supplied by a thermoelectric module. The 48-watt heat source incorporates a vent and capillary tube assembly for release of the decay-produced helium into the abdominal cavity. The system components are mounted within a hermetically sealed containment vessel which is sized to provide sufficient surface area so the heat flux to the tissue does not exceed 0.07 watt/cm².

The unit operates with the vapor from the boiler driving the engine bellows. Exhaust vapor from the engine passes through a regenerator then into a condenser where it is condensed and subcooled. The condenser is designed to reject the heat to the package walls where it is then rejected to the body fluids and tissue. The fluid from the condenser passes through the feed pump where it is pumped through the regenerator into the boiler. The unit is controlled by regulating the vapor flow rate into and out of the engine. Since the engine operates with a constant pressure, the inlet valve is open during the entire engine stroke. At the end of the stroke the inlet valve is closed and the exhaust is opened. The exhaust valve remains open during the entire exhaust stroke. By regulating the inlet and exhaust rate, the engine speed can be varied over the desired range.

The heat source fuel inventory was based on the average blood pump power requirements. To accommodate periods of peak activity, 69 watt-hours of TESH are required. The average daily efficiency of the engine is 8.0%.

The actuator unit provides a mechanical push rod linkage with the engine. Power is delivered on the push stroke and a preloaded spring in the engine provides the force for returning the actuator rod during blood pump filling. An antisuction valve in the actuator prevents negative pressure from being developed in the blood pump.

The engine has an end of fill stroke switch to initiate the power stroke. As a result, the actuator unit operates at variable frequency (60-120 cpm) depending on the rate of blood flow into the pump.

System Data

Operating Conditions

Boiler	95 psia, 900°F
Condenser	3 psia, 142°F
Subcooler	3 psia, 115°F

Efficiency

Ideal Cycle	14.2%
Machine	49 to 60%
Overall	7 to 8.6%

Engine Characteristics

Speed	60 to 120 rpm
Stroke	1.3 inch
Diameter	0.58 inch
Power Output	2.7 to 5.9 watts

Feed Pump

Stroke	0.10 inch
Diameter	0.18 inch
Power Requirement	0.21 watt

Thermoelectric Module

Material	SiGe
Number of Couples	14
Electrical Power Output	0.96 watt
Voltage	0.6 volt

Actuator Unit

Efficiency	76%
Weight	0.4 Kg
Volume	0.267 liter
Frequency	60 to 120 bpm

System Summary

Weight	1.77 Kg (3.91 lb)
Volume	1.42 liters (86.4 in ³)
Specific Gravity	1.25
Heat Source Size	48 watts
TESM	69 watt-hours

Figure 5-3. Linear Vapor/TESM Converter System

BLOOD PUMP ACTUATOR

LINEAR VAPOR ENGINE

SYSTEM DESCRIPTION

The vapor engine uses water as a working fluid in conjunction with a regenerative nonexpansion cycle. The engine operates at constant frequency (120 bpm) and delivers constant power at all times. The power output is equivalent to the maximum blood pump requirements. Since it delivers a constant power output, no thermal energy storage material is required. The unit consists of a bellows with a mechanical output drive that is directly coupled to the blood pump. An electrical control system is used and its power is supplied by a thermoelectric module. The 57-watt heat source incorporates a vent and capillary tube assembly for release of the decay-produced helium into the abdominal cavity. The system components are mounted within a hermetically sealed containment vessel which is sized to provide sufficient surface area so the heat flux to the tissue does not exceed 0.07 watt/cm².

The unit operates with the vapor from the boiler driving the engine bellows. Exhaust vapor from the engine passes through a regenerator into a condenser where it is condensed and subcooled. The condenser is designed to reject the heat to the package walls where it is then rejected to the body fluids and tissue. The fluid from the condenser passes through the feed pump where it is pumped through the regenerator into the boiler. The unit is controlled by regulating the vapor flow rate into and out of the engine. Since the engine operates with a constant pressure, the inlet valve is open during the entire engine stroke. At the end of the stroke the inlet valve is closed and the exhaust is opened. The exhaust valve remains open during the entire exhaust stroke.

The linear vapor engine generates a mechanical back-and-forth motion which is transmitted to the pump actuator through a flexible plastic-coated braided metal cable. A spring within the actuator unit provides a force bias which effectively maintains the cable in tension which is the preferred operating mode during all phases of the pumping cycle.

No sensors are required to maintain control of the pumping action, the output power is automatically modulated by a variation in the pumping duty cycle (between 25 and 50% at a fixed rate of 120 beats/minute). The duty cycle is automatically regulated by the actuator design and the controlled constant-speed characteristic of the electric motor.

The actuator automatically delivers a partial stroke, according to the physiological demand; it provides the positive filling action necessary to maintain system control ("passive autoregulation") with the mandatory antisuction control; and it presents a simple mechanical interface to the actuator drive unit.

System Data

Operating Conditions	
Boiler	95 psia, 900°F
Condenser	3 psia, 142°F
Subcooler	3 psia, 115°F
Efficiency	
Ideal Cycle	14.2%
Machine	72%
Overall	10.2%
Engine Characteristics	
Speed	120 rpm
Stroke	1.3 inch
Diameter	0.58 inch
Power Output	5.9 watts
Feed Pump	
Stroke	0.10 inch
Diameter	0.18 inch
Power Requirement	0.21 watt
Thermoelectric Module	
Material	SiGe
Number of Couples	14
Electrical Power Output	1.14 watts
Voltage	0.7 volt
Actuator Unit	
Efficiency	76%
Weight	0.191 Kg
Volume	0.103 liter
Frequency	120 bpm
System Summary	
Weight	1.333 Kg (2.94 lb)
Volume	1.4 liters (85.2 in ³)
Specific Gravity	0.95
Heat Source Size	57 watts

Figure 5-4. Linear Vapor Converter System

BLOOD PUMP ACTUATOR

MOTOR-RECIPROCATOR

ROTARY VAPOR/BATTERY ENGINE

SYSTEM DESCRIPTION

The rotary vapor/battery system uses CP-34 as a working fluid in conjunction with a regenerative Rankine cycle. The system includes a turbogenerator unit (turbine, pump, dc generator), three heat exchangers (boiler, condenser, regenerator), a solid electrolyte battery (sodium-sulfur), an isotope heat source, insulation (Min-K/xenon) and a solid state electronics package for speed control and battery charge control. In operation, the working fluid is heated in the boiler to peak cycle conditions (430°F and 250 psia) and expanded through the turbine from which work is abstracted. The turbine exhaust vapor, (250°F and 4 psia) is passed through the regenerator giving up a portion of its heat, and thus preheating the high pressure pump discharge liquid. From the regenerator, the cycle vapor enters the condenser where it rejects additional heat to the body fluids and tissue. The working fluid leaves the condenser as a subcooled liquid and enters the centrifugal pump. The pump increases the liquid pressure back to 250 psia and after passing through the liquid side of the regenerator is returned to the boiler and the cycle is repeated.

The overall system produces 9.6 watts electrical output when combined with a 60-watt thermal heat source at an overall efficiency of 16%. This rotary vapor system provides the 2.81 watts average power to the blood pump with a 41-watt thermal heat source, through a 45%-efficient dc motor-driven mechanical actuator. Under these conditions the output from the turbogenerator need be only 6.25 watts electric. By using a 12.5 watt-hr battery, the system will meet the maximum blood pump power of 4.44 watts.

The heat engine components have been arranged to minimize volume and insulation requirements. To reduce overall heat losses the battery, which operates at 570°F, has been designed as a cylindrical receptacle to enclose a high percentage of the heat source surface area. The boiler, which operates at 430°F, has been placed around the outer surface of the battery. The high temperature turbine and the regenerator are located adjacent to the heat source. Finally, the condenser which operates in the 115° to 117°F temperature range, has been mounted on the inner surface of the engine container permitting heat rejection directly to the body fluids and tissue. The heat flux to the outside of the container does not exceed 0.07 watt/cm². The heat source incorporates a vent and capillary tube assembly for release of the decay-produced helium into the abdominal cavity.

The turbogenerator unit is provided with a speed control circuit which will automatically protect the unit from changes in the load.

The motor/reciprocator unit consists of a 9,000 rpm brushless dc motor operating on gas bearings in a hermetically sealed housing. The motor is coupled through a magnetic clutch and a set of reduction gears to a 120 rpm drum-cam reciprocator. The cam follower imparts reciprocating motion to a rod which has an overall stroke of 1.3 inches.

The electrical motor/reciprocator unit generates a mechanical back-and-forth motion which is transmitted to the pump actuator through a flexible plastic-coated, braided metal cable. A spring within the actuator unit provides a force bias which effectively maintains the cable in tension which is the preferred operating mode during all phases of the pumping cycle.

No sensors are required to maintain control of the pumping action, the output power is automatically modulated by a variation in the pumping duty cycle (between 25 and 50% at a fixed rate of 120 beats/minute). The duty cycle is automatically regulated by the actuator design and the controlled constant-speed characteristic of the electric motor.

The actuator automatically delivers a partial stroke, according to the physiological demand; it provides the positive filling action necessary to maintain system control ("passive autoregulation") with the mandatory ant suction control; and it presents a simple mechanical interface to the actuator drive unit.

System Data

Rotary Vapor Turbogenerator

Working Fluid	CP-34
Peak Cycle Temperature	430°F
Peak Cycle Pressure	250 psia
Condensing Temperature	116°F
Condensing Pressure	4 psia

Power Output	6.25 watts at 24 volts, dc
Overall Heat Engine Efficiency	15.7%

Battery

Energy Storage	12.5 watt-hrs
Output Voltage	24 volts
Operating Temperature	570°F

Motor/Reciprocator

Efficiency	59%
Speed Reduction	9000 to 120 rpm
Input Voltage	24 volts, dc

Actuator

Efficiency	76%
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System Summary

Weight	1.42 Kg (3.13 lb)
Volume	1.07 liters (64.7 in ³)
Specific Gravity	1.34
Heat Source Size	41 watts

Figure 5-5. Rotary Vapor/Battery Converter System

BLOOD PUMP ACTUATOR

THERMOELECTRIC/BATTERY ENGINE

SYSTEM DESCRIPTION

The thermoelectric engine makes use of a thermoelectric converter to transform heat generated in the radioisotope heat source into electrical power by completely static means. The 54 watt heat source incorporates a vent and capillary tube assembly for release of the decay-produced helium into abdominal cavity. Two stages of thermoelectric semiconductor materials have been arranged thermally and electrically in series to take advantage of the optimum operating temperature range of each material and thereby produce a high efficiency device. The hot stage of the converter consists of silicon-germanium thermoelectrics and the cold stage is made up of segmented telluride materials. The converter is divided into two modules which are loaded against the flat ends of the right circular cylinder heat source; the heat source and converter are surrounded by Min K fibrous insulation which is backfilled with an inert gas.

The system components are mounted within a hermetically sealed containment vessel which is sized to provide sufficient surface area such that the heat flux to the tissue does not exceed 0.07 watt/cm². The size of the container with this heat flux is sufficient to also house the motor-reciprocator. The waste heat from the thermoelectrics and motor/reciprocator is conducted to the outer walls of the vessel through the mounting plate. A uniform temperature is maintained along the surface of the vessel with a minimum weight penalty by proper tapering of the wall thickness. The sodium-sulfur battery, provided to meet peak power is contained within an annular space around the RTG and surrounded by insulation to maintain it at a mean temperature of 570° F.

The motor/reciprocator unit consists of a 9,000 rpm brushless dc motor operating on gas bearings in a hermetically sealed housing. The motor is coupled through a magnetic clutch and a set of reduction gears to a 120 rpm drum-cam reciprocator. The cam follower imparts reciprocating motion to a rod which has an overall stroke of 1.3 inches.

The electrical motor/reciprocator unit generates a mechanical back-and-forth motion which is transmitted to the pump actuator through a flexible plastic-coated braided metal cable. A spring within the actuator unit provides a force bias which effectively maintains the cable in tension which is the preferred operating mode during all phases of the pumping cycle.

No sensors are required to maintain control of the pumping action; the output power is automatically modulated by a variation in the pumping duty cycle (between 25 and 50% at a fixed rate of 120 beats/minute). The duty cycle is automatically regulated by the actuator design and the controlled constant-speed characteristic of the electric motor.

The actuator automatically delivers a partial stroke, according to the physiological demand; it provides the positive filling action necessary to maintain system control ("Passive autoregulation") with the mandatory antisuction control; and it presents a simple mechanical interface to the actuator drive unit.

System Data

Silicon Germanium Stage (8 couples)	
Couple Length	0.3 in.
N-element Cross-Sectional Area	0.0179 in ²
P-element Cross-Sectional Area	0.0098 in ²
Segmented Stage (26 couples)	
Couple Length	0.6 in.
N-element Cross-Sectional Area	0.0712 in ²
PbTe-GeTe Length	0.495 in.
2N Length	0.105 in.
P-element Cross-Sectional Area	0.0170 in ²
TAGS Length	0.432 in.
BiSbTe Length	0.168 in.
SiGe Operating Temperatures	T _H - 950° C, T _C - 525° C
Lead Telluride Operating Temperatures	T _H - 500° C, T _C - 50° C
Maximum Heat Source Surface Temperature	1079° C
RTG Power Output	6.25 watts (e)
Load Current	3.28 amps
Load Voltage	1.91 volts
Motor/Cam Power Output	2.81 watts (m)

System Summary

Weight	1.5 Kg (3.31 lb)
Volume	1.37 liters (83.75 in ³)
Specific Gravity	1.10
Heat Source Size	54 watts

Figure 5-6. Thermoelectric/Battery Converter System

BLOOD PUMP ACTUATOR

MOTOR-RECIPROCATOR

HYBRID BATTERY ENGINE

SYS EM DESCRIPTION

The hybrid/battery heat engine subsystem consists of a 37-watt heat source which incorporates a vent and capillary tube assembly for release of the decay produced helium into the abdominal cavity. Mounted in each end of this cylindrical capsule are two cascaded thermoelectric converters. These converters are composed of 24 silicon germanium, high temperature couples and 60 2N/TAGS couples. Combined with this thermoelectric converter is a rotary vapor cycle turbogenerator operating at a peak cycle temperature of 221°C. This temperature is compatible with the cold junction temperature of the 2N/TAGS thermoelectric couples. A sodium-sulfur battery providing 12.5 watt-hours of electrochemical energy storage, operates at 300°C and is also compatible with the turbogenerator peak cycle temperature. The engine unit is insulated with fibrous Min-K filled with xenon gas. The waste heat is rejected through condenser tubes attached to the inside of the titanium container. The surface area of the container package is sufficient to permit heat rejection to the local body tissues at an acceptable rate of 0.07 watt/cm². In order to share the load, the output from the thermoelectric generator must be put through a dc-to-dc converter to boost it to 15 volts, reducing the power output to 1.7 watts (e). The rotary vapor cycle output is 4.55 watts. Thus, the total power input to the motor/reciprocator becomes 6.25 watts (e). The overall efficiency (thermal to electrical) of the hybrid/battery heat engine is 17%. These outputs are integrated by an electronic load-sharing circuit which connects the brushless dc motor load alternately to each generator in a constant voltage, pulse modulated manner. Using this technique, each generator sees an effectively constant load. The switching rate used is approximately four times the dc motor speed so that switching transients are effectively eliminated by the inertia of the motor rotor. For a dc motor operating at 9000 RPM, the electronic commutation speed selected was 600 Hz.

The thermoelectric generator is designed to deliver a constant power output of 2.11 watts at 3.64 vdc. By employing a dc-to-dc converter, using a 50 kHz conversion frequency, this is increased to 15 vdc to be compatible with the 15 vdc output of the turbogenerator. Additional circuit logic is provided to maintain the thermoelectric output voltage within desired tolerances. The turbogenerator unit is provided with a speed control unit employing a parasitic load bank. Hence, at reduced loads, the load-sharing electronic circuitry diverts a portion of the turbogenerator output to the parasitic load bank, thus maintaining this unit at constant speed while matching the overall system output power to the varying load.

The motor/reciprocator unit consists of a 9,000 rpm brushless dc motor operating on gas bearings in a hermetically sealed housing. The motor is coupled through a magnetic clutch and a set of reduction gears to a 120 rpm drum-cam reciprocator. The cam follower imparts reciprocating motion to a rod which has an overall stroke of 1.3 inches.

The electrical motor/reciprocator unit generates a mechanical back-and-forth motion which is transmitted to the pump actuator through a flexible plastic-coated braided metal cable. A spring within the actuator unit provides a force bias which effectively maintains the cable in tension which is the preferred operating mode during all phases of the pumping cycle.

No sensors are required to maintain control of the pumping action, the output power is automatically modulated by a variation in the pumping duty cycle (between 25 and 50% at a fixed rate of 120 beats/minute). The duty cycle is automatically regulated by the actuator design and the controlled constant-speed characteristics of the electric motor.

The actuator automatically delivers a partial stroke, according to the physiological demand; it provides the positive filling action necessary to maintain system control ("passive autoregulation") with the mandatory antisuction control; and it presents a simple mechanical interface to the actuator drive unit.

System Data

Thermoelectrics

SiGe	
Hot Junction	950°C
Cold Junction	500°C
2N/TAGS	
Hot Junction	475°C
Cold Junction	240°C
Power Output	2.11 watts 3.64 volts, dc
Power Out of dc to dc Converter	1.7 watts 15 volts

Motor/Reciprocator

Efficiency	59%
Speed Reduction	9,000 to 120 rpm
Dc/dc Conversion	3.64 to 15 volts

Rotary Vapor Turbogenerator

Working Fluid	CP-34
Peak Cycle Temperature	221°C
Condensing Temperature	47°C
Power Output	4.55 watts 15.0 volts, dc
Efficiency	15.25%

Battery

Energy Storage	12.5 watt/hr
Output Voltage	15 volts
Operating Temperature	300°C

Actuator

Efficiency	76%
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System Summary

Weight	1.840 Kgm (4.06 lb)
Volume	1.396 liters (85.2 in ³)
Specific Gravity	1.32
Heat Source Size	37 watts

Figure 5-7. Hybrid/Battery Converter System

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HYBRID CONVERTER SYSTEM SUMMARY

SYSTEM DESCRIPTION

The hybrid heat engine subsystem consists of a 49-watt heat source which incorporates a vent and capillary tube assembly for release of the decay-produced helium into the abdominal cavity. Mounted on each end of this cylindrical capsule are two cascaded thermoelectric converters. These converters are composed of 24 silicon-germanium high-temperature couples and 60 2N/TAGS couples. Combined with this thermoelectric converter is a rotary vapor cycle turbogenerator operating at a peak cycle temperature of 221°C. This temperature is compatible with the cold junction temperature of the 2N/TAGS thermoelectric couples. The rotary vapor cycle system uses CP-34 (thiophene) as a working fluid

The engine unit is insulated with fibrous Min-K filled with xenon gas. The waste heat is rejected through condenser tubes attached to the inside of the titanium container. The surface area of the container package is sufficient to permit heat rejection to the local body tissues at an acceptable rate of 0.07 watt/cm².

In order to share the load, the output from the thermoelectric generator must be put through a dc-to-dc converter to boost it to 15 volts, reducing the power output to 2.61 watts (e). Thus, the total power input to the motor/reciprocator becomes 9.85 watts (e). The overall efficiency (thermal to electrical) of the hybrid heat engine is 20%. These outputs are integrated by an electronic load-sharing circuit which connects the brushless dc motor load alternately to each generator in a constant voltage, pulse-modulated manner. Using this technique, each generator sees an effectively constant load. The switching rate used is approximately four times the dc motor speed so that switching transients are effectively eliminated by the inertia of the motor rotor. For a dc motor operating at 9000 RPM, the electronic commutation speed selected was 600 Hz.

The thermoelectric generator is designed to deliver a constant power output of 3.21 watts at 3.64 vdc. By employing a dc-to-dc converter, using a 50 kHz conversion frequency, this is increased to 15 vdc to be compatible with the 15 vdc output of the turbogenerator. Additional circuit logic is provided to maintain the thermoelectric output voltage within desired tolerances. The turbogenerator unit is provided with a speed control unit employing a parasitic load bank. Hence, at reduced loads, the load-sharing electronic circuitry diverts a portion of the turbogenerator output to the parasitic load bank, thus maintaining this unit at constant speed while matching the overall system output power to the varying load.

The motor/reciprocator unit consists of a 9,000 rpm brushless dc motor operating on gas bearings in a hermetically sealed housing. The motor is coupled through a magnetic clutch and a set of reduction gears to a 120 rpm drum-cam reciprocator. The cam follower imparts reciprocating motion to a rod which has an overall stroke of 1.3 inches.

The electrical motor/reciprocator unit generates a mechanical back-and-forth motion which is transmitted to the pump actuator through a flexible plastic-coated braided metal cable. A spring within the actuator unit provides a force bias which effectively maintains the cable in tension which is the preferred operating mode during all phases of the pumping cycle.

No sensors are required to maintain control of the pumping action, the output power is automatically modulated by a variation in the pumping duty cycle (between 25 and 50% at a fixed rate of 120 beats/minute). The duty cycle is automatically regulated by the actuator design and the controlled constant-speed characteristics of the electric motor.

The actuator automatically delivers a partial stroke, according to the physiological demand; it provides the positive filling action necessary to maintain system control ("passive autoregulation") with the mandatory antisuction control; and it presents a simple mechanical interface to the actuator drive unit.

System Data

Thermoelectrics		
SiGe	Hot Junction	950°C
	Cold Junction	500°C
2N/TAGS	Hot Junction	475°C
	Cold Junction	240°C
	Power Output	3.21 watts 3.64 volts, dc
	Efficiency	6.55%
	Power Out of dc to dc Converter	2.61 watts 15 volts
Motor/Reciprocator		
Efficiency		59%
Speed Reduction		9,000 to 120 rpm
Dc/dc Conversion		3.64 to 15 volts
Rotary Vapor Turbogenerator		
Working Fluid		CP-34
Peak Cycle Temperature		221°C
Condensing Temperature		47°C
Power Output		7.24 watts 15.0 volts, dc
Efficiency		15.75%
Actuator		
Efficiency		76%
System Summary		
Weight		1.83 Kg (4.03 lb)
Volume		1.33 liters (81.4 in ³)
Specific Gravity		1.37
Heat Source Size		49 watts

Figure 5-8. Hybrid Converter System

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6. EVALUATION CRITERIA AND SCORING METHODOLOGY

The candidate converter system design options were required to meet all the design groundrules specified in Table 2-1. In addition, 24 evaluation criteria were developed during Task 1 of the Project. These are explained in Appendix A and summarized in Figure 6-1. Four of the original 24 evaluation criteria have been utilized as additional groundrules for defining acceptable design options rather than for scoring candidate systems. These include: Pump Filling Pressure (A1) wherein all design options must have a mean back pressure during pump filling of 2-10 mm Hg; and Blood Overtemperature (B7), Chronic Tissue Temperature (C3), and Chronic Blood Temperature (C4) which are obviated by designing all the thermal converters to reject all heat to the abdominal cavity at no more than 0.07 watt/cm^2 .

Each of the proposed evaluation criteria is not considered to be equally important in determining the best candidate system design. Therefore, we have allocated to each parameter a numerical weighting factor which represents its value (on an arbitrary scale) relative to the rest of the group of evaluation parameters. These factors are included along with each criterion in Figure 6-1. Table 6-1 presents the relative value coefficients in descending order of importance.

The process for determining which is the "best" candidate is as follows:

- Each candidate system is examined against the evaluation criteria and is allocated a primary score against each of the parameters. If any of the evaluation criteria fail to establish a spread between the candidates of more than 10% in the primary scores, the criteria in question will be eliminated from the scoring process. This precaution will reduce the tendency of the less discriminating parameters to reduce the significant difference between the candidates.
- Each of these primary scores is then multiplied by the relative value coefficient allocated to that particular evaluation parameter to arrive at a weighted score in each category.

Figure 6-1. Evaluation Criteria

Table 6-1. Relative Value Coefficients Summary

10.0	B-1	Reliability against random failure
9.5	C-6	Specific gravity
9.0	C-5	Total volume
8.5	E-3	Design growth potential
8.0	B-7	Blood overtemperature
7.0	C-2	Maximum rate of heat rejection
7.0	D-2	Sensitivity to ambient pressure changes
6.5	E-1	Component technology readiness
6.0	E-4	Estimated unit production cost
6.0	B-3	Confidence levels for the reliability estimates
5.5	C-8	Isotope inventory
5.5	E-2	Estimated development costs
5.0	B-6	Tissue overtemperature
4.5	C-1	Endogenous heat
4.5	C-7	Maximum linear dimension
4.0	B-4	Ability to detect incipient system failure
3.5	B-2	Reliability against wear-out
3.0	B-5	Leakage of toxic materials
3.0	C-4	Chronic blood film temperature
2.5	D-1	Ease of surgical installation
2.0	A-1	Pump filling pressure
1.5	D-4	Sensitivity to electromagnetic fields
1.0	C-3	Chronic tissue temperatures
1.0	D-3	Sensitivity to mechanical shock

- The sum of the weighted scores in each category, for each of the candidate systems, is considered to be a compound figure of merit representing the relative performance of that system according to that particular relative value scheme.
- The systems are then ranked with the highest figure of merit representing the "best" candidate.
- In Section 8.0 the sensitivity of the ranking to both judgmental and procedural aspects of the scoring methodology is discussed

7. CANDIDATE SCORING

Tables 7-1 through 7-8 present scoring data for each of the eight candidate systems. Table 7-9 presents a summary matrix of candidate systems, evaluation criteria, and comparative scores.

Six of the evaluation criteria did not provide greater than a 10 percent variation among the candidate systems and are considered "wash-out" criteria. These included Installation time (D1), Sensitivity to Ambient Pressure Changes (D2), Mechanical Shock (D3), Electro-magnetic Fields (D4) and Production Costs (E-4). Since four criteria became design groundrules and six were found to be "wash-outs", the candidates were scored on the remaining fourteen evaluation criteria.

Table 7-1. Scoring of Thermoelectric/Battery System

Evaluation Criteria		System Raw Score	System Weighted Score	Candidate System Data/Comments
Description	Relative Value Coefficient			
A1 Pump Pressure	2.0	10.0	--	Groundrule: Designed for 6-10mm Hg
B1 Reliability Random	10.0	4.8	48.0	Reliability is 0.58
B2 Reliability Wearout	3.5	4.0	14.0	Reliability is 0.67
B3 Reliability Confidence	6.0	5.3	31.8	Subscores are 12, 2.5, 5.5, 3.85
B4 Failure Detection	4.0	1.0	4.0	System runs several hours off battery
B5 Toxic Material	3.0	10.0	--	Wash criterion (less than 10% difference)
B6 Tissue Temperature	5.0	8.9	44.5	Reliability is 0.89
B7 Blood Temperature	8.0	10.0	--	Groundrule: No heat exchange into blood
C1 Endogenous Heat	4.5	4.5	20.3	54-w heat source
C2 Maximum Heat Rejection	7.0	1.6	11.2	51.16-w constant + 1.73-w battery = 52.89-w
C3 Chronic Tissue, °F	1.0	0	--	Groundrule: Designed for 0.07 w/cm ²
C4 Chronic Blood, °F	3.0	0	--	Groundrule: Designed for 0.07 w/cm ²
C5 Volume	9.0	1.8	16.2	1.372 liters (83.75 in ³)
C6 Specific Gravity	9.5	9.6	91.2	Specific Gravity is 1.10
C7 Dimensions	4.5	6.3	28.4	3.4 in OD cylinder
C8 Isotope Inventory	5.5	0.9	5.0	54-w heat source
D1 Installation Time	2.5	5.4	--	Wash criterion (less than 10%)
D2 Pressure	7.0	10.0	--	Wash criterion (less than 10%)

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Table 7-1 Scoring of Thermoelectric/Battery System (Continued)

Evaluation Criteria		System Raw Score	System Weighted Score	Candidate System Data/Comments
Description	Relative Value Coefficient			
D3 Shock	1.0	10.0	--	Wash criterion (less than 10%)
D4 Electromagnetic Fields	1.5	10.0	--	Wash criterion (less than 10%)
E1 Technological Readiness	6.5	6.1	39.7	Subscores 11, 3.2, 3.2, 3.4, 8, 7.8
E2 Development Cost	5.5	1.7	9.4	6.0 M
E3 Potential	8.5	7.5	51.0	Subscores 7, 5
E4 Production Cost	6.0	10.0	--	2 6K (wash criterion at 10%)
Total			414.7	

Table 7-2. Scoring of Rotary Vapor/Battery System

Evaluation Criteria			System Raw Score	System Weighted Score	Candidate System Data/Comments
Description		Relative Value Coefficient			
A1	Pump Pressure	2.0	10.0	--	Groundrule: Designed for 6-10mm Hg
B1	Reliability Random	10.0	4.3	43.0	Reliability is 0.55
B2	Reliability Wearout	3.5	4.0	14.0	Reliability is 0.67
B3	Reliability Confidence	6.0	4.7	28.2	Subscores are 10, 2.5, 5.5, 2.93
B4	Failure Detection	4.0	1.0	4.0	System runs several hours off battery
B5	Toxic Material	3.0	9.9	--	Wash criterion (less than 10% difference)
B6	Tissue Temperature	5.0	8.4	42.0	Reliability is 0.84
B7	Blood Temperature	8.0	10.0	--	Groundrule: No heat exchanger into blood
C1	Endogenous Heat	4.5	8.4	37.8	41-w heat source
C2	Maximum Heat Rejection	7.0	2.8	19.6	38.2-w constant + 1.8-w battery = 40-w
C3	Chronic Tissue, °F	1.0	0	--	Groundrule: Designed for 0.07 w/cm^2
C4	Chronic Blood, °F	3.0	0	--	Groundrule: Designed for 0.07 w/cm^2
C5	Volume	9.0	7.0	63.0	1.06 liters (64.7 in^3)
C6	Specific Gravity	9.5	5.0	47.5	Specific Gravity = 1.34
C7	Dimensions	4.5	0	0	3.25 in OD cylinder
C8	Isotope Inventory	5.5	3.0	16.5	41-w heat source
D1	Installation Time	2.5	5.4	--	Wash criterion (less than 10%)
D2	Pressure	7.0	10.0	--	Wash criterion (less than 10%)

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Table 7-2. Scoring of Rotary Vapor/Battery System (Continued)

Evaluation Criteria		System Raw Score	System Weighted Score	Candidate System Data/Comments
Description	Relative Value Coefficient			
D3 Shock	1.0	10.0	--	Wash criterion (less than 10%)
D4 Electromagnetic Fields	1.5	10.0	--	Wash criterion (less than 10%)
E1 Technological Readiness	6.5	5.8	37.7	Subscores 10, 2.8, 3.2, 3.2, 7.8, 7.6
E2 Development Cost	5.5	1.4	7.7	7.0 M
E3 Potential	8.5	7.0	51.0	Subscores 6, 6
E4 Production Cost	6.0	10.0	--	3.1 K (wash criterion at 10%)
Total			412.0	

Table 7-3. Scoring of Hybrid/Battery System

Evaluation Criteria			System Raw Score	System Weighted Score	Candidate System Data/Comments
Description		Relative Value Coefficient			
A1	Pump Pressure	2.0	10.0	--	Groundrule: Designed for 6-10mm Hg
B1	Reliability Random	10.0	5.2	52.0	Reliability is 0.61
B2	Reliability Wearout	3.5	3.7	13.0	Reliability is 0.66
B3	Reliability Confidence	6.0	4.8	28.8	Subscores are 11, 2.5, 5.5, 2.73
B4	Failure Detection	4.0	5.0	20.0	Redundant System. Av. 3 day warning
B5	Toxic Material	3.0	9.9	--	Wash criterion (less than 10% difference)
B6	Tissue Temperature	5.0	8.6	43.0	Reliability is 0.86
B7	Blood Temperature	8.0	10.0	--	Groundrule: No heat exchanger into blood
C1	Endogenous Heat	4.5	8.6	38.7	37-w heat source
C2	Maximum Heat Rejection	7.0	8.4	58.8	34.19-w constant + 1.76-w battery = 35.95-w
C3	Chronic Tissue, °F	1.0	0	--	Groundrule: Designed for 0.07 w/cm ²
C4	Chronic Blood, °F	3.0	0	--	Groundrule: Designed for 0.07 w/cm ²
C5	Volume	9.0	1.6	14.4	1.396 liters (85.2 in ³)
C6	Specific Gravity	9.5	5.8	55.1	Specific Gravity - 1.32
C7	Dimensions	4.5	0	0	3.6 in OD cylinder
C8	Isotope Inventory	5.5	3.8	20.9	37-w heat source
D1	Installation Time	2.5	5.4	--	Wash criterion (less than 10%)
D2	Pressure	7.0	10.0	--	Wash criterion (less than 10%)

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Table 7-3. Scoring of Hybrid/Battery System (Continued)

Evaluation Criteria		System Raw Score	System Weighted Score	Candidate System Data/Comments
Description	Relative Value Coefficient			
D3 Shock	1.0	10.0	--	Wash criterion (less than 10%)
D4 Electromagnetic Fields	1.5	10.0	--	Wash criterion (less than 10%)
E1 Technological Readiness	6.5	5.3	34.5	Subscores 9.4, 2.2, 3, 3, 7.2, 7.2
E2 Development Cost	5.5	0.9	5.0	8.5 M
E3 Potential	8.5	7.0	46.8	Subscores 6, 5
E4 Production Cost	6.0	10.0	--	4.1 K (wash criterion at 10%)
Total			431.0	

Table 7-4. Scoring of Hybrid System

Evaluation Criteria			System Raw Score	System Weighted Score	Candidate System Data/Comments
Description		Relative Value Coefficient			
A1	Pump Pressure	2.0	10.0	--	Groundrule: Designed for 6-10mm Hg
B1	Reliability Random	10.0	6.4	64.0	Reliability is 0.68
B2	Reliability Wearout	3.5	6.6	23.1	Reliability is 0.78
B3	Reliability Confidence	6.0	6.5	39.0	Subscores are 16, 3.3, 7.3, 2.73
B4	Failure Detection	4.0	5.0	20.0	Redundant System. Av. 3 day warning
B5	Toxic Material	3.0	9.9	--	Wash criterion (less than 10% difference)
B6	Tissue Temperature	5.0	8.6	43.0	Reliability is 0.86
B7	Blood Temperature	8.0	10.0	--	Groundrule: No heat exchanger into blood
C1	Endogenous Heat	4.5	8.0	36.0	49-w heat source
C2	Maximum Heat Rejection	7.0	2.2	15.4	49-w - 2.22-w = 46.78-w
C3	Chronic Tissue, °F	1.0	0	--	Groundrule: Designed for 0.07 w/cm ²
C4	Chronic Blood, °F	3.0	0	--	Groundrule: Designed for 0.07 w/cm ²
C5	Volume	9.0	2.5	22.5	1.334 liters (81.4 in ³)
C6	Specific Gravity	9.5	4.4	41.8	Specific Gravity = 1.37
C7	Dimensions	4.5	0	0	3.5 in OD cylinder
C8	Isotope Inventory	5.5	1.8	9.9	49-w heat source
D1	Installation Time	2.5	5.4	--	Wash criterion (less than 10%)
D2	Pressure	7.0	10.0	--	Wash criterion (less than 10%)
D3	Shock	1.0	10.0	--	Wash criterion (less than 10%)

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Table 7-4. Scoring of Hybrid System (Continued)

Evaluation Criteria		System Raw Score	System Weighted Score	Candidate System Data/Comments
Description	Relative Value Coefficient			
D4 Electromagnetic Fields	1.5	10.0	--	Wash criterion (less than 10%)
E1 Technological Readiness	6.5	7.4	48.1	Subscores 14, 3.6, 5.2, 4.4, 8.8, 8.6
E2 Development Cost	5.5	1.8	9.9	5.5 M
E3 Potential	8.5	9.5	80.8	Subscores 9, 10
E4 Production Cost	6.0	10.0	--	3.9 K (wash criterion at 10%)
Total			453.5	

Table 7-5. Scoring of Gas Reciprocating/TESM System

Evaluation Criteria			System Raw Score	System Weighted Score	Candidate System Data/ Comments
Description		Relative Value Coefficient			
A1	Pump Pressure	2.0	10.0	--	Groundrule: Designed for 6-10mm Hg
B1	Reliability Random	10.0	3.5	35.0	Reliability is 0.50
B2	Reliability Wearout	3.5	1.6	5.6	Reliability is 0.57
B3	Reliability Confidence	6.0	2.8	16.8	Subscores are 7, .8, 2, 3.03
B4	Failure Detection	4.0	0	0	System stops with engine failure
B5	Toxic Material	3.0	10.0	--	Wash criterion (less than 10% difference)
B6	Tissue Temperature	5.0	6.4	32.0	Reliability is 0.64
B7	Blood Temperature	8.0	10.0	--	Groundrule: No heat exchanger into blood
C1	Endogenous Heat	4.5	8.0	36.0	49-w heat source
C2	Maximum Heat Rejection	7.0	0.3	2.1	76.28-w maximum rejected
C3	Chronic Tissue, °F	1.0	0	--	Groundrule: Designed for 0.07 w/cm ²
C4	Chronic Blood, °F	3.0	0	--	Groundrule: Designed for 0.07 w/cm ²
C5	Volume	9.0	4.2	37.8	1.233 liters
C6	Specific Gravity	9.5	2.8	26.6	Specific Gravity is 1.42
C7	Dimensions	4.5	0	0	3.25 in OD cylinder
C8	Isotope Inventory	5.5	1.8	9.9	49-w heat source
D1	Installation Time	2.5	5.4	--	Wash criterion (less than 10%)
D2	Pressure	7.0	10.0	--	Wash criterion (less than 10%)
D3	Shock	1.0	10.0	--	Wash criterion (less than 10%)

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Table 7-5. Scoring of Gas Reciprocating/ TESM System (Continued)

Evaluation Criteria			System Raw Score	System Weighted Score	Candidate System Data/Comments
Description	Relative Value Coefficient				
D4 Electromagnetic Fields	1.5	10.0	--		Wash criterion (less than 10%)
E1 Technological Readiness	6.5	6.9	44.9		Subscores 12.8, 3.4, 3.8, 4.2, 8.8, 8.6
E2 Development Cost	5.5	1.4	7.7		7.0 M
E3 Potential	8.5	8.5	72.3		Subscores 10, 7
E4 Production Cost	6.0	10.0	--		3.6 K (wash criterion at 10%)
Total				326.7	

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Table 7-6. Scoring of Gas Reciprocating System

Evaluation Criteria			System Raw Score	System Weighted Score	Candidate System Data/Comments
Description	Relative Value Coefficient				
A1 Pump Pressure	2.0	10.0	--		Groundrule: Designed for 6-10mm Hg
B1 Reliability Random	10.0	3.5	35.0		Reliability is 0.50
B2 Reliability Wearout	3.5	3.5	12.3		Reliability is 0.65
B3 Reliability Confidence	6.0	3.1	18.6		Subscores are 6.7, 1.3, 2.7, 3.03
B4 Failure Detection	4.0	0	0		System stops with engine failure
B5 Toxic Material	3.0	10.0	--		Wash criterion (less than 10% difference)
B6 Tissue Temperature	5.0	6.4	32.0		Reliability is 0.64
B7 Blood Temperature	8.0	10.0	--		Groundrule: No heat exchanger into blood
C1 Endogenous Heat	4.5	4.4	19.8		54-w heat source
C2 Maximum Heat Rejection	7.0	1.7	11.9		54.4-w - 2.22-w = 52.18-w
C3 Chronic Tissue, °F	1.0	0	--		Groundrule: Designed for 0.07 w/cm ²
C4 Chronic Blood, °F	3.0	0	--		Groundrule: Designed for 0.07 w/cm ²
C5 Volume	9.0	2.0	18.0		1.36 liters
C6 Specific Gravity	9.5	9.5	90.3		Specific Gravity = 1.15
C7 Dimensions	4.5	7.1	32.0		3.04 in OD cylinder
C8 Isotope Inventory	5.5	0.9	5.0		54-w heat source
D1 Installation Time	2.5	5.4	--		Wash criterion (less than 10%)
D2 Pressure	7.0	10.0	--		Wash criterion (less than 10%)
D3 Shock	1.0	10.0	--		Wash criterion (less than 10%)

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Table 7-6. Scoring of Gas Reciprocating System (Continued)

Evaluation Criteria		System Raw Score	System Weighted Score	Candidate System Data/Comments
Description	Relative Value Coefficient			
D4 Electromagnetic Fields	1.5	10.0	--	Wash criterion (less than 10%)
E1 Technological Readiness	6.5	7.2	46.8	Subscores 12.8, 3.6, 5.2, 4.2, 8.8, 8.6
E2 Development Cost	5.5	1.5	8.3	6.5 M
E3 Potential	8.5	8.5	72.3	Subscores 10, 7
E4 Production Cost	6.0	10.0	--	3.5 K (wash criterion at 10%)
Total			402.3	

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Table 7-7. Scoring of Linear Vapor/TESM System

Evaluation Criteria			System Raw Score	System Weighted Score	Candidate System Data/Comments
Description	Relative Value Coefficient				
A1 Pump Pressure	2.0	10.0	--		Groundrule: Designed for 6-10mm Hg
B1 Reliability Random	10.0	4.6	46.0		Reliability is 0.57
B2 Reliability Wearout	3.5	1.8	6.3		Reliability is 0.58
B3 Reliability Confidence	6.0	4.3	25.8		Subscores are 9, 2.3, 4, 3.9
B4 Failure Detection	4.0	0	0		System stops with engine failure
B5 Toxic Material	3.0	10.0	--		Wash criterion (less than 10% difference)
B6 Tissue Temperature	5.0	6.5	32.5		Reliability is 0.65
B7 Blood Temperature	8.0	10.0	--		Groundrule: No heat exchanged into blood
C1 Endogenous Heat	4.5	8.1	36.5		48-w heat source
C2 Maximum Heat Rejection	7.0	0.9	6.3		65.40 - 4.25 = 61.15-w rejected
C3 Chronic Tissue, °F	1.0	0	--		Groundrule: Designed for 0.07 w/cm ²
C4 Chronic Blood, °F	3.0	0	--		Groundrule: Designed for 0.07 w/cm ²
C5 Volume	9.0	1.2	10.8		1.42 liters
C6 Specific Gravity	9.5	8.0	76.0		Specific Gravity is 1.25
C7 Dimensions	4.5	0	0		4.0 in OD cylinder
C8 Isotope Inventory	5.5	1.9	10.5		48-w heat source
D1 Installation Time	2.5	5.4	--		Wash criterion (less than 10%)
D2 Pressure	7.0	10.0	--		Wash criterion (less than 10%)

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Table 7-7. Scoring of Linear Vapor/TESM System (Continued)

Evaluation Criteria		System Raw Score	System Weighted Score	Candidate System Data/Comments
Description	Relative Value Coefficient			
D3 Shock	1.0	10.0	--	Wash criterion (less than 10%)
D4 Electromagnetic Fields	1.5	10.0	--	Wash criterion (less than 10%)
E1 Technological Readiness	6.5	7.2	46.8	Subscores 13.8, 3.6, 3.8, 4.2, 8.8, 8.8
E2 Development Cost	5.5	1.5	8.3	6.5 M
E3 Potential	8.5	8.5	72.3	Subscores 10, 7
E4 Production Cost	6.0	10.0	--	3.3K (wash criterion at 10%)
Total			378.1	

Table 7-8. Scoring of Linear Vapor System

Evaluation Criteria			System Raw Score	System Weighted Score	Candidate System Data/Comments
Description		Relative Value Coefficient			
A1	Pump Pressure	2.0	10.0	--	Groundrule: Designed for 6-10 mm Hg
B1	Reliability Random	10.0	4.9	49.0	Reliability is 0.59
B2	Reliability Wearout	3.5	2.5	8.8	Reliability is 0.61
B3	Reliability Confidence	6.0	5.6	33.6	Subscores are 13.3, 2.7, 5.3, 3.9
B4	Failure Detection	4.0	0	0	System stops with engine failure
B5	Toxic Material	3.0	10.0	-	Wash criteris (less than 10% difference)
B6	Tissue Temperature	5.0	6.5	32.5	Reliability is 0.65
B7	Blood Temperature	8.0	10.0	--	Groundrule: No heat exchanger into blood
C1	Endogenous Heat	4.5	2.1	9.5	57-w heat source
C2	Maximum Heat Rejection	7.0	1.4	9.8	51 w - 2.22 w = 54.88 w
C3	Chronic Tissue, °F	1.0	0	--	Groundrule: Designed for 0.07 w/cm ²
C4	Chronic Blood, °F	3.0	0	--	Groundrule: Designed for 0.07 w/cm ²
C5	Volume	9.0	1.4	12.6	1.50 liters (85.4 in ³)
C6	Specific Gravity	9.5	10.0	95.0	Specific Gravity is 0.95
C7	Dimensions	4.5	6.1	27.5	3.5 inch diameter cylinder
C8	Isotope Inventory	5.5	0.5	2.8	57-w heat source
D1	Installation Time	2.5	5.4	--	Wash criterion (less than 10%)
D2	Pressure	7.0	10.0	--	Wash criterion (less than 10%)

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Table 7-8. Scoring of Linear Vapor System (Continued)

Evaluation Criteria		System Raw Score	System Weighted Score	Candidate System Data/Comments
Description	Relative Value Coefficient			
D3 Shock	1.0	10.0	--	Wash criterion (less than 10%)
D4 Electromagnetic Fields	1.5	10.0	--	Wash criterion (less than 10%)
E1 Technological Readiness	6.5	7.4	48.1	Subscores 13.8, 3.8, 5.2, 4.2, 8.8, 8.8
E2 Development Cost	5.5	1.7	9.4	6.0 M
E3 Potential	8.5	8.5	72.3	Subscores 10, 7
E4 Production Cost	6.0	10.0	--	3.2 K (wash criterion at 10%)
Total			410.9	

Table 7-9. Evaluation Criteria, Weighted Scoring Summary

Criteria	Candidate Systems							
	Thermoelectric/ Battery	Rotary Vapor/ Battery	Hybrid/ Battery	Hybrid	Gas Reciprocating/ TESM	Gas Reciprocating	Linear Vapor/ TESM	Linear Vapor
B1 Reliability (Random)	48.0	43.0	52.0	64.0	35.0	35.0	46.0	49.0
B2 Reliability (Wearout)	14.0	14.0	13.0	23.1	5.6	12.3	6.3	8.8
B3 Reliability Confidence	31.8	28.2	28.8	39.0	16.8	18.6	25.8	33.6
B4 Failure Detection	4.0	4.0	20.0	20.0	0	0	0	0
B6 Tissue Overtemperature	44.5	42.0	43.0	43.0	32.0	32.0	32.5	32.5
C1 Endogenous Heat	20.3	37.8	38.7	36.0	36.0	19.8	36.5	9.5
C2 Maximum Heat Rejection	11.2	19.6	58.8	15.4	2.1	11.9	6.3	9.8
C5 Volume	16.2	63.0	14.4	22.5	37.8	18.0	10.8	12.6
C6 Specific Gravity	91.2	47.5	55.1	41.8	26.6	90.3	76.0	95.0
C7 Maximum Linear Dimension	28.4	0	0	0	0	32.0	0	27.5
C8 Isotope Inventory	5.0	16.5	20.9	9.9	9.9	5.0	10.5	2.8
E1 Technological Readiness	39.7	37.7	34.5	48.1	44.9	46.8	46.8	48.1
E2 Development Cost	9.4	7.7	5.0	9.9	7.7	8.3	8.3	9.4
E3 Growth Potential	51.0	51.0	46.8	80.8	72.3	72.3	72.3	72.3
TOTAL	414.7	412.0	431.0	453.5	326.7	402.3	378.1	410.9

NOTE: Groundrule criteria and wash criteria excluded

8. SENSITIVITY ANALYSIS

8.1 INTRODUCTION

The objective of a sensitivity analysis is to determine the sensitivity of a selection process to subjective judgments.

As described earlier, the candidate scoring methodology requires application of some 24 evaluation criteria, each of which has a possible score of zero to ten depending on the particular value of the parameter being evaluated. Each evaluation criterion in turn has a relative value coefficient (weighting factor) already fixed at a value between one and ten. The total numerical score for each candidate is simply the sum of the products of each evaluation criteria score (ECS) and its respective relative value coefficient (RVC). In theory, the ECS's are the objective judgments (maximum weight, volume, etc.) while the RVC's are the subjective judgments (relative importance of weight versus volume, etc.).

The traditional sensitivity analysis is carried out to determine the sensitivity of the selection process to the relative importance accorded the RVC's. This is usually done by changing only one RVC at a time while keeping all the others fixed. Often this is done on a normalized basis so that the RVC is not limited to its maximum point value, but can assume any percentage of the sum total of the RVC's. While this is potentially useful, even with only 14 of the original 24 evaluation criteria remaining, an enormous amount of data is generated in simply determining the stability of the ranking to the perturbation of just one RVC. The amount of data generated when two or more RVC's are varied, grows geometrically. Also, we find that the assumption that the ECS's are "correct" and free from subjective uncertainty is not really valid. One reason is that the evaluation criteria scoring is necessarily an iterative process. As the designer gets new information, he should change his groundrules. Because of the requirement for freezing the scoring methodology at the end of the first month of Phase I, it was not possible to do this, and some specific examples of the problems this can present will be given later. A second reason is that the evaluation criteria scoring curves are seldom linear over the range of acceptable values, and therefore, the shape of the curve permits judgmental latitude.

For the above reasons and the closeness of the scores of the top candidates, we concluded that any attempt at a purely mechanical manipulation of the RVC's would not provide the required insight into the soundness of our concept selection process. Rather, we chose to undertake a more focused examination of the scoring methodology and a reevaluation of the ranking in light of selective perturbations.

8.2 DISTRIBUTION OF CRITERIA BY CATEGORY

As stated in Section 6, we originally had five general classes of evaluation criteria as shown in the first two columns of Table 8-1. In columns 3, 4, and 5, we show the number of original criteria, the sum of the RVC's for each class, and the percentage of the RVC total represented by each class. However, as indicated in Section 6, prior to final scoring some of the evaluation criteria were made into design groundrules and others were dropped from the scoring because there was less than a 10 percent spread in their ECS's. In addition, we purposely chose not to normalize the highest candidate scores for each criterion to a fixed maximum (say 10 points) and this reduces the effective value of the RVC's. Because of this and our reduction in the number of evaluation criteria from 24 to 14, there was concern that possibly an imbalance would be created among the various criteria classes. The effect on reducing the number of criteria is shown in columns 6, 7, and 8 of Table 8-1. As can be seen, the only result is a modest increase in the importance of the physical parameters as compared to reliability and practicability considerations. The de facto reduction in the values of the RVC's, by not normalizing to a common fixed maximum, is shown in the last three columns of Table 8-1 to have negligible additional impact.

The implications of this can be seen from Table 8-2, which is simply a review of the final candidate rankings, broken down by criteria class subtotals. The hybrid system had the highest subtotal of any of the candidates in the reliability-related (B) and practicability considerations (E) classes and had next to the lowest subtotal in the physical parameter (C) class. However, it was this "C" criteria class that received heavier-than-intended emphasis as a result of our evaluation process. Therefore, the fact that

Table 8-1. Evaluation Criteria Importance by Class

Class	Principal Characteristic	Original			Final				
		No. of Criteria	Actual		No. of Criteria	Actual		Effective	
			ΣRVC's	% Total		ΣRVC's	% Total	ΣRVC's	% Total
A	Pump Filling Pressure	1	2.0	2	0	—	—	—	—
B	Reliability Considerations	7	39.5	32	5	28.5	32	19.1	31
C	Physical Parameters (weight, volume, isotope inventory, etc.)	8	44.0	35	6	40.0	45	28.4	46
D	Environmental Sensitivity	4	12.0	10	0	—	—	—	—
E	Practicability Considerations	4	26.5	21	3	20.5	23	13.9	23
Total			124.0	100	14	89.0	100	61.4	100

Table 8-2. Evaluation Criteria Scoring by Class

System	B	C	E	Total	Rank
	Reliability	Physical Characteristics	Practicability		
Thermoelectric/Battery	142.3	172.3	100.1	414.7	3
Rotary Vapor/Battery	131.2	184.4	96.4	412.0	4
Hybrid/Battery	156.8	187.9	86.3	431.0	2
Hybrid	189.1	125.6	138.8	453.5	1
Gas Reciprocating/TESM	89.4	112.4	124.9	326.7	8
Gas Reciprocating	97.9	177.0	127.4	402.3	6
Linear Vapor/TESM	110.6	140.1	127.4	378.1	7
Linear Vapor	123.9	157.2	129.8	410.9	5

the hybrid engine still emerged with the highest point score indicates that the selection process was insensitive to this perturbation. It should also be noted that the hybrid/battery system, which provides growth potential for the hybrid system, was ranked highest in the "C" criteria class.

8.3 EFFECTS OF CRITERIA SCORING REVISIONS

Since our evaluation criteria scoring methodology was frozen early enough in Phase I to prevent any substantive iteration, it was not surprising to find that several criteria scoring curves or procedures were obsolete.

The most obvious example of this was evaluation criteria C2, maximum rate of heat rejection. Our original scoring curve is shown in Figure 8-1. As described in the Task 1 Final Report, our reasoning was that up to about 40 watts of heat could be rejected through the heart pump directly into the blood stream. Above this value, an additional blood heat exchanger would be required. Therefore, the point value dropped precipitously from 8 to 3 points as the 40-watt value was exceeded. As documented in detail in Section 2.6, we believe that not only is waste heat rejection to the blood pump optional, but it may be difficult to achieve in practice. On the other hand, we found that the thermal converter volume and specific gravity considerations tended to result in package surface areas which were adequate to dissipate up to 60 watts of heat directly to the abdominal fluids and tissue without exceeding heat flux values that appeared safe based on available experimental data.

Therefore, the dashed curve in Figure 8-1 would be more consistent with our current approach to rejecting waste heat. The effect of using this updated scoring curve is shown in Table 8-3. The hybrid system remains the top-ranked candidate by an increased margin while the rotary vapor/battery system replaces the hybrid/battery system as the second-ranked system. Since the C-2 (maximum rate of heat rejection) criterion scoring was the only obvious candidate for modification, we next turned to an examination of the candidate rankings for internal consistency.

An obvious question is raised by the very small percentage point spread (in the original scoring, only 5%) between the hybrid (ranked No. 1) and hybrid/battery (ranked No. 2) systems. This is suspicious since although the hybrid/battery system has the smallest heat source of all the candidates (37 watts), it requires a solid electrolyte battery which is still under development. Therefore, the hybrid battery system should

Figure 8-1. Maximum Rate of Heat Rejection

Table 8-3. Revised Totals and Ranking Using Updated C2 Scoring Criterion

System	Original Scoring	Original Ranking	Revised C2 Scoring	Revised Ranking
Thermoelectric/Battery	414.7	3	427.3	4
Rotary Vapor/Battery	412.0	4	448.4	2
Hybrid/Battery	431.0	2	431.0	3
Hybrid	453.5	1	477.3	1
Gas Reciprocating/TESM	326.7	8	326.7	8
Gas Reciprocating	402.3	6	414.9	6
Linear Vapor/TESM	378.1	7	378.1	7
Linear Vapor	410.9	5	419.3	5

rank low in reliability confidence (B3) and technology readiness (E1). However, as can be seen in Table 7-9, the B1 and E1 scores for all three battery systems are quite presentable.

Inspection of the respective evaluation criteria score sheets showed that solid electrolyte batteries were indeed given a zero on both counts. However, for both these criteria, scoring was done separately on a component basis (engine, insulation, energy storage, power conditioning and control) and the results were added. This is obviously an incorrect procedure since both criteria represent confidence factors which should be determined by the product rather than the sum of the separate component confidence factors. As with reliability, the overall system value can never be higher than that of its weakest component. If the B3 and E1 scores are therefore determined by the products rather than the sums of their component scores, the results are shown in Table 8-4. As can be seen, the hybrid system is now almost 13% ahead of the second-place candidate and the total point spread among the candidates is 33%. The scores and rankings are also shown in Table 8-4 for the case in which all three criteria revisions are applied. The hybrid system's lead is now over 15% and the point spread among the candidates exceeds 36%.

Table 8-4. Revised Totals and Ranking for Updated B3 and E1 Scoring

System	Revised B3 and E1 Scoring	Revised Rank	Revised B3, E1 and C2 Scoring	Revised Rank
Thermoelectric/Battery	358.7	5	371.3	6
Rotary Vapor/Battery	359.5	6	395.9	3
Hybrid/Battery	378.3	3	378.3	5
Hybrid	449.0	1	472.8	1
Gas Reciprocating/TESM	301.0	8	301.0	8
Gas Reciprocating	374.8	4	387.4	4
Linear Vapor/TESM	346.4	7	346.4	7
Linear Vapor	392.3	2	400.8	2

8.4 ALTERNATE SCORING TECHNIQUES

The next question we addressed was whether alternate methods of utilizing the evaluation criteria scores (ECS's) and relative value coefficients (RVC's) might change the relative rankings of the candidates.

As might be anticipated by inspection of the equation $\Sigma ECS \times RVC$: it does not matter whether or not the RVC's are first normalized to unity (or any other number). The relative numerical scores of the candidates, which is what uniquely determines the ranking, will be unchanged. However, there are three different techniques for generating the ECS's which can influence the relative ranking. These include taking the raw scores (zero to a maximum of 10) from the evaluation criteria curves and

- (1) Using them numerically unaltered
- (2) Assigning a value of 10 to the best candidate and normalizing the scores of the remaining candidates
- (3) Adding the scores and normalizing to unity

In the original scoring calculations, method #1 was chosen since both the second and third approaches tend to produce ECS's that reflect relative standings rather than how well each candidate has done in meeting the many evaluation criteria. However, since method #3 is frequently used in decision modeling, we recalculated the scores for each of the candidate systems by adding the scores for each of the criteria and normalizing unity (method #3) rather than using them numerically unaltered (method #1).

We also repeated the selected criteria scoring revisions (C2, B3, and E1) described in Section 8.3. The results are shown in Tables 8-5 and 8-6. The principal difference in the original scoring using method #3 rather than #1 is that the hybrid battery system emerges in first place and the hybrid without a battery comes in second. However, application of any or all of the updated scoring criteria restores the hybrid system to first place and produces roughly the same final standing as presented in Tables 8-3 and 8-4. The percentage point spread, however, is somewhat increased using method #3 with the updated C2, B3, and E1 criteria. The percentage spread is 20% between first and second place and 44% between first and last place; whereas using method #1, the corresponding percentage differences were 15% and 36%.

Table 8-5. Revised Totals and Ranking Using Updated C2 Scoring Criterion and Normalized ECS's (Method #3)

System	Original Scoring	Original Ranking	Revised C2 Scoring	Revised Ranking
Thermoelectric/Battery	11.25	4	11.41	4
Rotary Vapor/Battery	12.28	3	12.95	2
Hybrid/Battery	13.74	1	12.46	3
Hybrid	12.92	2	13.31	1
Gas Reciprocating/TESM	8.43	8	8.38	8
Gas Reciprocating	10.54	5	10.70	5
Linear Vapor/TESM	9.00	7	8.87	7
Linear Vapor	10.46	6	10.54	6

Table 8-6. Revised Totals and Ranking for Updated B3 and E1 Scoring and Normalized ECS's (Method #3)

System	Revised B3 and E1 Scoring	Revised Rank	Revised B3, E1 and C2 Scoring	Revised Rank
Thermoelectric/Battery	10.25	6	10.41	6
Rotary Vapor/Battery	11.35	4	12.02	2
Hybrid/Battery	12.75	2	11.47	3
Hybrid	14.70	1	15.08	1
Gas Reciprocating/TESM	8.52	8	8.47	8
Gas Reciprocating	10.64	5	10.80	5
Linear Vapor/TESM	9.02	7	8.88	7
Linear Vapor	11.38	3	11.46	4

8.5 CONCLUSIONS

Two factors governed the type of sensitivity analysis that we carried out. The first was that our scoring criteria and methodology had not been allowed to change as our understanding of the problems and solutions matured during Phase I. Therefore, several of the criteria we applied were no longer appropriate in either content or procedure. The second consideration was the closeness of the candidate scores; the top five candidates were all within 10% of each other. Therefore, we could have carried out a traditional sensitivity analysis and showed myriad ways of having any one of these five end up as the top candidate.

Instead we first noted that even with the original scoring methodology, the hybrid system scored highest in both the reliability and practicability categories and still retained its total point supremacy even though it was next to last in the heavily-weighted physical characteristics category. We also noted that the hybrid/battery system, which was one of the growth potentials for the hybrid system was also the highest ranking system on the basis of physical characteristics.

We then presented the logic for revising three of the criteria scoring techniques because they were either no longer valid, or they produced internal inconsistencies. We also considered an alternate calculational method for combining the ECS's and RCV's. Whether these revised scoring procedures were applied separately or in concert, they reaffirmed the top ranking of the hybrid system by statistically significant margins.

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APPENDIX A

A total of 24 evaluation criteria were derived during Task I of the Project. Scoring of the physiologically-related criteria determined with the assistance of our medical advisory group which consisted of the following consultants:

Dr. H. J. C. Swan	Director, Department of Cardiology, Cedars-Sinai Medical Center, Cedars of Lebanon Hospital Division Los Angeles, California
Dr. Yukihiro Nose	Director, Artificial Organs Program, and Head, Department of Artificial Organs Research The Cleveland Clinic Foundation Cleveland, Ohio
Dr. J. Van De Water	Director, Cardiovascular and Thoracic Surgery City of Hope National Medical Center Duarte, California
H. P. Roth	Private consultant, Environmental and Thermal Physiology Manhattan Beach, California

The scoring curves referenced throughout the text that follows are summarized in Figure A-1 which is the same as Figure 6-1.

A-1 PUMP FILLING PRESSURE

It is important that the pressure reflected back into the left atrium by the blood pump during the filling phase be maintained between 0 and 18 mm Hg. At pressures greater than 18 mm Hg, it is believed that the recipient will begin to experience pulmonary congestion; at pressures below ambient, the distribution of blood flow within the lungs is significantly disturbed.

Since pressures between 2 and 10 mm Hg were considered optimum, full score was accorded this range with scores linearly decreasing to zero at 0 and 18 mm Hg. This became a "wash" criterion since the

DEVICE
RELIABILITY

Figure A-2. Interim and Long-Term Reliability

pump/actuator units for all candidate concepts were designed to provide a backpressure of 2-10 mm Hg.

B-1 RELIABILITY AGAINST RANDOM FAILURE AND
B-2 RELIABILITY AGAINST WEAROUT

The reliability estimates for each of the candidate system designs were based on accepted reliability computation procedures as described in Section 4.0.

In discussing what might be a reasonable ten-year reliability goal with our medical consulting group, it became apparent that they would much prefer an intermediate or short-term equivalent measure of the reliability performance of the device. This information, rather than the equivalent ten-year specification, would be generally more useful to them as the basis for their recommendation of the treatment to the patient. This particular factor is illustrated in Figure A-2. For example, they felt that a device implantation procedure with a statistical life expectancy (50% reliability) of five years would be a very acceptable alternative to the patient who has a natural life expectancy of 6 months or less, provided that account is taken of the 10% risk associated with the surgery. Converted into equivalent ten-year reliability terms this requires a device reliability of only 27.8% $\left[\frac{0.50}{2} \times \frac{1}{(1-0.10)} \right]$.

If we assume that the five-year reliability is almost totally dominated by the random failure modes, the significantly increased incidence of wear-out failures at ten years will reduce the overall system reliability even further below the 27.8% level.

To determine what might be reasonable reliability design goals, we adopted the following approach. Based on available "mortality from heart disease" statistics, we took the ten-year survival probability for a middle-aged cardiac patient to be about 50%. This is conservative since virtually all candidates for this type of treatment will have a much poorer prognosis. Thus, increasing his ten-year survival probability from 50% to 75% would appear to be a very reasonable goal. Factoring in the surgical risk, this calls for an overall reliability performance from the device of 83.3% $\left[\frac{0.75}{(1-0.1)} \right]$. This overall performance could be achieved from various combinations of the random failure performance and the wear-out performance.

We can derive a specific division between wear-out and random failure performance by setting a goal for overall five-year reliability. If the total ten-year performance were attributed completely to random failure modes, the above 83.3% would correspond to a five-year performance at the 91.2% level. If we again assume that virtually all of the failures up to the five-year point will be due to random failures, rather than wear-out, then the five-year performance must be taken to be greater than the 91.2% level.

We elected to make the five-year goal 95%. Over a ten-year period, this level of reliability against random failure is equivalent to 89.5%. The difference between this and the net ten-year device reliability of 83.3% defines the ten-year reliability against wear-out at 93.1% $\left[\frac{0.833}{0.895} \right]$.

The minimum acceptable ten-year reliability against random failure level was set, as we have discussed by the medical consulting group at 27.8%. By definition, the minimum acceptable ten-year reliability against wear-out is 50%; a lower level would imply a design lifetime of less than ten years. We therefore adopted the levels summarized below as design goals and acceptable minima for the overall system reliability.

Ten year device reliability:

- Against random failures - goal 90%, minimum 27.8%
- Against wear-out - goal 93%, minimum 50%

Device reliability at the (intermediate) five-year point:

- Against random failures - goal 95%, minimum 55.5%
- Failures due to wear-out - negligible

Scoring between "minimum" and "goal" reliability values was assumed to be linear.

B-3 CONFIDENCE LEVELS FOR THE RELIABILITY ESTIMATES

A representative 'figure of merit' was generated to reflect the level of confidence which could be associated with each of the system reliability estimates. A number of semi-quantitative criteria are listed below. The scores allocated in response to the criteria were summed and normalized to scale of ten.

1. Are there experimental data available from tests on actual component hardware in this power range and in this physical size?

(20/45)

2. Are there data available from 2 or more sources? (5/45)

3. Are the data scaled from experimental investigations of 'similar' components?

(10/45)

4. What is the level of complexity of the candidate system? (10/45)

Scoring the response to item 4 was determined by adding the scores in each of the following categories and normalizing to a scope of 10. Scoring curves are shown in Figure A-3. A response of zero in any category was scored at 10.

- 1) Total number of moving parts (curve #2)
- 2) Total number of cyclically-stressed parts (curve #2)
- 3) Number of metallic components exposed to corrosion stress (curve #2)
- 4) Total parts count (curve #1)
- 5) Number of parts under static stress (curve #1)
- 6) Number of joints (risk of fluid leakage) (curve #1)

B-4 ABILITY TO DETECT INCIPIENT SYSTEM FAILURE

Although the title is misleading, the purpose of this criterion was to take into account the time interval available for remedial action after a typical system malfunction.

Since the recipient can reasonably be expected to undergo physiological function testing at 2 - 3 month intervals, systems with malfunction incidence-to-criticality intervals in this range were awarded the higher scores. Shorter intervals were given an intermediate score because they allow some chance of remedial action if the fault can be diagnosed by a special service or by the recipient himself. It was assumed that at least three days would

Figure A-3. Scoring Curve for Complexity Rating Portion of Criterion B-3.

be a highly desirable interval for the latter case; and therefore, a point score of five was awarded this value. The score falls linearly to zero below the 3-day point and rises less rapidly to 10 above this point.

A failure modes and effects analysis was performed for each candidate system design. For reciprocating systems without a battery, it was assumed that post-engine-failure survival time would be negligible. For those systems with energy storage downstream of the engine (batteries), it was estimated that several hours might be available for remedial action. The hybrid systems were a special case since the recipient could function essentially indefinitely on the reduced power provided by the rotary vapor stage alone (assuming failure of the thermoelectric stage). However, the rise in heat source temperature would probably limit the power from the thermoelectric stage alone (assuming failure of the rotary vapor engine) to several hours at most. Considering the relative failure probabilities of the two power-producing components of the hybrid system, the mean survival time would probably be several days.

B-5 LEAKAGE OF TOXIC MATERIALS

The probability of each malfunction that could lead to an unacceptable concentration of toxic material in the tissue was determined from a failure modes and effects analysis.

The sum of all the probabilities for each candidate system were accumulated and the score calculated as follows

$$\text{Score}_{\text{system } j} = 10(1 - \sum_{\text{system } j} \text{critical failure mode probabilities})$$

Materials in the system inventory (and possible degradation products) that could conceivably be released to the surrounding body tissues were identified. The most obvious toxic materials, TESM, and the sodium and sulfur in the solid electrolyte battery, are solid at body temperature and therefore it is unlikely that these materials would have sufficient mobility to breach the outer thermal converter container. The only identified toxic material that would be liquid at room temperature is the thiophene (CP-34) working fluid employed in the rotary vapor systems. The probability of leakage through the length and type of weldments is estimated at less than 1 %

B-6 TISSUE OVERTEMPERATURE

The most probable degradation, malfunction and wearout modes that could lead to thermal converter surface temperatures higher than the design limit of 43°C, are identified in Table 4-18. All of these modes lead to the result that energy from the heat source is not efficiently converted to blood pump work, but rather becomes additional thermal energy to be dissipated at the thermal converter surface. The overtemperature modes do not necessarily have the same probability of occurrence in all candidate designs, and in some design options they are not even applicable. Therefore, the probabilities of these modes occurring in each of the candidate designs was rated on a basis of 0 to 10% failure probability with the highest probability of tissue overtemperature assigned 10%. These probability rates are closely related to the wearout failure probabilities since almost all wearout modes are concurrent with decreased performance and therefore increased waste heat load. In addition, the systems with TESM cause larger temperature fluctuations even during the course of normal operation

than those systems incorporating electrochemical energy storage. Therefore, the TESM systems generally have higher tissue overtemperature probabilities. The hybrid system has a lower probability of tissue overtemperature than either the thermoelectric or the rotary vapor engines because as the thermoelectric module power drops due to material degradation, the lost power is not rejected directly to the body but becomes an increased thermal input to the turbogenerator. Finally, those systems that have relative motion with metal-to-metal sliding contact on dynamic seals (such as the solenoid valves in the linear vapor engine) are more prone to wearout and a steadily decreasing efficiency, leading to a higher probability of tissue overtemperature.

The scores were determined from the following equation:

$$\text{Score}_{\text{system } j} = 10 (1 - \sum_{\text{system } j} \text{critical failure mode probabilities})$$

The results are shown in the bottom row of Table A-1.

B-7 BLOOD OVERTEMPERATURE

The blood heat exchangers were to be designed to accommodate the maximum rate of heat rejection under conditions within the design specifications. This rate would correspond to a specified maximum blood film temperature (probably 41°C). However, all our candidate concepts were designed to reject all their waste heat directly to the abdominal fluids and tissue at acceptable heat flux levels. Since there would be no blood heat exchanger, this became "wash" criteria as discussed in detail in Section 2.4. However, a separate blood heat exchanger can readily be incorporated if required.

C-1 ENDOGENOUS HEAT

A special weighted value of the overall system efficiency was used to establish the required heat source size, which, by definition becomes the time-averaged endogenous heat burden to the body. The 60-watt upper limit is a specified design groundrule. The full score for values up to about 10 watts reflects the "credit" that is probably available due to the reduced work (heat) output of the natural heart.

Table A-1. Tissue Overtemperature Failure Mode Analysis
(Failure Mode Probability/Reliability)

Possible Failure Mode Leading to Protein Damage	T/E with Battery	Rotary Vapor with Battery	Hybrid with Battery	Hybrid	Gas Reciprocating with TESM	Gas Reciprocating	Linear Vapor with TESM	Linear Vapor
	Failure Proba- bility Reliability	Failure Proba- bility Reliability	Failure Proba- bility Reliability	Failure Proba- bility Reliability	Failure Proba- bility Reliability	Failure Proba- bility Reliability	Failure Proba- bility Reliability	Failure Proba- bility Reliability
Feed Water Pump Efficiency Loss	0.00 1.00	0.04 0.96	0.04 0.96	0.04 0.96	0.00 1.00	0.00 1.00	0.09 0.91	0.09 0.91
PCU Efficiency Loss Caused by Wear	0.05 0.95	0.05 0.95	0.05 0.95	0.05 0.95	0.05 0.95	0.05 0.95	0.01 0.99	0.01 0.99
Fatigue of Springs	0.02 0.98	0.02 0.98	0.01 0.99	0.01 0.99	0.02 0.98	0.02 0.98	0.06 0.94	0.07 0.93
Fatigue of Bellows	0.02 0.98	0.02 0.98	0.02 0.98	0.02 0.98	0.02 0.98	0.02 0.98	0.07 0.93	0.07 0.93
Fouling of Heat Exchange Surfaces	0.01 0.99	0.02 0.98	0.02 0.98	0.02 0.98	0.05 0.95	0.05 0.95	0.05 0.95	0.05 0.95
Engine Efficiency Losses Caused by Wear/Degradation	0.015 0.985	0.02 0.98	0.01 0.99	0.01 0.99	0.02 0.98	0.02 0.98	0.05 0.95	0.05 0.95
Efficiency Losses Caused by Valve Leakage	0.00 1.00	0.00 1.00	0.00 1.00	0.00 1.00	0.06 0.94	0.06 0.94	0.08 0.92	0.08 0.92
(1 - Summation of Critical Failure Mode Probabilities) = Tissue Overtemperature Reliability	0.89	0.84	0.86	0.86	0.64	0.64	0.65	0.65

A-9

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C-2 MAXIMUM RATE OF HEAT REJECTION

The maximum rate of heat rejection occurs at different times for the various systems. For the battery group, it occurs at peak blood pump power and is equal to

$$Q_{\max}(\text{Battery}) = Q_{hs} - P_{ave} + \frac{(P_{\max} - P_{ave})(1 - \eta_{mra})}{\eta_{mra}}$$

$Q_{\max}$ = Maximum rate of heat rejection (watts)

Q_{hs} = Heat source inventory (watts)

$P_{\max}$ = Maximum power into blood pump (4.25 watts)

P_{ave} = Average power into blood pump (2.81 watts)

η_{mra} = Combined efficiency of motor/reciprocator and automatic actuator

The last two terms on the right are about -1.05 watts; and therefore, the maximum rate of heat rejection is a little over one watt less than the thermal inventory.

For systems using thermal energy storage upstream of the engine, the $Q_{\max}$ also occurs at $P_{\max}$ and is given by

$$Q_{\max}(\text{TESM}) = Q_{hs} - P_{ave} + \frac{(P_{\max} - P_{ave})(1 - \eta_{tot})}{\eta_{tot}}$$

where

η_{tot} = Overall heat-to-blood-pump conversion efficiency at a power level of 4.25 watts into the blood pump.

$Q_{\max}$ will generally be higher than Q_{hs} for these systems.

For the systems with no energy storage, $Q_{\max}$ occurs at the time of minimum power delivered to the blood pump ($P_{\min}$) and is given by

$$Q_{\max}(\text{no energy storage}) = Q_{hs} - P_{\min}$$

For these systems $Q_{\max}$ is approximately 22 watts less than the Q_{hs}

The shape of the C-2 scoring curve reflects the following considerations:

- 1) The body can probably accommodate a temporarily raised heat burden for periods comparable with the anticipated periods of sustained greater-than-average activity.
- 2) 5 to 10 watts of waste heat is required to restore the body to its original thermal status prior to the reduction in the heart's work load resulting from introduction of the assist pump.

C-3 CHRONIC TISSUE TEMPERATURES

The scoring of this criterion was to be based on the mean and peak temperatures to which the tissues are exposed during the normal, within-specification operation of the candidate systems

The scores were to be computed from the C-3 scoring curves as follows

$$\text{Score}_{\text{System } j} = \frac{\text{Score}_{\text{peak temp}} \times F_{\text{peak}} + \text{Score}_{\text{mean temp}} \times (1 - F_{\text{peak}})}{1}$$

where F_{peak} is an estimate of the fraction of the time that the exposed surface is at the peak temperature. However, as indicated in Section 2.4, all candidate systems were designed to limit the heat flux to the tissues to 0.07 w/cm^2 and the temperatures to 41°C . Therefore, this became a "wash" criterion in the final scoring.

C-4 CHRONIC BLOOD FILM TEMPERATURES

The mean and peak temperatures of any surface to which the blood is exposed during normal, within-specification operation of the candidate system were to be calculated in a manner exactly analogous to that specified in C-3. Since all the candidate concepts were designed to reject all the heat directly to the abdominal tissues and fluids rather than a blood heat exchanger, the criterion also became a "wash". However, a separate blood heat exchanger could readily be incorporated into any of the candidate concepts if desired.

C-5 TOTAL VOLUME

The total volume of each of the complete candidate thermal converter systems (including the blood pump actuator but not the blood pump itself) was calculated from the component engineering analyses. The C-5 scoring curve was derived in consultation with the medical advisory group and reflects the consensus that a volume up to about 1.0 liters can be accommodated without much difficulty within the lower abdomen. The curve falls to the zero at 1.5 liters to reflect the design groundrule that this value be the maximum allowable volume.

C-6 SPECIFIC GRAVITY

The overall specific gravity of the implanted system was computed from the component engineering analyses, and the C-6 scoring curve used to derive a numerical score for each candidate.

A design goal strongly recommended by our medical advisory group is to achieve an overall package density close to that of body tissue. Neutral buoyancy will reduce the stress on the surrounding body organs and eliminate the need for load-bearing structural ties. The package would ideally be only loosely tethered to the skeletal frame. Surgically, the concept of firm attachment to a bone support is poorly favored. It was recommended that systems that cannot be reduced to an overall specific gravity of less than 2.0 not be considered.

C-7 MAXIMUM VENTRAL DORSAL DIMENSION

The maximum ventral-dorsal dimension of each of the candidate systems was derived from the component engineering analyses. The shape of C-7 scoring curve reflects the surgical feasibility and convenience of locating packages within various parts of the abdominal cavity. The overall maximum of 5 inches represents a typical spacing between the abdominal aorta and the anterior wall of the cavity where a package of close-to-neutral buoyancy might be placed. The 3-inch dimension reflects the narrower space available for more dense packages which might have to be anchored to the pelvic rim.

C-8 ISOTOPE INVENTORY

Systems requiring a smaller amount of radioisotope are obviously preferred. Many of the consequences of the size of the isotope inventory,

such as endogenous heat levels and cost per unit are specifically accounted for elsewhere in the evaluation scheme. However, the actual inventory level itself forms a useful overall figure of merit that will generally take account of several factors such as a smaller radiation dose to the recipient and a smaller exposure per unit to the general population, that are difficult to quantitize precisely. The scoring was assumed to vary linearly from ten to zero over the 60-watt range.

D-1 EASE OF SURGICAL INSTALLATION

The shape of the curve represents the general concensus that there is really no technical basis for limiting the time that the recipient can remain in surgery since a cardiopulmonary bypass is not required. However, there is an increasing risk of death that can be correlated with length of time in surgery, and this accounts largely for the declining slope of the D-1 curve. There are other factors to consider such as labor and surgical facility costs.

The candidate design concepts were presented to members of our medical consulting group who estimated the time required to install each system. There were insufficient differences among the candidate concepts to suggest any significant variations in installation time and therefore this became a "wash" criterion.

D-2 SENSITIVITY TO AMBIENT PRESSURE CHANGES

The objective of this criterion was to determine the change in output power corresponding to a reduction in the ambient pressure to 523 mm Hg (atmospheric pressure at 10,000 feet or 31% reduction in the mean sea level barometric pressure). A 40% loss of power was to have been considered grounds for discarding the candidate. The selected actuator concepts may show a slight decrease in power with decreasing pressure; but since all candidate concepts would have roughly the same response, this became a "wash" criterion.

D-3 SENSITIVITY TO MECHANICAL SHOCK

All the candidate concepts were designed to be unshockable due to mechanical shock. Therefore, since the duration of any power loss would be short, even complete (100%) loss can be tolerated. Considerable shock levels are encountered within the body in most common lifestyles. However,

the risks of interference from accelerations encountered in elevators, and vibration levels in motor vehicles are not considered to be significant hazards. Since all the candidate concepts appeared to be comparably insensitive to mechanical shock, this became a "wash" criterion.

D-4 SENSITIVITY TO ELECTROMAGNETIC FIELDS

All the candidate concepts appeared to be insensitive to power loss from electromagnetic radiation (or a corresponding inductive field) encountered close to fairly common domestic items such as power lawnmowers and microwave ovens. Therefore, this also became a "wash" criterion.

E-1 COMPONENT TECHNOLOGY READINESS

The major components of the candidate systems were evaluated against the series of qualitative criteria listed below. Relative scores were compiled in the matrix format shown for criterion E-1. The sum of the box scores were normalized to a scale of ten.

E-2 ESTIMATED DEVELOPMENT COST

Development costs were estimated under the following assumptions:

- Phase II of the Practicability Evaluation has been successfully completed, including the 5-month test of the Bench Model Converter. (This model, with only minor modification, is assumed suitable for early animal experimentation.)
- The objective of the Development Phase will be to take the technology from Bench Model Converter status to a device appropriate for long-term animal experimentation. This phase would extend over a 3-5 year period and include both long-term in vitro testing and short-term animal in vivo testing.

Component development costs were estimated as follows:

<u>Cost</u> <u>(\$M)</u>	<u>Component</u>
5.0	Gas reciprocating engine
4.5	Vapor reciprocating engine
3.0	Solid electrolyte battery
2.5	Rotary vapor engine
1.5	PCCS (electrical or gas)
1.5	Thermoelectric engine (pure or hybrid)
1.0	Thermoelectric (vapor reciprocating control)
0.5	PCCS (vapor)
0.5	TESM

System-level development costs were determined from the sum of the appropriate component-level development costs.

E-3 DESIGN GROWTH POTENTIAL

The purpose of this criterion was to provide a means of taking into account the technology status and design flexibility of each of the candidate concepts. As for the technology status, a full 10 points was given to the gas and vapor reciprocating concepts already under study by the National Heart and Lung Institute. Since a rotary vapor engine in this size range is not currently under development, a one-point penalty was given all systems employing this component. Three points were subtracted for systems requiring the use of a solid electrolyte battery. For the design flexibility (i. e., options available for performance improvement) a full ten points was given to the hybrid without battery because it provided by far the largest number of design options and potential performance improvements. The lowest score of five points was awarded to the pure thermoelectric system since performance characteristics are very well characterized and significant near-term improvements seem unlikely. Intermediate scores were given to systems under development by NHLI which have demonstrated performance characteristics that have not yet reached predicted levels.

E-4 ESTIMATED UNIT PRODUCTION COST

As with development costs, production costs in quantities of 10,000/year were estimated at the component level and then combined as appropriate to determine system-level costs. A packaging cost of \$1.0K was added to the component-level cost of each system. The component costs are as shown below.

<u>Cost (\$K)</u>	<u>Component</u>
2.0	Gas reciprocating engine
1.5	Vapor reciprocating engine
1.5	Rotary vapor engine
1.0	Thermoelectric (pure or hybrid)
1.0	Thermal converter packaging
0.5	Thermoelectric (vapor reciprocating control)
0.5	PCCS (gas)
0.4	PCCS (electrical)
0.2	PCCS (vapor)
0.2	Solid electrolyte battery
0.1	TESM

Since system-level costs were below \$5.0K in case, all systems received full score, and this became a 'wash' criterion. Development and production cost estimates for each of the candidates are summarized in Table A-2.

Table A-2. System Development and Production Costs

	Development Costs (\$ Million)	Production Costs (\$ Thousand)
Thermoelectric/Battery	6.0	2.6
Rotary Vapor/Battery	7.0	3.1
Hybrid/Battery	8.5	4.1
Hybrid	5.5	3.9
Gas Reciprocating/TESM	7.0	3.6
Gas Reciprocating	6.5	3.5
Linear Vapor/TESM	6.5	3.3
Linear Vapor	6.0	3.2