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25 kWe Solar Thermal Stirling Hydraulic Engine System

Final Conceptual Design Report

Stirling Technology Company

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Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
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for

U.S. DEPARTMENT OF ENERGY
Conservation and Renewable Energy
Office of Solar Heat Technologies

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Washington, D.C. 20545
Under Interagency Agreement DE-AT04-85AL33408

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5. APPENDIXES

ACKNOWLEDGEMENTS

The work reported here represents a cooperative effort with valuable inputs from several subcontractors in support of the Stirling Technology Company contract.

GEDEON ASSOCIATES David Gedeon	Thermodynamic Simulation
PIONEER ENGINEERING & MANUFACTURING COMPANY Dick Heitsch	Manufacturability Assessments and Production Cost Estimates (In process)
SANDERS ASSOCIATES Jim Kesseli	Receiver Design and Analysis
SAASKI TECHNOLOGIES, INC. Elric Saaski	Heat Pipe and Pool Boiler Design and Analysis
WESTINGHOUSE HANFORD COMPANY Jerry Johnson (No cost support courtesy of Westinghouse AESD)	High Temperature Materials Assessment
STIRLING TECHNOLOGY COMPANY Grant Emigh	Stirling Hydraulic Engine Design
Don Lewis	Drafting
Jack Noble	Receiver/Heat Transport/ Heater Head Design
Pete Riggle	Analysis and Systems Integration
Torvald Sorenson	Design and Systems Integration
Linda Stephens	Editing and Report Coordination
Maury White	Project Management

Special thanks to Dick Shaltens of NASA Lewis Research Center for program direction and to Rich Diver and Kevin Linker of Sandia National Laboratories for support in the receiver/concentrator interface area.

Section 1

OVERVIEW

Section 1 provides a program overview which includes an introduction, a summary of the system approach, a description of the Stirling hydraulic engine concept, highlights of system performance, conclusions, and recommendations. Section 2 provides a more detailed description of the system and the conceptual design. Section 3 addresses reliability and maintenance. Sections 4 and 5 include references and appendices. The appendices provide technical detail in specific areas including the complete cost report from Pioneer Engineering in Appendix I.

1.1 INTRODUCTION

The overall objective of this NASA/DOE contract (DEN3-371) is development of a high confidence conceptual design for a free-piston Stirling engine system designed to deliver 25 kW of three-phase electric power to a utility grid when coupled to the 11 meter Test Bed Concentrator (TBC) at SNLA. Further specific objectives include a design life of 60,000 hours, minimum life cycle cost and dynamic balancing.

The approach used to achieve these objectives is a hermetically sealed Stirling hydraulic engine concept based on technology developed to an advanced level during the past 20 years for a fully implantable artificial heart power source. Such engines and critical components have demonstrated operating times in the desired range. This approach provides full film hydrodynamic lubrication of all sliding parts, simple construction with conventional automotive manufacturing tolerances, proven hydraulically coupled counterbalancing, and simple but effective power control to optimally follow insolation variations. This concept maximizes use of commercially available components including hydraulic motors and rotary induction generators which can optionally be mounted at the focus or placed on the ground or behind the mirror to minimize or redistribute suspended weight. The output from several engine concentrator modules can optionally be directed to one large motor/generator.

The final conceptual design is a simple, rugged system which can be prototyped with a high degree of confidence. The design was supported by solid engineering analysis and was carried through to a higher level of detail than is typical for conceptual designs.

1.2 SUMMARY

Section 1.2 briefly describes the major aspects of the conceptual design. It is broken into three subsections which address the overall system, the Stirling hydraulic engine, and the overall performance.

1.2.1 System Approach

The stand-alone version of the advanced solar thermal Stirling power system is depicted in the artist's sketch of Figure 1-1. It illustrates a parabolic concentrator which focuses the incident solar energy into a cavity receiver. This thermal energy is converted by a Stirling hydraulic engine to provide pumped hydraulic fluid which generates conditioned electrical output directly from a commercially proven hydraulic motor and rotary induction generator. A separate fan coil heat exchanger provides the necessary thermodynamic heat rejection to the ambient air. These components can be mounted adjacent to the concentrator focus as illustrated, or any of them except receiver and the Stirling hydraulic engine may be mounted remotely, either behind the mirror or on the ground.

An example of the remote mounting option is illustrated in the artist's concept of Figure 1-2. This shows the central region in an array of twenty engine/concentrator units where the hydraulic output of twenty engines is used to drive six commercial hydraulic motor/induction generator units. The primary motivation for this approach was to realize both economies of scale and some improvement in system efficiency by grouping the components. The final results of the Pioneer cost study however, show that extra piping and other components in the array system actually increase the per kilowatt capital cost by 11.5 percent. The stand-alone version is therefore the standard.

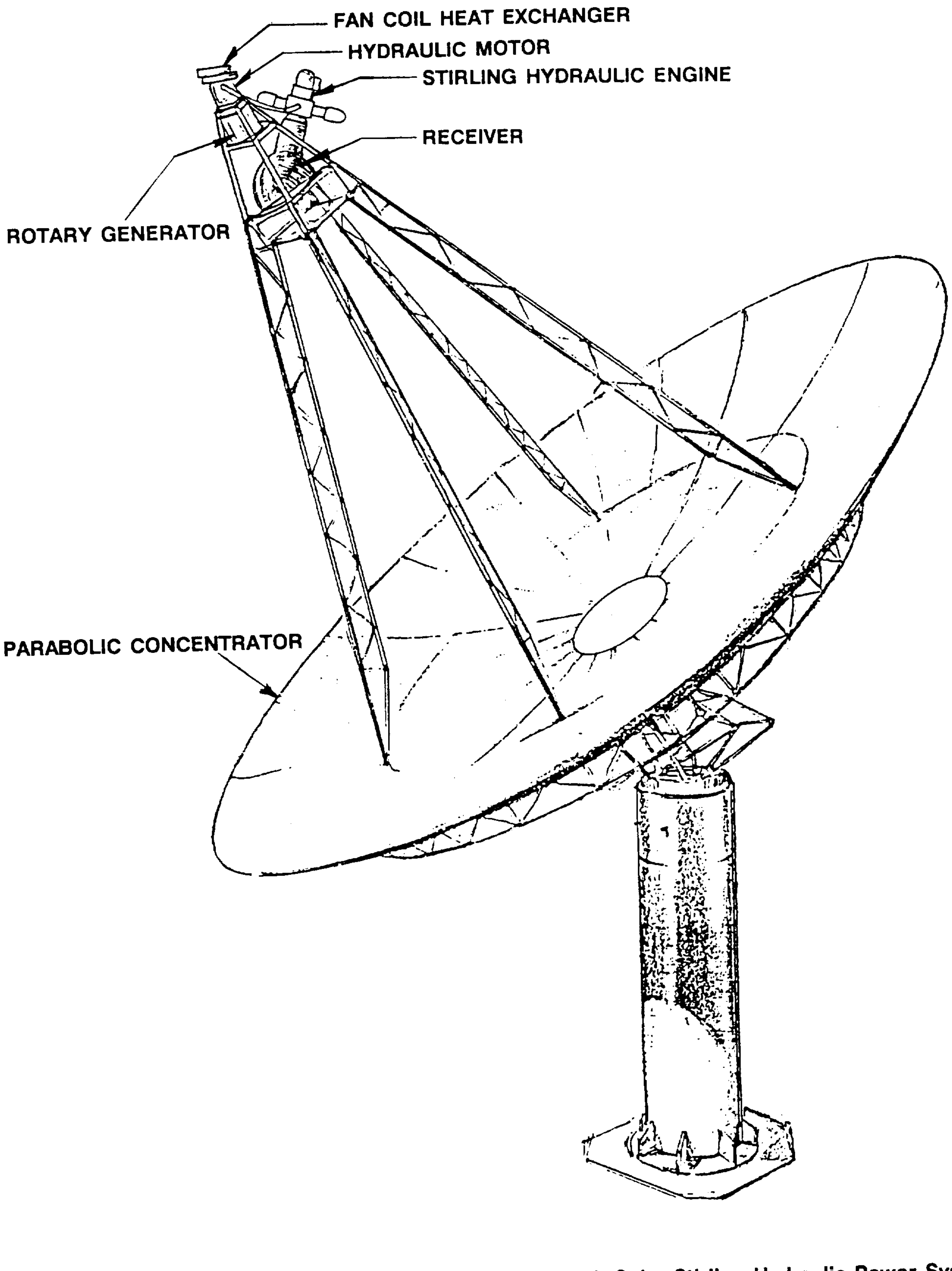

Figure 1-1. Artist's Concept of Stand-Alone Dish Solar Stirling Hydraulic Power System

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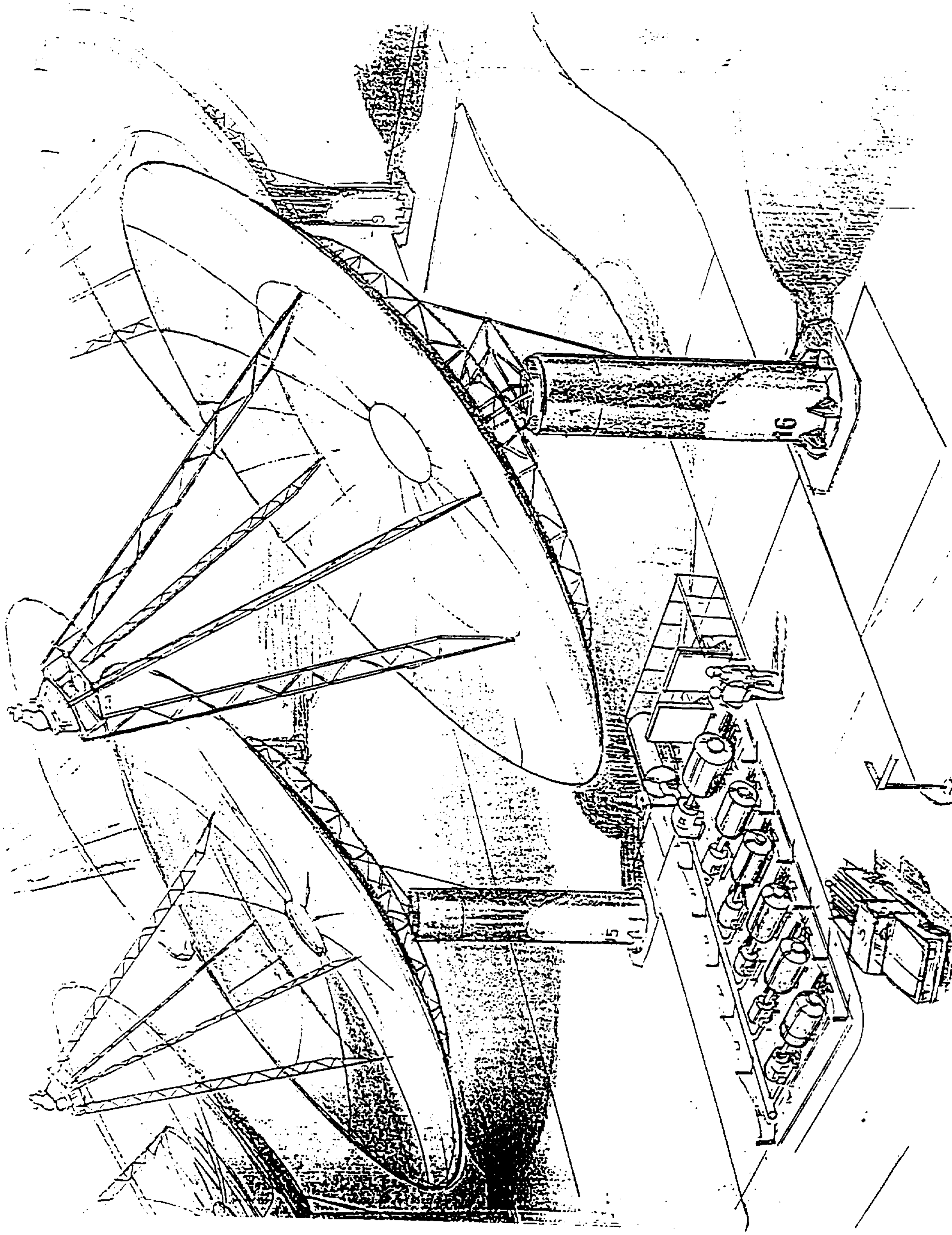

Figure 1-2. Artist's Concept Illustrates Center of Array Using Twenty Engine/Dish Modules

The relationship of the system elements can be visualized by the block diagram of Figure 1-3. The insolation reflected by the concentrator is absorbed in the receiver where it is transferred to the Stirling engine heater head by a simple, rugged and reliable, pool boiler reflux heat pipe. Pulsatile hydraulic flow from the Stirling hydraulic engine is smoothed by the pulsation suppressors for use by the hydraulic motor. Engine waste heat is rejected to the atmosphere by the cooler. A simple, energy conservative, automatic control system adjusts engine power to maintain constant hot end temperature over a wide range of insolation power levels. The output shaft of the hydraulic motor couples directly to the induction generator, which easily switches to or from the grid and produces inherently conditioned power. The cooler, surge suppressors, hydraulic motor, and induction generator are all proven, reliable, commercially available components. Thus, major portions of the system have an established track record and provide total confidence in their performance and cost parameters.

Figure 1-4 illustrates the system configuration on the left side with a simplified representation of the major components shown on the right. The receiver at the bottom left absorbs the solar flux from the concentrator on a surface backed by liquid potassium. The heat boils the potassium which then transfers the heat to the Stirling engine by condensing on the heater tubes. This heat transfer mechanism is described as a reflux boiler heat pipe. It is analogous to the function of a conventional double boiler in which heat from a cook stove boils water in the lower kettle. The water vapor condenses on the bottom of the upper kettle to provide uniform heat at a constant temperature.

The STC Stirling hydraulic engine is a simple but dynamically stable free-piston engine which, in small sizes, has demonstrated years of maintenance-free operation without performance degradation. It is functionally described in Section 1.2.2 with more detailed description in Section 2. Scaling evaluation of the basic Stirling cycle, fluid flow losses, and bellows dynamics show that the proven small engines can be scaled to the required power level.

The high pressure hydraulic fluid output from the Stirling hydraulic engine is used by commercially proven components to generate the desired three-phase

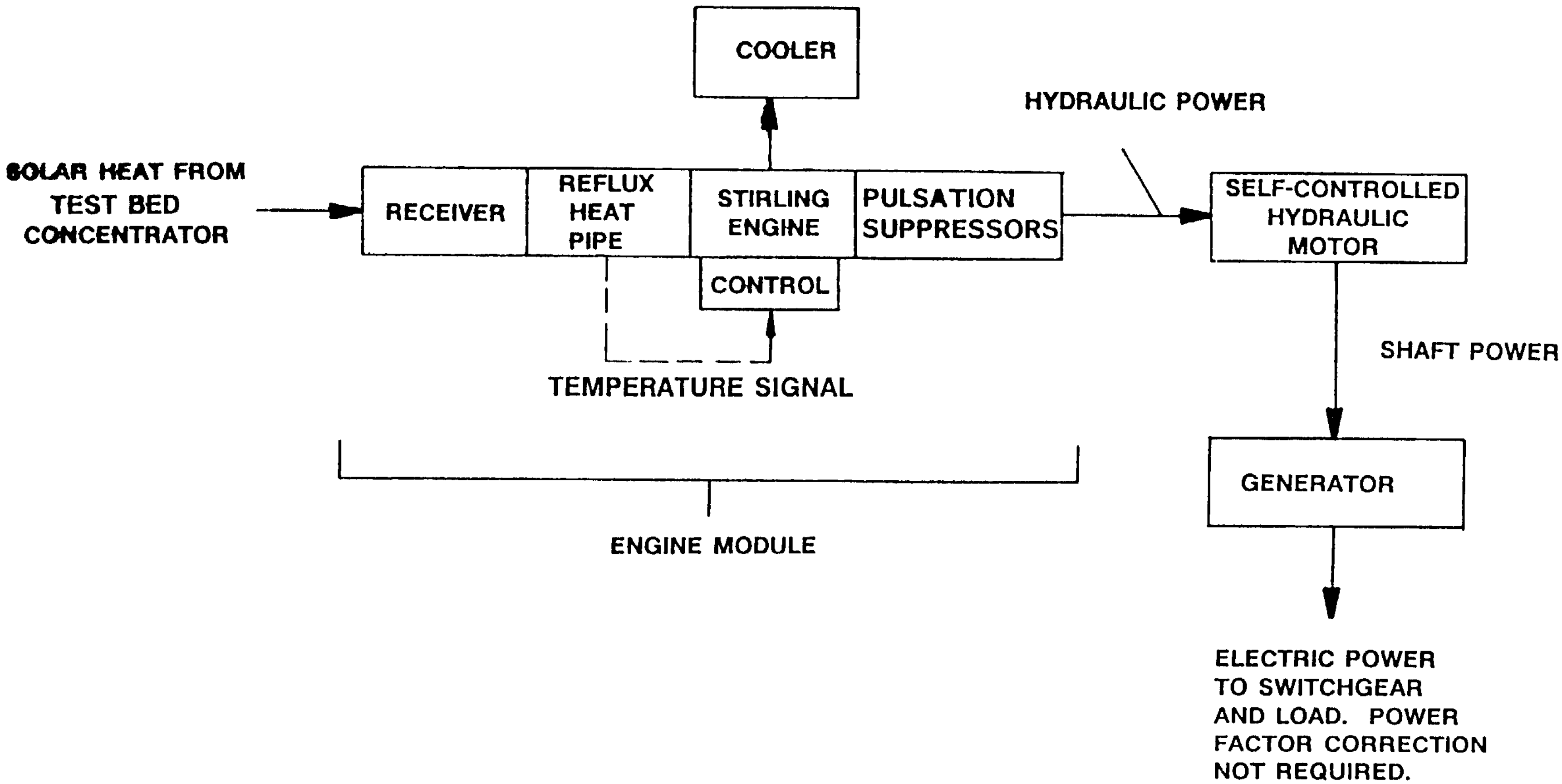

Figure 1-3. Block Diagram of Stand-Alone Engine Generator System

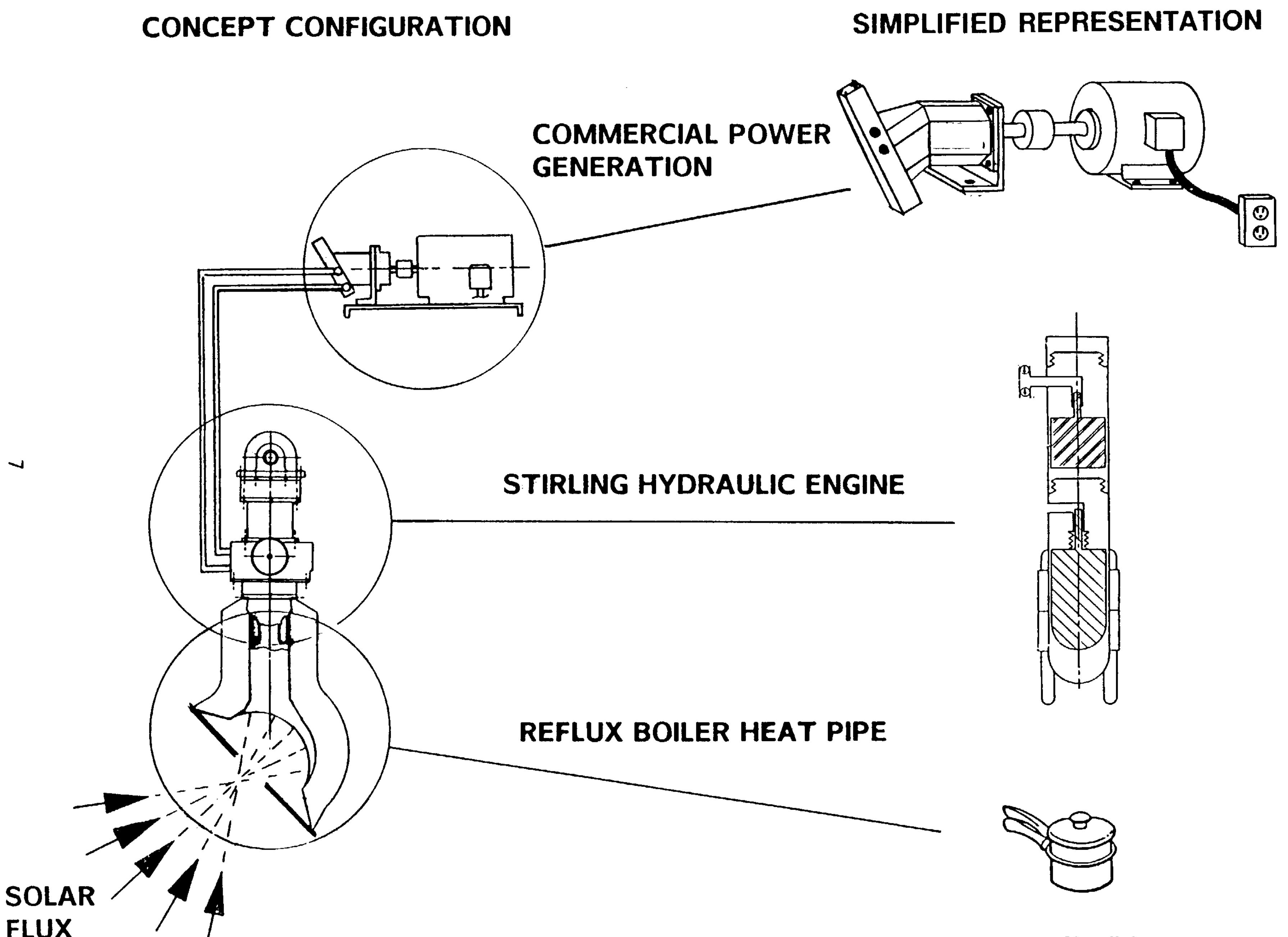

Figure 1-4. Breakout of STC System Components Illustrates Inherent Concept Simplicity.

electric output with low harmonic distortion and high power factor. These components include a Volvo hydraulic motor connected to a GE induction generator which are off-the-shelf components with field proven reliability, performance, and cost.

These elements combine to produce a high confidence, high reliability, high performance, cost effective system design which can be developed with minimal risk.

1.2.2 Stirling Hydraulic Engine

A free-piston Stirling engine which delivers power as pumped hydraulic fluid is referred to in this report as a Stirling hydraulic or STIRLIC™ engine. This is the key developmental area of the design developed under this contract. The free-piston Stirling hydraulic engine is quite simple. A functional comparison with the more familiar Free-Piston Stirling Engine/Linear Alternator (FPSLA) is illustrated in Figure 1-5. The heater, cooler, regenerator and displacer for both systems are conceptually identical.

The difference in displacer drives is that, whereas the FPSLA drive rod resonates through a gas clearance seal against the displacer gas spring with no positive means of amplitude stabilization and control, the STIRLIC™ drive rod is hydraulically coupled to the displacer gas spring by means of the stabilizer/controller which stabilizes the displacer amplitude. Throttling this hydraulic coupling flow with a spool valve provides a well proven, remarkably simple and energy efficient method of engine speed and power control over a turndown ratio of five or more.

The stabilizer/controller prevents damaging overstroking of the displacer and consequently the power piston under all operating conditions and load changes. It also perfectly counterbalances the displacer, since it is on axis with the displacer in the conceptual layout design.

The power pistons are conceptually the same, with a variation in how power is extracted. The FPSLA typically attaches permanent magnets (or, moving copper or moving iron) to the power piston which interact with electromagnetic fields to generate electricity. The Stirling hydraulic power piston has an integral

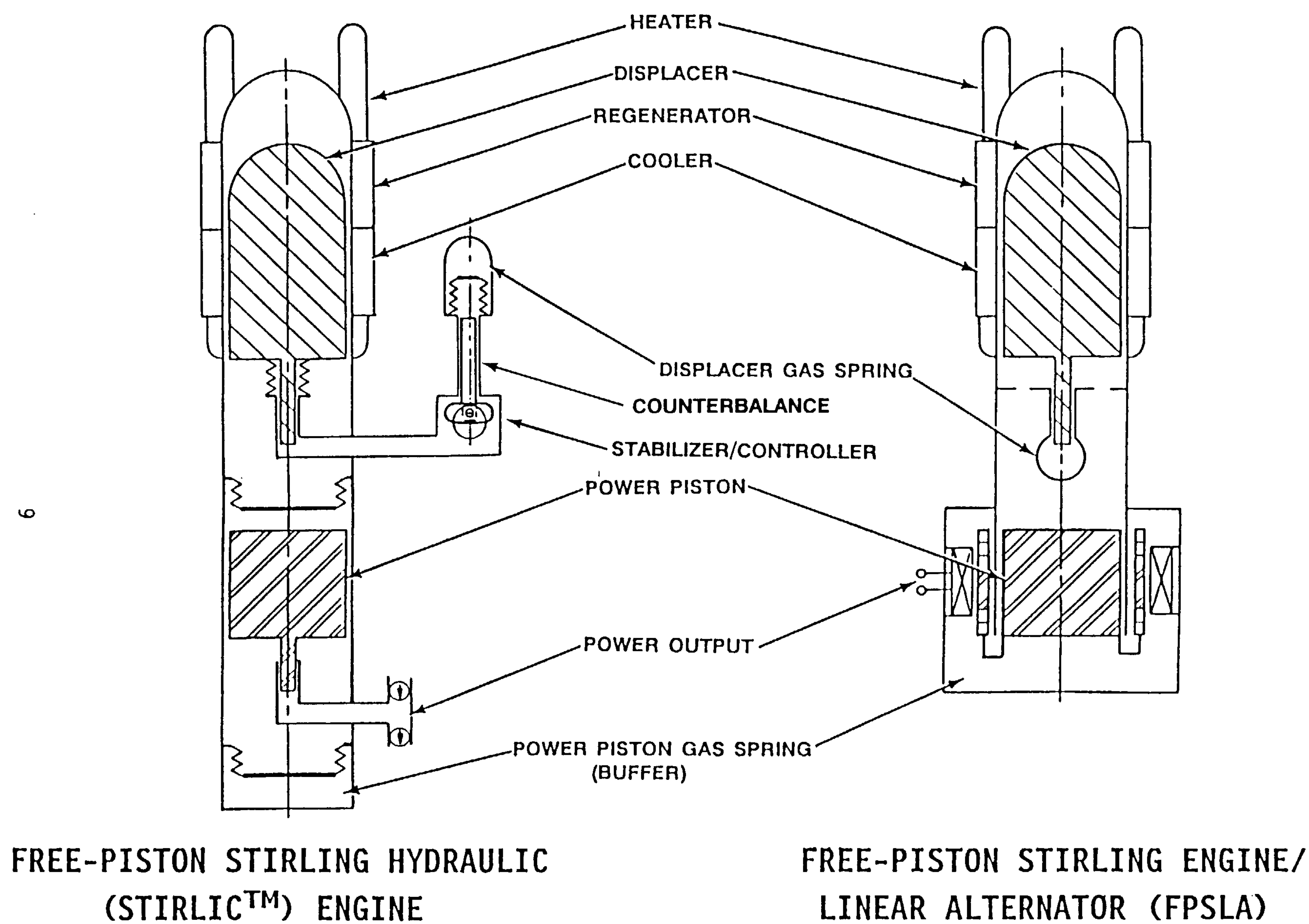

Figure 1-5. Schematic Functional Comparison of Alternative Concepts Illustrates Basic Similarity

pumping intensifier piston which directly produces high pressure hydraulic flow. As with the displacer drive rod, the FPSLA power piston operates entirely in the helium working fluid with gas bearing support and clearance seals while the STIRLIC™ power piston operates in hydraulic fluid with hydrodynamic lubrication and hermetic bellows seals.

In both cases, the motion of the free power piston is accommodated by the power piston gas spring or buffer gas volume. One advantage of the Stirling hydraulic concept is that the hermetic bellows seals allow use of an optimum gas in the buffer and displacer gas springs. This can reduce the gas spring hysteresis losses by as much as 97 percent (Reference 1) and/or allow a reduction in the size and therefore cost of the gas spring pressure vessels.

1.2.3 Performance Summary

The key performance parameters for the solar thermal Stirling hydraulic system are summarized in Table 1-1. The insolation levels are as specified in RFP3-117122, Conceptual Design of a Solar Electric Advanced Stirling Power System. The engine was designed to operate continuously at the peak insolation of 1100 W/m^2. The specific requirement was for the engine to survive this flux for 15 minutes, but designing it to operate continuously at this level allows generation of significantly more kW-hr per year. The nominal receiver input power was specified in the RFP as 75 kW for an insolation of 950 W/m^2. This linearly extrapolates to 86.8 kW at the peak survival flux. According to analysis by Sanders Associates, at the controlled hot engine temperature of 700°C, the receiver loses 7.5 kW regardless of heat input level. This determines engine heat input for the two conditions in Table 1-1.

The output to the grid is determined on the basis of thermodynamic engine analysis by Gedeon Associates and STC, hydraulic losses, and published commercial specifications for the hydraulic motor, rotary induction generator, and fan coil heat exchanger. The net result of 25.2 kW delivered to the grid for nominal conditions is right on the target objective of 25 kW, while the peak output of 29.6 kW maximizes the overall annual system effectiveness.

Component efficiencies relating to the above energy flows are also provided in Table 1-1. Other pertinent information is highlighted at the bottom.

Table 1-1

SUMMARY OF KEY PERFORMANCE PARAMETERS

	DESIGN POINT OPERATION (SURVIVAL POWER)	NOMINAL POWER OPERATION (75 kW TO RECEIVER)
Insolation W/m^2	1100.0	950.0
Receiver Heat Input kW	86.8	75.0
Engine Heat Input kW	79.3	67.5
Output to Grid kW	29.6	25.2
Receiver Efficiency %	91.4	90.0
Engine-Generator Efficiency %	37.3	37.3
Receiver-Engine Generator Efficiency %	34.1	33.6

<u>System Annual Net Output</u>

65,200 kW hr to Grid

<u>System Annual Gross Input (from RFP)</u>

206,800 kW hr

<u>Annualized Energy Efficiency</u>

$\bar{\eta}$ = 65,200/206,800 = 31.5%

<u>Stand-Alone System Weights</u>		
Suspended Weight	320 kg	705 lb
Optionally Suspended or Ground Based Weight	549 kg	1209 lb
<u>Weights Per Engine for 20 Engine Array</u>		
Suspended Weight	320 kg	705 lb
Ground Based Weight	807 kg	1770 lb

The net annual system output of 65,200 kW-hr is based on the annualized insolation table provided in the RFP. It represents the maximum practical level obtainable by any machine with the indicated efficiencies, since it takes advantage of virtually all insolation levels with a highly efficient power control mechanism.

1.3 CONCLUSIONS

Conclusions reached in development of the dish solar Stirling hydraulic engine concept are the following:

- All of the technical requirements of the RFP are satisfactorily addressed by the present conceptual design.

- The present conceptual design is based on technology which has been proven on other programs and/or in the commercial marketplace.

- The STC free-piston engine is similar in many ways to gas bearing free-piston engines, but incorporates specific, distinct improvements. STC suggests that these improvements be given significant weight in the comparison of the STC engine with gas bearing engines.

- The simple hermetically sealed Stirling hydraulic engine is based on 20 years of development of reliable long-life engines. Engines of this type have demonstrated unattended operating times in the range of 60,000 hours with completely internal lubrication and makeup systems.

- The primary concerns of scaling from small engines with long term life tests to the engine for the present 30 kW system involve the Stirling engine gas circuit, bellows dynamics, and hydraulic fluid flow losses. These concerns have been addressed in the present design.

- The heat transport system is a simple potassium reflux boiler, which is inexpensive, rugged, predictable, and provides excellent performance.

- Potassium is clearly preferred over sodium as the heat transport medium of choice for the 700°C heat source.

- The stand-alone configuration and the multi-engine-generator array configuration both meet the suspended weight criteria of the solicitation.

- Engine-generator efficiency is 37.3%. Receiver-engine-generator efficiency is 34.1%.

- Annual net energy generation is 65,200 kW hr.

- Demonstrated engine balancing methods completely eliminate vibration at all operating conditions.

- System operation is fully automatic.

- A simple and lightly loaded stabilizer/controller eliminates stability and control problems which often complicate operation of non-stabilized free-piston Stirling engines.

- A simple control provides smooth and efficient variation of engine power from very low levels to peak survival conditions while precisely regulating receiver temperatures.

- In the stand-alone version the hydraulic motor self regulates. External control is not required.

- The rotary induction generator requires no control.

- The clearance between the displacer and the cylinder liner is an effective displacer seal.

- The baseline concept employs a proven porous wire screen regenerator.

- The selected concept is highly manufacturable. Piston diametral clearances up to 1.5 mils per inch of diameter are acceptable.

- The engine and generator require zero maintenance for 60,000 hours of operation. The hydraulic motor requires less than 12 man hours of maintenance in 60,000 hours of operation. Although the engine is designed for zero maintenance, replacement of most engine components is possible.

- The system is designed to handle operation at survival power for the entire 60,000-hour life.

- The system employs a commercial hydraulic motor, a commercial rotary induction generator, and a commercial cooling system. These commercial components provide the advantages of zero development cost and well characterized performance, life, and operating characteristics.

- Hydraulic power systems have proven to be very reliable and easy to use in field applications, as exemplified in Appendix H.

Extensive design trades were conducted for the heat transport system before selecting the simple, rugged, reliable pool boiler approach. Several versions of wicked and reflux heat pipes were evaluated, but it was concluded that they were more costly (fine mesh screen), more complex (wick and artery installation), and less reliable (wick imperfections, burnout and thermal cycling sensitivity). For space applications, these problems are minimized by lack of pumping against a gravity head. Since a reflux boiler requires gravity, a wicked heat pipe is the obvious choice for space applications. For terrestrial use in limited orientations with high heat fluxes and low cost objectives, the pool boiler is a similarly obvious choice.

Early hot end work on this contract used a nominal 800°C operating temperature, but practical materials problems led to reducing hot end temperature to 700°C. Sodium was clearly the optimum choice for heat transport at 800°C, but at 700°C its low vapor pressure and consequent low mass transport capability make sodium impractical. Potassium, on the other hand, has excellent properties at 700°C, making it the obvious choice.

The Stirling hydraulic engine design went through a series of iterations to improve manufacturability and to make it more cost effective. The final design, as described in Section 2, is a simple design with relatively loose tolerances and large clearances which should adapt well to reasonable cost mass production. The engine technology is well grounded with years of maintenance-free operating experience on small Stirling hydraulic engines providing a high level of confidence that design objectives can be met.

One of the major advantages of the Stirling hydraulic concept is that the power generation mechanism consists entirely of highly refined field proven commercial components requiring no development. Hydraulic motors have demonstrated long life and high reliability in such diverse and demanding environments as food processing facilities, sewage treatment plants, and heavy construction and farming equipment. Rotary induction generators (identical with induction motors) are used in a very wide variety of applications. One of the most relevant uses is with thousands of wind turbines where the generator output is connected to the grid.

1.4 RECOMMENDATIONS

This contract has resulted in a conceptual design with outstanding potential for meeting the specified reliability and performance objectives. The stability, control, conventional tolerances and manufacturing, fully developed power generator, straightforward pool boiler heat transport, and proven multiyear lifetime of conceptually similar engines cannot be duplicated by any known alternative system. The following categories of recommendations can be objectively supported by the results of the contract.

<u>General Recommendations</u>

1. Implement optional Task 3, ASCS Reference Design. This will allow consolidation of all design options considered into a complete and fully consistent set of layout drawings, with consideration for the results of the manufacturing cost and manufacturability study.

2. Initiate follow-on effort to complete the detail design, fabrication and testing of a prototype Stirling hydraulic system suitable for testing on the Sandia National Laboratories 11 meter Test Bed Concentrator.

Specific Recommendations

1. Implement coupon tests of CG-27 in a closed potassium reflux capsule which simulates the surface-to-volume ratio and material combinations to be used in the end system.

2. Conduct heater tube braze joint compatibility tests under conditions similar to those for the CG-27 coupon tests.

3. (Long term option for maximum cost and performance potential.) Conduct technology development of the annular foil regenerator concept evaluated for the Preliminary Design Review. Such development should include demonstration of fabrication practicality and measurement of regenerator performance in a suitable test rig.

Section 2

CONCEPTUAL DESIGN DESCRIPTION

Section 2 of the report discusses all aspects of the conceptual design. Topics include the system approach, system performance, receiver, reflux heat pipe, Stirling Hydraulic engine, commercial components, system integration, and controls.

2.1 SYSTEM APPROACH AND PERFORMANCE

This subsection explains the overall approach from a system viewpoint, discusses the advantages provided by the chosen system, and briefly summarizes overall performance. The numerous and significant advantages specific to the Stirling hydraulic engine, are discussed in Section 2.2, Engine Module Design.

2.1.1 System Approach

The STC conceptual design takes advantage of a remarkable long life Stirling engine technology demonstrated for artificial heart and portable compressor applications. In the selected system approach, Stirling engines produce hydraulic power which is converted to electric power by motor-generator sets employing commercial hydraulic motors driving commercial induction generators. The system can be configured as a stand-alone power plant, in which a Stirling engine, a commercial hydraulic motor, and a commercial induction generator are supported by the concentrator, or can be configured as an array in which the hydraulic output from twenty engines is connected in parallel to feed six centrally located ground mounted motor-generator sets.

At the conclusion of the technical effort, STC selected the array configuration as the baseline design over the stand-alone configuration partly because the cost per kilowatt of motor-generator components, particularly the hydraulic motor, is reduced in larger component sizes. Also, five of the six hydraulic motors in the array are fixed displacement motors with higher efficiency than the variable displacement motor used with the stand-alone configuration. This gives the array system an efficiency advantage of about one percentage point over the stand-alone unit. Basing the motor-generator sets on the ground to minimize the concentrator supported weight is another point supporting the array as the baseline concept. However, the supported weight for both the stand-alone and the array system is within the limits established for the Test Bed Concentrator.

The array configuration was the baseline design for the cost analysis by Pioneer and the stand-alone configuration was an alternative. Final results of the Pioneer study, included in total in Appendix I, concluded that the stand-alone system offered manufacturing costs 10.3% below those of the array system. Therefore, the Stirling hydraulic stand-alone configuration is preferable to the array.

The stand-alone system is be discussed before the array. The stand-alone system is shown in the artist's concept in Figure 2-1 and in the block diagram of Figure 2-2.

As shown in Figure 2-2, heat delivered to the receiver is used by the engine to produce a flow of hydraulic fluid which drives a hydraulic motor coupled directly to an induction generator. Control is simple. A key advantage of this concept is a remarkably simple engine control system which varies engine frequency, thereby modulating heat input and power output. High efficiency is provided over a very wide insolation range, taking advantage of very low to very high levels of incident solar heating. The energy efficient engine power output turndown ratio exceeds 5 to 1. The controlled variable of the engine control system is the temperature of the potassium surrounding the heater head. The engine control system precisely regulates heater head temperature, which is important because all candidate heater head materials lose strength dramatically above the design operating temperature. The alternatives to precise control of heater head temperature are premature heater head failure or lower efficiency than predicted, both of which are unacceptable.

Pursuing the subject of system control further, the engine produces a flow rate of hydraulic fluid which is roughly proportional to the heat input rate

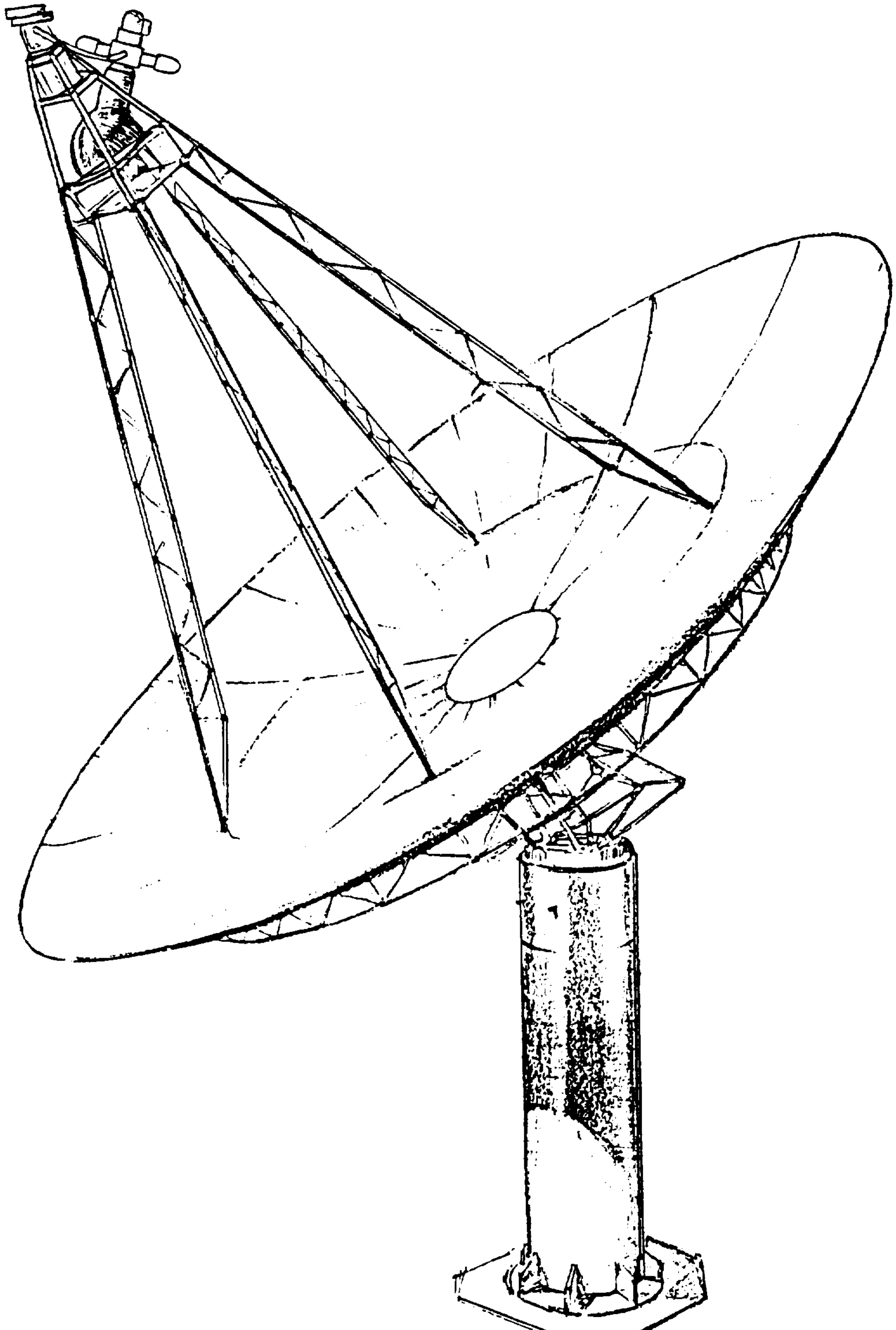

Figure 2-1. Artist's Concept of the Stand-Alone System

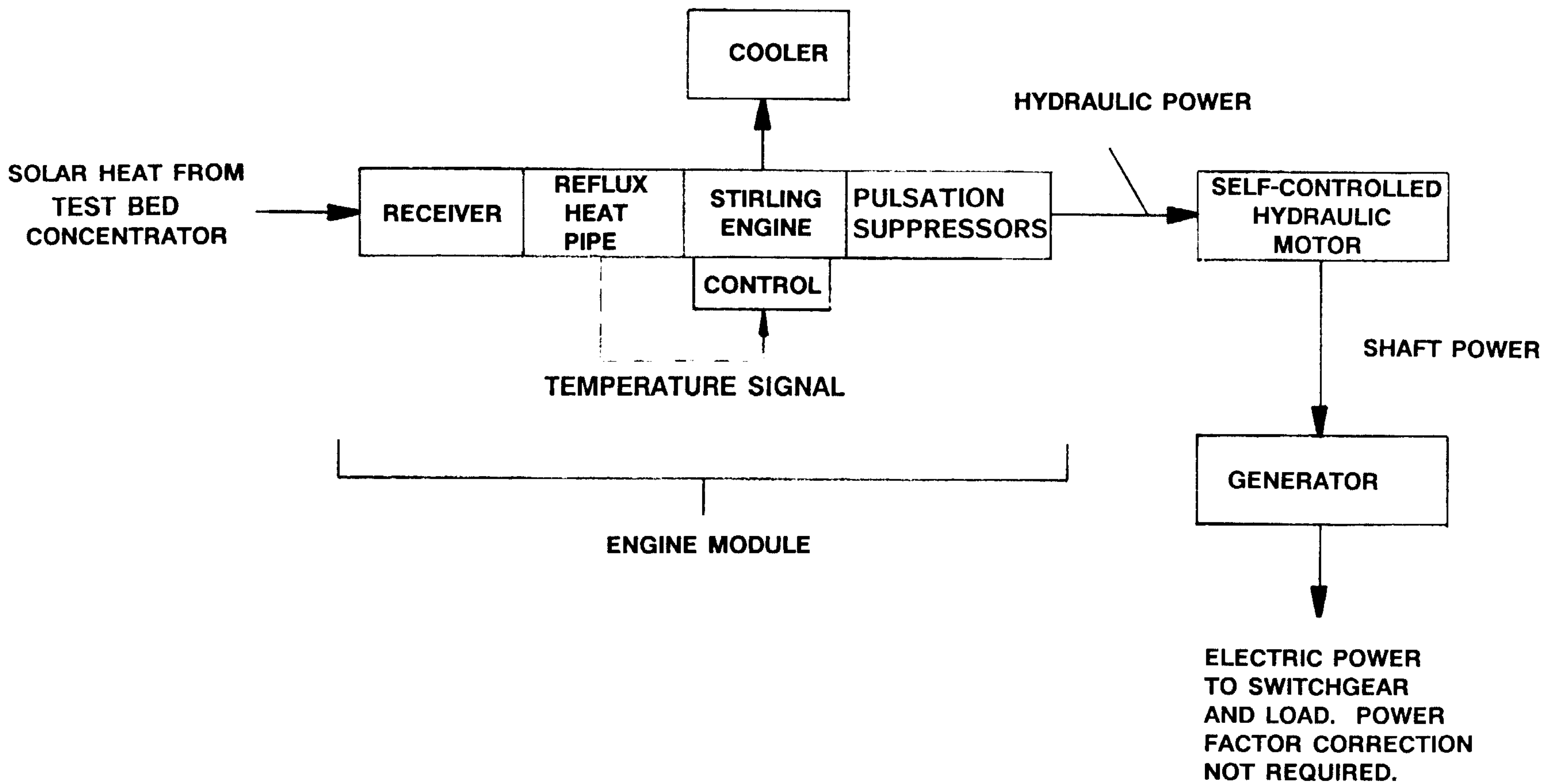

Figure 2-2. Block Diagram of Stand-Alone Engine Generator System

delivered by the receiver. In the stand-alone system shown in Figure 2-1, the variable displacement hydraulic motor self regulates by using a factory engineered control option to maintain constant hydraulic system pressure difference and adjusts to the flow rate delivered by the engine. The motor-generator set runs at a constant pressure difference established by the hydraulic motor controller and slightly above 1800 rpm as established by the slip speed of the induction generator. Thus heat input is transformed by the engine to variable hydraulic flow rate at constant pressure. This flow is transformed to variable current at constant line voltage by the combination of the self-controlled hydraulic motor and the induction generator, both operating near generator synchronous frequency. As with any rotary induction machine, the generator can be switched onto the grid with no concerns for phasing and will operate as a motor or generator, slightly below or above synchronous speed, depending on the torque applied to the generator shaft.

The array configuration is shown as an artist's concept in Figure 2-3 and as a block diagram in Figure 2-4. An array of twenty engine modules, each mounted on its respective concentrator, supplies hydraulic power to an array of six motor-generator sets located on the ground in a location central to the cluster of concentrators.

Control of the array is very similar to control of the stand-alone system. The difference is that the combined output of twenty engines is consumed entirely by up to six hydraulic motors. This is accomplished by a motor-generator array controller which senses the hydraulic pressure delivered to the hydraulic motors and adjusts the combined hydraulic motor displacement of the array by valving fixed displacement hydraulic motors into and out of operation. The combined hydraulic motor displacement is increased or decreased as required to maintain the one variable displacement hydraulic motor within its pressure control band, where it can automatically regulate its own displacement to accommodate small variations in flow. In summary, the fixed displacement motors turn on and off to handle large flow variations. The variable displacement motor modulates to handle small flow variations. In the array concept, heat is transformed into variable flow rate at constant pressure by an array of engines, and flow rate is transformed into variable

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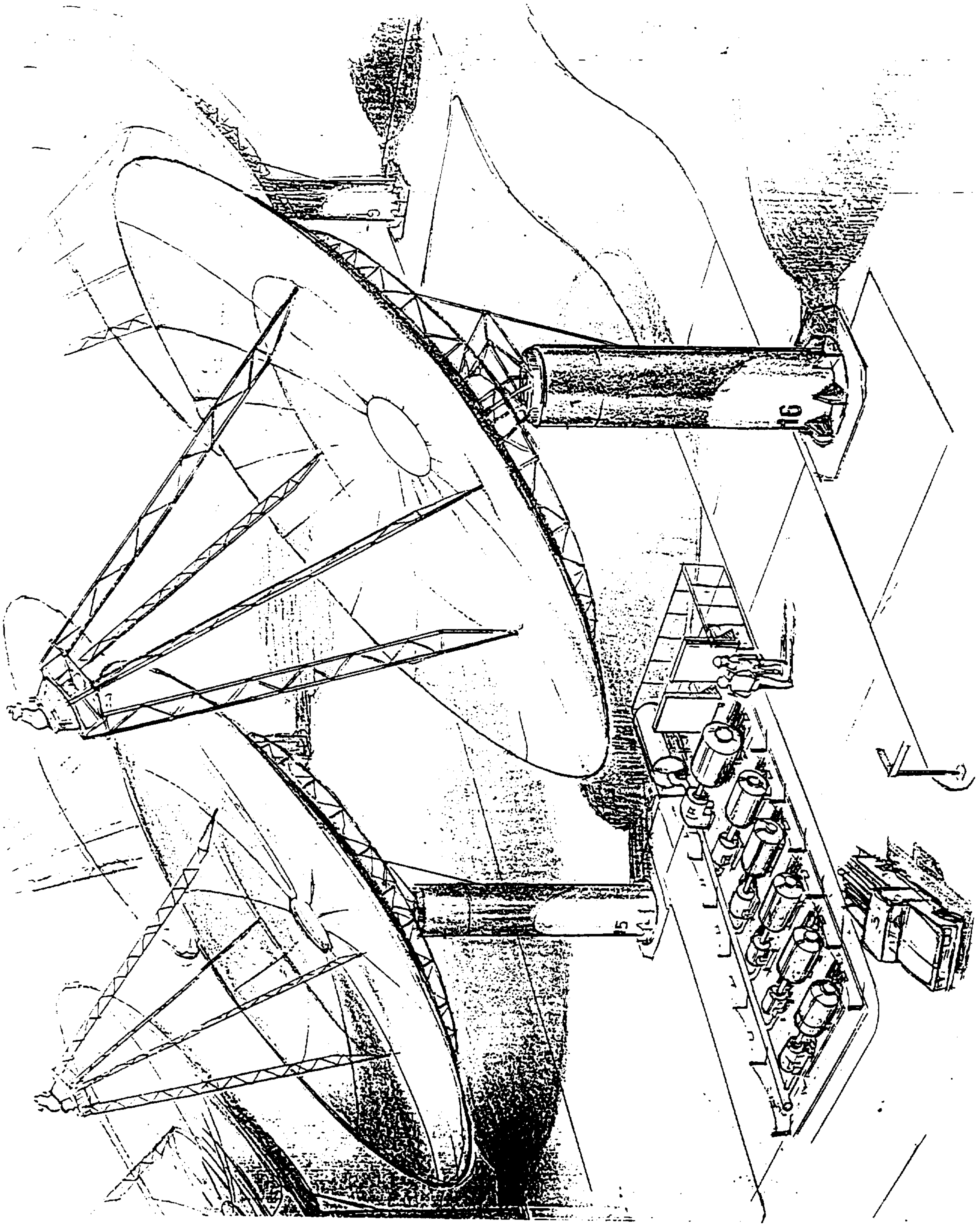

Figure 2-3. Artist's Concept of the Array System

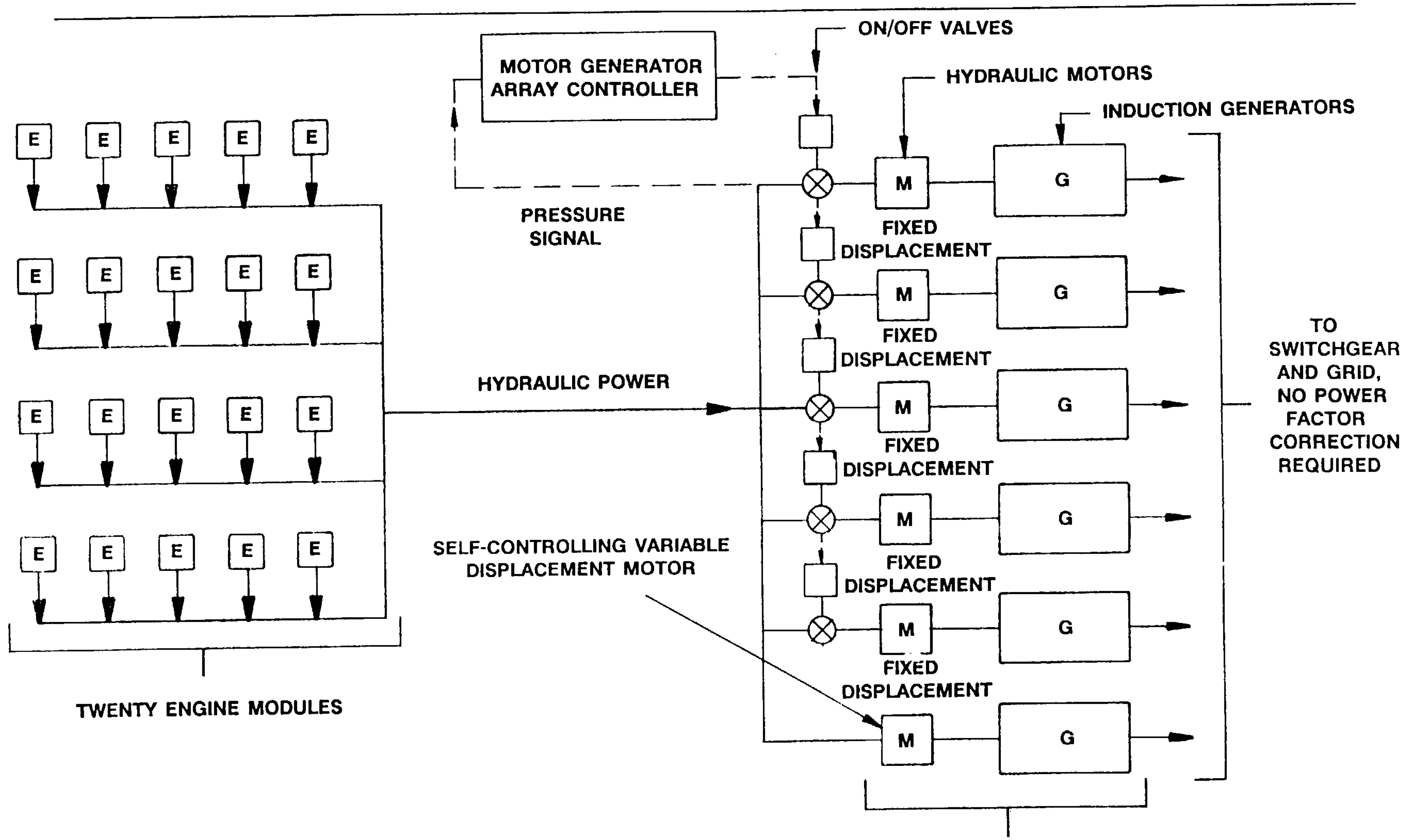

Figure 2-4. Block Diagram of Engine Generator Array

electric current by a variable displacement array of motor-generators operating at the constant voltage and frequency provided by the grid.

2.1.2 System Performance

System thermodynamic performance is summarized by Figure 2-5 which shows the total insolation heat flow to the concentrator, the heat flow to the receiver, the heat flow to the engine, and the net electric output to the grid. These parameters are plotted in kW as functions of the time in hours power is available above a given level. Areas under the curve represent annual energy in kilowatt-hours.

The design parameters of the engine and power generation equipment were established to allow routine operation of the system at the survival power level. In addition, the excellent stability and control provided by the engine concept allow the engine to operate routinely at any available power down to very low levels. Thus the ability of the system to utilize the entire spectrum of insolation levels is outstanding.

Two particular system power levels are identified on the graph. The first of these is the survival power case with 86.8 kW available to the receiver and nearly 30 kW delivered to the grid. The second is the nominal power case with 75 kW available to the receiver and just above 25 kW delivered to the grid.

Table 2-1 lists key power throughput and efficiency terms for design point operation and nominal power operation. The overall efficiency is about 34% for the combination of the receiver, the engine, and the generator, which are the components defined by the conceptual design process. Weights are given for the stand-alone configuration and the 20 engine array configuration. In the stand-alone configuration, the 320 Kg engine module must be suspended at the receiver mounting ring. The remaining 549 Kg can be suspended at the mounting ring, behind the mirror, or on the ground. The ground based weight for the array is larger than that for the stand-alone configuration primarily because of pipe runs. The weights given in Table 2-1 tend to be estimated on the high side.

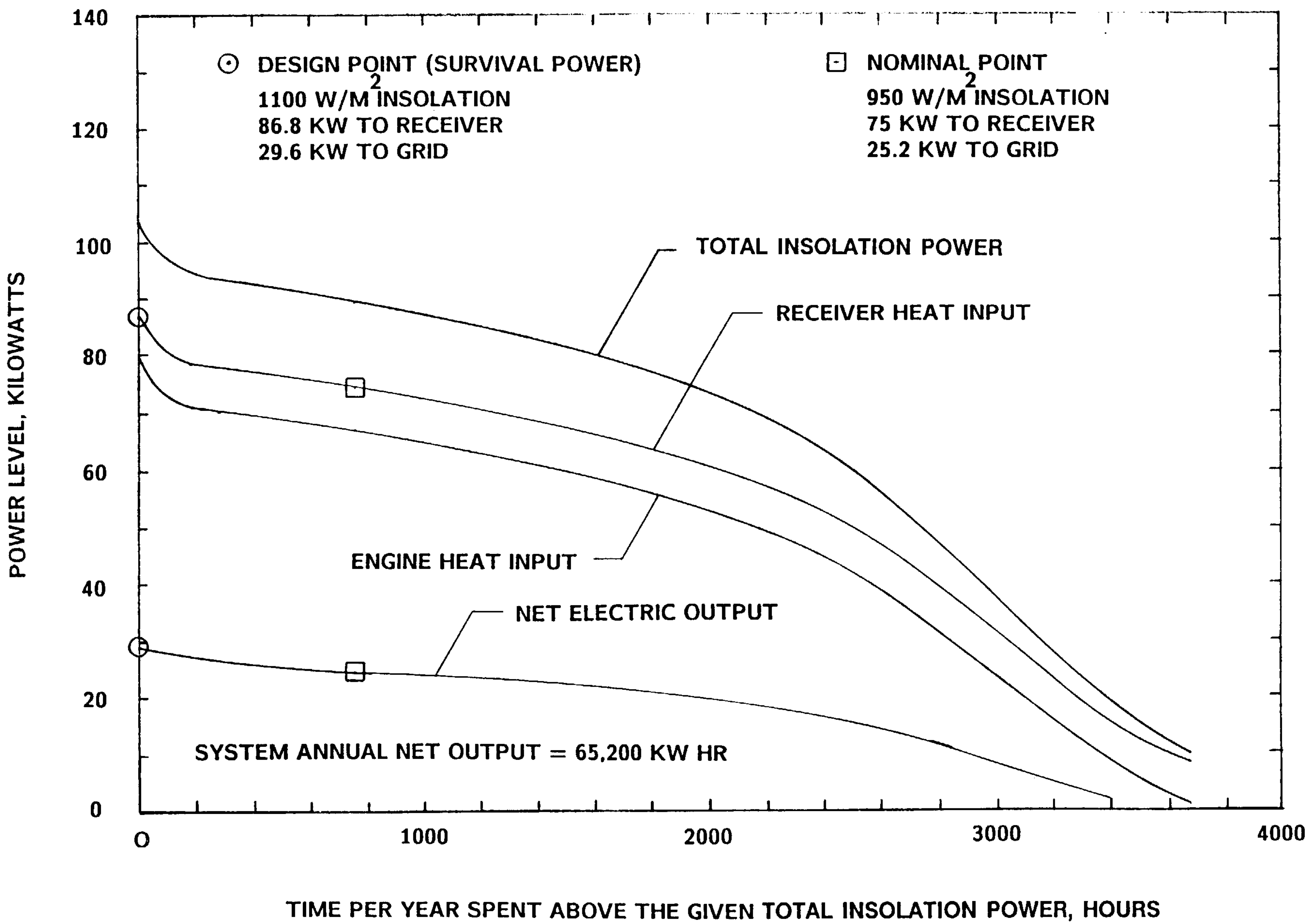

Figure 2-5. System Thermodynamic Performance

Table 2-1

SUMMARY OF KEY PERFORMANCE PARAMETERS

	DESIGN POINT OPERATION (SURVIVAL POWER)	NOMINAL POWER OPERATION (75 kW TO RECEIVER)
Insolation W/m^2	1100.0	950.0
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Engine-Generator Efficiency %	37.3	37.3
Receiver-Engine Generator Efficiency %	34.1	33.6

System Annual Net Output

65,200 kW hr to Grid

System Annual Gross Input (from RFP)

206,800 kW hr

Annualized Energy Efficiency

$\bar{\eta}$ = 65,200/206,800 = 31.5%

Stand-Alone System Weights

Suspended Weight	320 kg	705 lb
Optionally Suspended or Ground Based Weight	549 kg	1209 lb

Weights Per Engine for 20 Engine Array

Suspended Weight	320 kg	705 lb
Ground Based Weight	807 kg	1770 lb

Annual energy delivery to the grid is 65,200 kW-hours. The annual solar insolation distribution was specified in the original request for proposal (RFP). The specified insolation ranges and the time spent in each range are repeated from the RFP in Table 2-2. Also included in Table 2-2 are the thermal input at the mid-point of each range and the net electrical output associated with each thermal input level. Thermal input values were calculated by a linear extrapolation from the nominal operating point of 75 kW thermal input to the receiver at an insolation level of 950 W/m^2. Output power was determined by subtracting all fixed heat losses at 700 C, then multiplying thermal input by all component efficiencies at the given power level. Annualized thermal input and net electrical output were determined by plotting the thermal input and net output as function of the number of hours in a year and integrating the areas under the curves, as shown in Figure 2-5. These figures were determined by the evaluation of system efficiency at all operating power levels, in conjunction with the annual solar insolation distribution as described below.

2.2 ENGINE MODULE DESIGN

This section discusses the design of the engine module, shown in outline form in Figure 2-6. It consists of the receiver, the reflux boiler heat pipe, and the Stirling hydraulic engine. Topics covered include the receiver and reflux boiler heat pipe, the Stirling hydraulic engine, evaluation of materials considered in the design, computer simulation, and other analyses performed in support of the design.

2.2.1 Receiver and Reflux Boiler Heat Pipe

Alternatives investigated for the heat transport system include the reflux boiler heat pipe, discussed in Appendix A, the capillary heat pipe, discussed in Appendix B, and other concepts, discussed in Appendix C. Receiver analysis is discussed in Appendix A. A chart comparing the various concepts considered is presented as Figure 2-7. The reflux boiler was chosen over the other heat transport system alternatives for the reasons given below. All other candidates had serious disadvantages.

Table 2-2

ANNUAL SOLAR INSOLATION DISTRIBUTION AND RESULTANT POWER LEVELS

INSOLATION WATTS/m^2	TIME HOURS/YEAR	THERMAL INPUT AT MEAN ISOLATION kW	NET ELECTRICAL OUTPUT kW
0 to 99	5,091	3.9	-
100 to 199	276	11.8	1.5
200 to 299	201	19.7	4.4
300 to 399	216	27.6	7.4
400 to 499	181	35.5	10.3
500 to 599	229	43.4	13.2
600 to 699	261	51.3	16.1
700 to 799	444	59.2	19.0
800 to 899	674	67.1	21.9
900 to 999	1,021	75.0	24.8
1,000 to 1,099	168	82.9	27.8

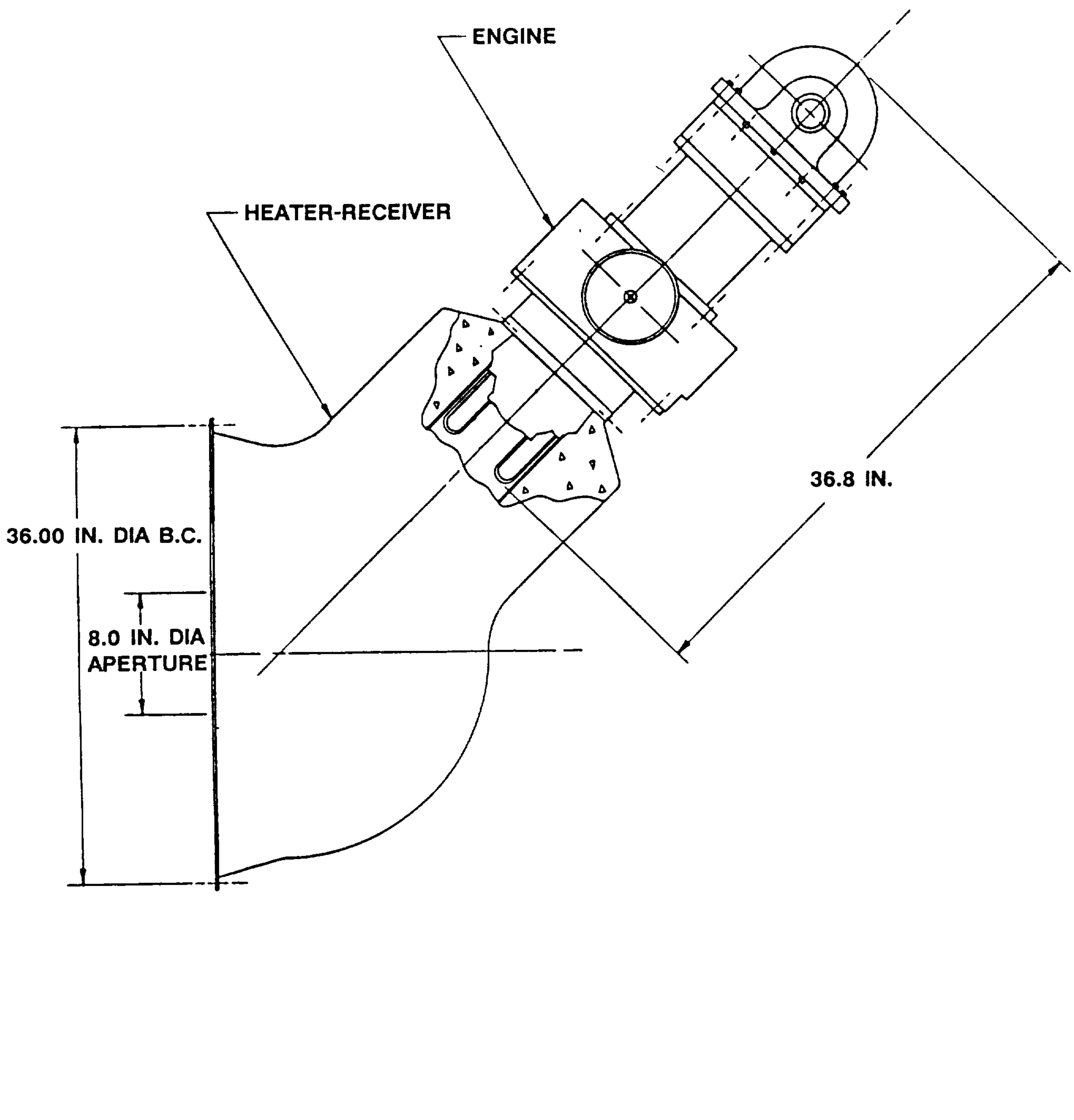


Figure 2-6. Engine Module Outline

	A WICKED HEAT PIPE SELF-PRIME *	B WICKED HEAT PIPE NON SELF-PRIME *	C** POOL BOILING HP (REFLUX BOILER)	D POOL IMMERSION	E TWO-PHASE BOILING	F CONVECTIVE TRANSFER	G PUMPED LOOP
ADVANTAGES	• SATISFIES REQMTS	• SATISFIES REQMTS	• SATISFIES REQMTS • RUGGED • RELATIVELY INEXPENSIVE	• RUGGED	• SATISFIES REQMTS • RUGGED • RELATIVELY INEXPENSIVE	• SATISFIES REQMTS • SIMPLE • RUGGED	• SATISFIES REQMTS • RUGGED • MAINTAINS K IN LIQUID STATE • POSITIVE
DISADVANTAGES	• SEE NOTE 1 • SENSITIVE TO DYNAMIC LOADS	• SEE NOTE 1 • SENSITIVE TO DYNAMIC LOADS • WILL NOT SELF-PRIME IN ALL ORIENTATIONS	• LARGE POTASSIUM INVENTORY	• TEMPERATURE VARIATION AT TUBES • MAY REQUIRE DEVELOPMENT TESTING • LARGE K INVENTORY	• REQUIRES DEVELOPMENT TEST PROGRAM • DIFFICULT TO START	• REQUIRES DEVELOPMENT TESTING • MAY HAVE LOCALIZED IMPAIRED HEAT TRANSFER • GEOMETRY PROBLEMS--TUBES	• REQUIRES ACTIVE COMPONENT • REQUIRES CONTINUOUS POWER • EM PUMP MAY BE EXPENSIVE
COST	• HIGH	• HIGH	• LOW	• LOW	• LOW	• LOW	• MODERATE
CONFIDENCE RATING MANUFACTURING PERFORMANCE	 C C	 C C	 A A	 A ?	 A B	 A ?	 A A

* ALL ORIENTATIONS

**PREFERRED APPROACH

1:
- COSTLY
- SUBJECT TO CATASTROPHIC BURNOUT
- REQUIRES WICK DEVELOPMENT
- SENSITIVE TO LOCAL FLUX VARIATION
- SENSITIVE TO STARTUP TRANSIENTS

Figure 2-7. Comparison of Heat Transport System Alternatives (Reference Appendix C)

- Satisfies design requirements
- Rugged
- Inexpensive
- Manufacturable
- Good confidence in performance
- Insensitive to dynamic loads
- Does not require priming
- Negligible variation in heater tube temperature
- Minimal development testing
- Starts without difficulty
- No active components

A disadvantage of the reflux boiler is weight (25 kg) of the potassium inventory. A minor concern regarding the potassium inventory is the question of fire hazard. Much larger liquid metal inventories are used in fast nuclear reactor cooling systems where the requirement for safe systems is heightened by nuclear safety questions. Further, the spacing between concentrators is sufficient to virtually eliminate the risk of a fire spreading.

A cross section of the receiver and reflux boiler heat pipe is shown in Figure 2-8. In principle, the system is no more complex than a stove top double boiler. The heat pipe container is formed by the absorber surface, the heat pipe enclosure, and the engine hot end. The system is configured so that the heater tubes are never submerged in the potassium pool, regardless of the concentrator elevation angle.

The weight of the engine is transferred to the support cone by the heat pipe enclosure. The combined weight of the engine, the receiver, and the reflux boiler is transferred to the aperture plate mounting ring by the support cone. The heat pipe, the absorber, the support cone and the insulation retainer are a welded assembly of formed and sheet parts fabricated from AISI 316 stainless steel. Blanket insulation is held in place by the stamped aluminum clamshell cover.

Greater detail on the receiver and the reflux boiler heat pipe system is provided in Appendix A which provides a more detailed general description and

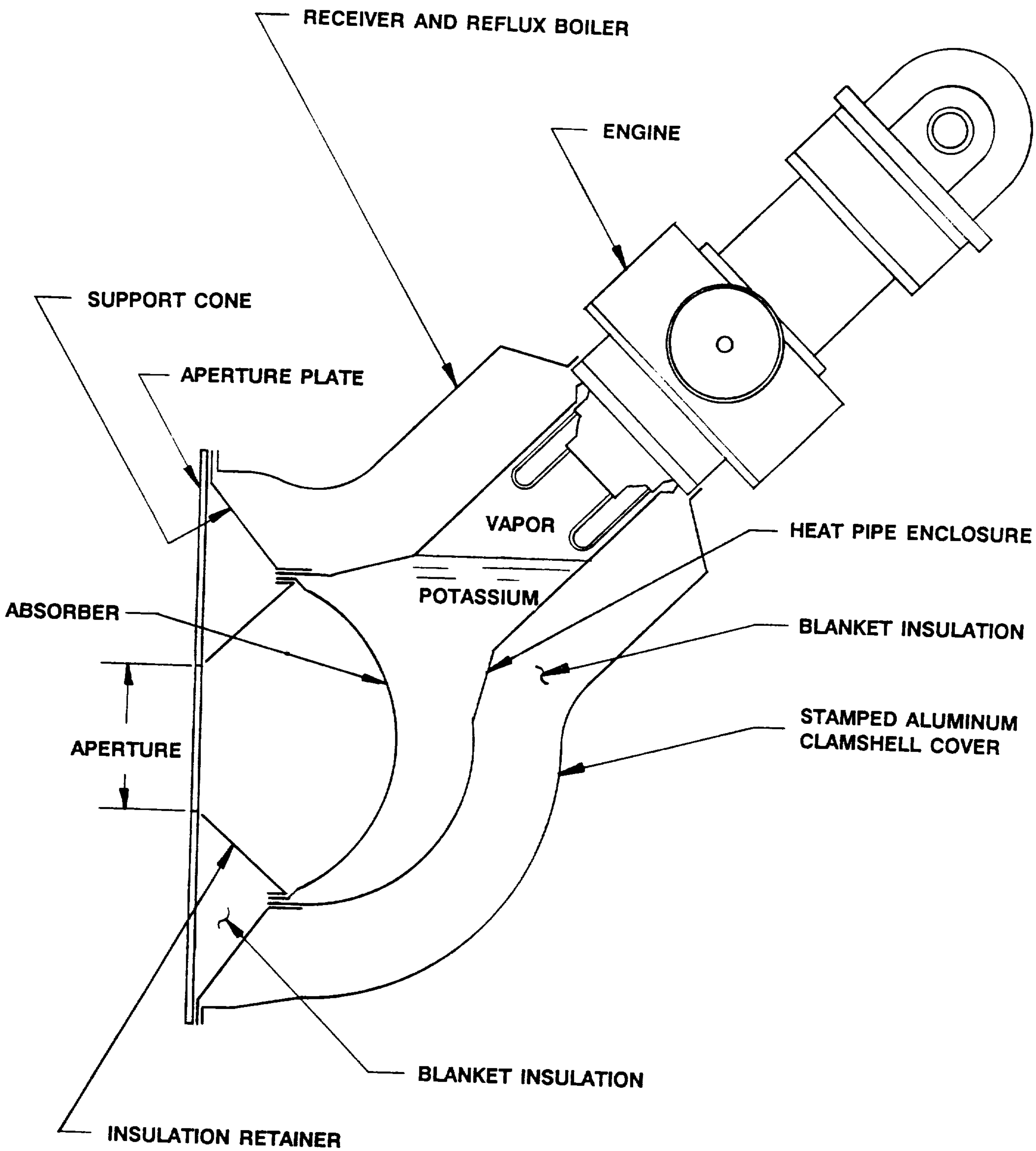


Figure 2-8. Receiver and Reflux Boiler Cross-Section

also covers the topics of liquid metal containment, joint construction, structural loading, stress analysis, working fluid selection, flooding limit, heat pipe materials and materials compatibility.

2.2.2 Stirling Hydraulic Engine

The hermetically sealed Stirling Hydraulic engine employed in the conceptual design is based on technology developed to an advanced level during the past 20 years for an artificial heart power source. Such engines and critical metal bellows components have demonstrated operating times in the desired range. This approach provides full film hydrodynamic lubrication of all sliding parts, simple construction with conventional manufacturing tolerances, proven counterbalancing, and simple but effective power control to follow insolation variations.

The principle of operation is identical to that of a free-piston Stirling engine linear alternator with the following exceptions:

- Power output is in the form of hydraulic power.
- The power pistons, the displacer rod and the stabilizer/controller are immersed in hydraulic fluid for ideal lubrication.
- Engine working gas is sealed from the hydraulic fluid (which provides lubrication and power transfer) by metal bellows, which are pressure balanced to provide virtually unlimited bellows life as proven by extensive component and system tests to 10^{10} cycles.
- A proven stabilizer/controller is incorporated into the design. This feature eliminates the stability and control problems that often complicate operation of non-stabilized free-piston Stirling engines. These problems, which have not been generally discussed in the literature or project reports, include limited and difficult control of power output, dropout of oscillations at low power levels, destructive collisions resulting from overstroking, and difficulty in debugging engines because of stability and control problems. Reference 2 provides some elaboration on the stability issue.

An illustration explaining the Stirling hydraulic engine concept is shown in Figure 2-9. Heat exchangers have been omitted from this figure for clarity, but are included in subsequent figures. The displacer employs a clearance seal. The helium working gas in the engine is separated from the hydraulic fluid by the moving cold plate assembly, which employs two bellows seals. The moving cold plate assembly acts as a diaphragm to communicate pressure-volume work between the engine gas and the hydraulic fluid. The fluid delivers net cyclic work to the opposed power pistons and to the displacer rod. The displacer rod is hydraulically coupled to the counterweight whose stroke is limited by the stabilizer, which is a very lightly loaded Scotch yoke. The stabilizer is so lightly loaded, as shown in Appendix F, that it is expected to meet the specified lifetime goals without difficulty.

The power pistons have equal masses and act in symmetrical opposition, thereby producing no vibration of the engine housing. The displacer and counterweight have equal masses and act in symmetrical opposition, so they likewise produce no vibration of the engine housing.

It is rationally demonstrable, and has been repeatedly demonstrated by STC development engineers for many operating engine designs and in many computer simulations, that addition of resistance to displacer motion will slow a stable engine in a controllable manner while simultaneously reducing heat input and power output in a manner that maintains high efficiency. Reference 4 discusses this issue in some detail. Addition of controlled resistance to the motion of the displacer is accomplished by the speed control spool valve, which is an inexpensive and simple device to manufacture and is simple to operate. The method of construction and operation of the valve is shown in the schematic diagram of Figure 2-10. A simple spring loaded spool valve with large-clearance low-cost construction is pressurized with high hydraulic pressure at the end of the spool opposite the spring. Actuation of solenoid valves adds or removes fluid to or from the spring loaded end of the spool, moving the spool and varying the flow resistance of the valve, which changes engine speed, heat input, and power output, while maintaining high engine efficiency.

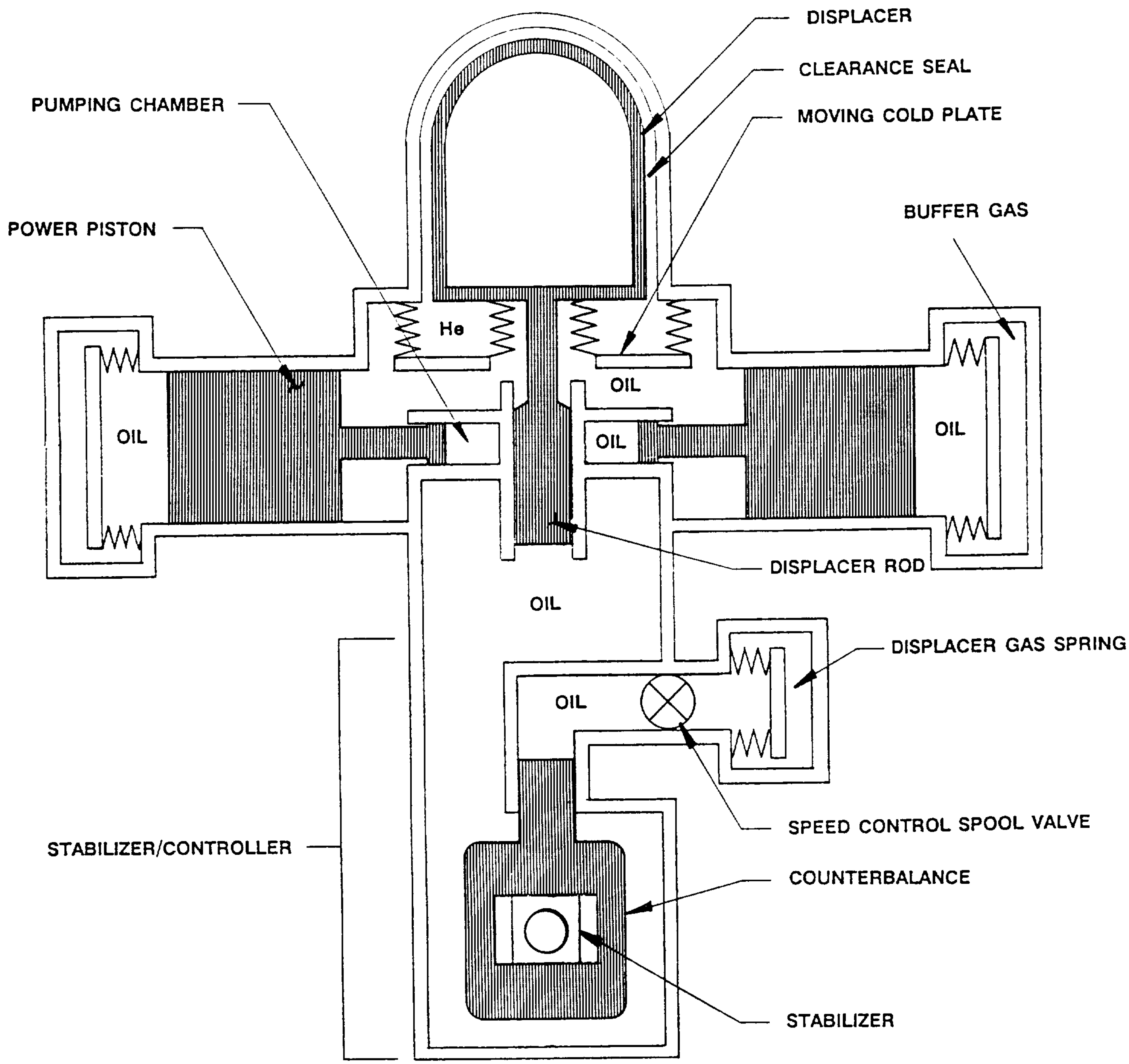


Figure 2-9. Stirling Hydraulic Engine Concept

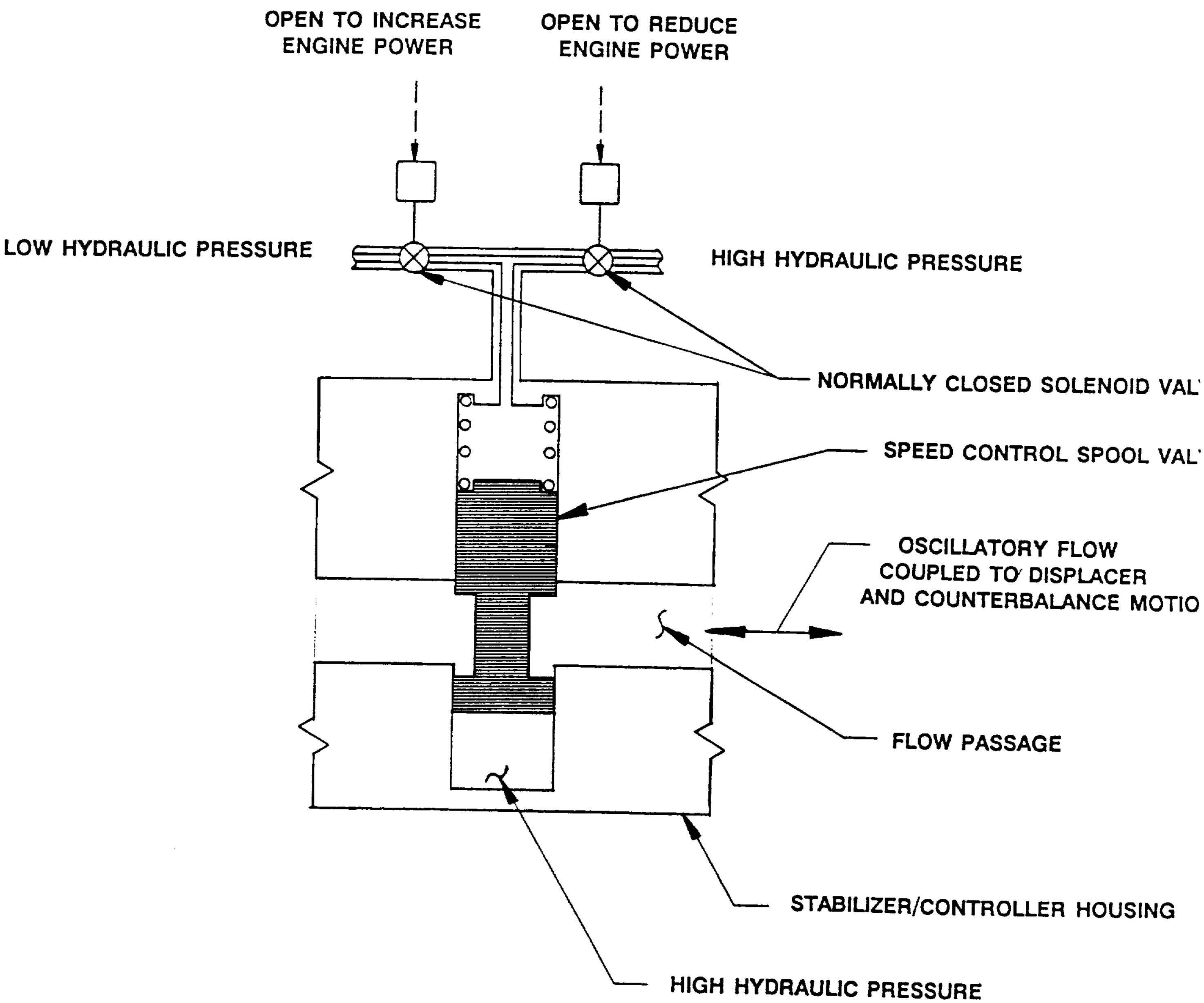


Figure 2-10. Engine Frequency Control Valve

The advantages of the engine concept presented include the following:

Stability and Control

A key advantage of the present concept is a remarkably simple engine control system which varies engine frequency, thereby modulating heat input and power output. High efficiency is provided over a very wide insolation range, taking advantage of very low and very high levels of incident solar heating. The engine power output turndown ratio exceeds 5 to 1. The controlled variable of the engine control system is the temperature of the potassium surrounding the heater head. The engine control system precisely regulates heater head temperature, which is important because candidate heater head materials lose strength dramatically above the design operating temperature. If heater head temperature is not precisely controlled, heater head failure or decreased efficiency will result.

As discussed above, instability of free-piston Stirling engines can be a serious problem. Past experience in operating small free-piston Stirling engines and in simulating free-piston Stirling engines in the target power range indicates that dropouts, damaging collisions, limited turndown ratio, and development difficulties are likely unless a stabilizing element, such as the stabilizer of the present design, is employed. A stabilizer similar to that used with the present design was used with the System 4 engine which underwent seven years of life testing. Reference 2 discusses Stirling engine stability in detail.

Despite the mechanical simplicity of free-piston Stirling engines, stability and control are significant technical problems which must be addressed to achieve a system which will provide fully automatic unattended operation in field use. That the present design fully addresses the stability and control issue is an important fact which should be given significant weight in comparison with alternative approaches.

Hermetic Sealing

The STC team has demonstrated with repeated long life engine designs that bellows designed with low stress levels will demonstrate essentially unlimited cycle life. Low stresses are achieved by pressure balancing the bellows.

Life tests of up to seven years operation of engines employing bellows seals have been demonstrated by the STC team. Hermetic bellows seals similar to those of the present conceptual design are approaching 10^{10} cycles in 27 continuing component life tests. Engine life tests and bellows component life tests are discussed in detail in Reference 3.

Demonstrated Internal Lubrication and Makeup Systems

The STC team has executed years of successful life testing of engines which require no external lubrication supply and which have internal hydraulic fluid makeup systems. These lubrication and makeup systems are conceptually identical with those incorporated into the present design. Operation of engines with completely automatic internal lubrication and makeup systems is an important consideration, which should be given significant weight in comparison of this approach with alternative approaches.

Perfect Balancing

The present conceptual design provides perfect balancing at all operating speeds for complete elimination of vibration.

Proof-of-principle of the dual opposed power pistons is provided by the Space Power Demonstrator Engine which has operated with nearly perfect balance with two matched opposed power pistons driven by the same engine pressure.

Proof-of-principle of the hydraulically coupled displacer counterbalance is provided by the System 8 artificial heart engine which operates with nearly imperceptible vibration. The use of proven counterbalancing systems which balance completely and are insensitive to line frequency variations is an important fact which should be given significant weight in the comparison of this approach with alternative approaches.

High Efficiency

In addition to the high efficiency provided by the basic engine parameters, high efficiency hydraulic components, and a high efficiency commercial rotary induction generator, the present concept enhances system efficiency by operating over a wide range of insolation levels, from very low levels up to continuous operation at the survival insolation level.

High Reliability and Long Life

The hermetic metal bellows and hydraulic lubrication features which have made possible years of continuous operation of the artificial heart power source are employed in the present design. Solid experience underlies the expectation of high reliability and long life for the present design.

Displacer Clearance Seal

Study of a displacer gas seal to prevent displacer blowby indicates that with careful control of tolerances it is possible to use the clearance between the displacer and the cylinder liner as an effective clearance seal. In addition to this approach, a self-aligning clearance seal which was investigated is a promising alternative. From these studies, it is clear that a preloaded or pressure-loaded seal will not be required, and that a true clearance seal will suffice.

Fully Automatic Unattended Operation

The engine control is so simple, and the engine so stable, that fully automatic unattended operation should be easily achieved.

Manufacturable Design

Analysis of the design shows that the engine will function with high efficiency and low wear with diametral clearances up to 1.5 mils per inch of diameter for all hydraulic piston clearance seals and an initial diametral clearance of 1 mil per inch of diameter for the stabilizer journals. Careful work has been done throughout development of the conceptual layout to establish very simple, manufacturable configurations.

Allowable seal clearances in oil lubricated systems and gas bearing systems are limited primarily by seal leakage power losses, which are inversely proportional to viscosity and are proportional to the cube of clearance. The viscosity of the oil used in the present design is 10 centipoise, which is 560 times the viscosity of helium at engine cold end conditions. For a given seal diameter, length, and power loss, clearance of an oil seal can be the cube root of 560, which is 8, times the clearance of a gas seal. It seems an inescapable conclusion that oil system clearances can be larger than gas bearing clearances, a fact related to manufacturing cost that should be given weight in the comparison of this approach with alternative approaches.

Maintainable Design; No Planned Maintenance

The engine concept presented is intended to operate for 60,000 hours with no planned maintenance. Nonetheless, the use of a Stirling hydraulic engine makes it practical and favorable in terms of initial cost to design the engine with demountable O-ring sealed interfaces between subassemblies. The hydraulic fluid employed is sufficiently viscous, and therefore resistant to leakage, that O-ring seals suffice in the place of welds for all of the assembly interfaces in the engine except the hermetic gas seals. This allows access for maintenance of most engine components, reduces the cost of engine assembly, and allows factory corrective action if production problems are discovered after engines are assembled. The maintainability of this engine is an important fact, which should be given weight in comparison of this approach with other approaches.

Compatiblity with Commercial Components

The hydraulic output engine allows the use of commercially developed hydraulic motors, hydraulic motor automatic controls, and rotary induction generators, all of which have zero development time, zero development risk, immediate availability from an established production and marketing base and proven and quantified performance and life. These are important facts which should be given weight in comparison of this approach with alternative approaches.

Protected Bellows

The bellows of the present design are protected against loss of hydraulic pressure by a makeup pump and an automatic isolation valve, both discussed in Section 2.3.1 of this report. STC considers these elements to be adequate protection against oil depressurization. In addition to these elements, it is possible to add an automatic gas depressurization system for further protection if desirable.

A conceptual engineering layout of the engine module is shown in Figure 2-11. Details provided by the layout which was not included in earlier figures include assembly and maintenance interfaces, the heat exchangers, the regenerator, and simple inexpensive startup positioning springs on the power pistons, on the displacer rod, and on the counterweight. More detail is also provided on the stabilizer and the hydraulic starter motor, which is controlled by a solenoid valve.

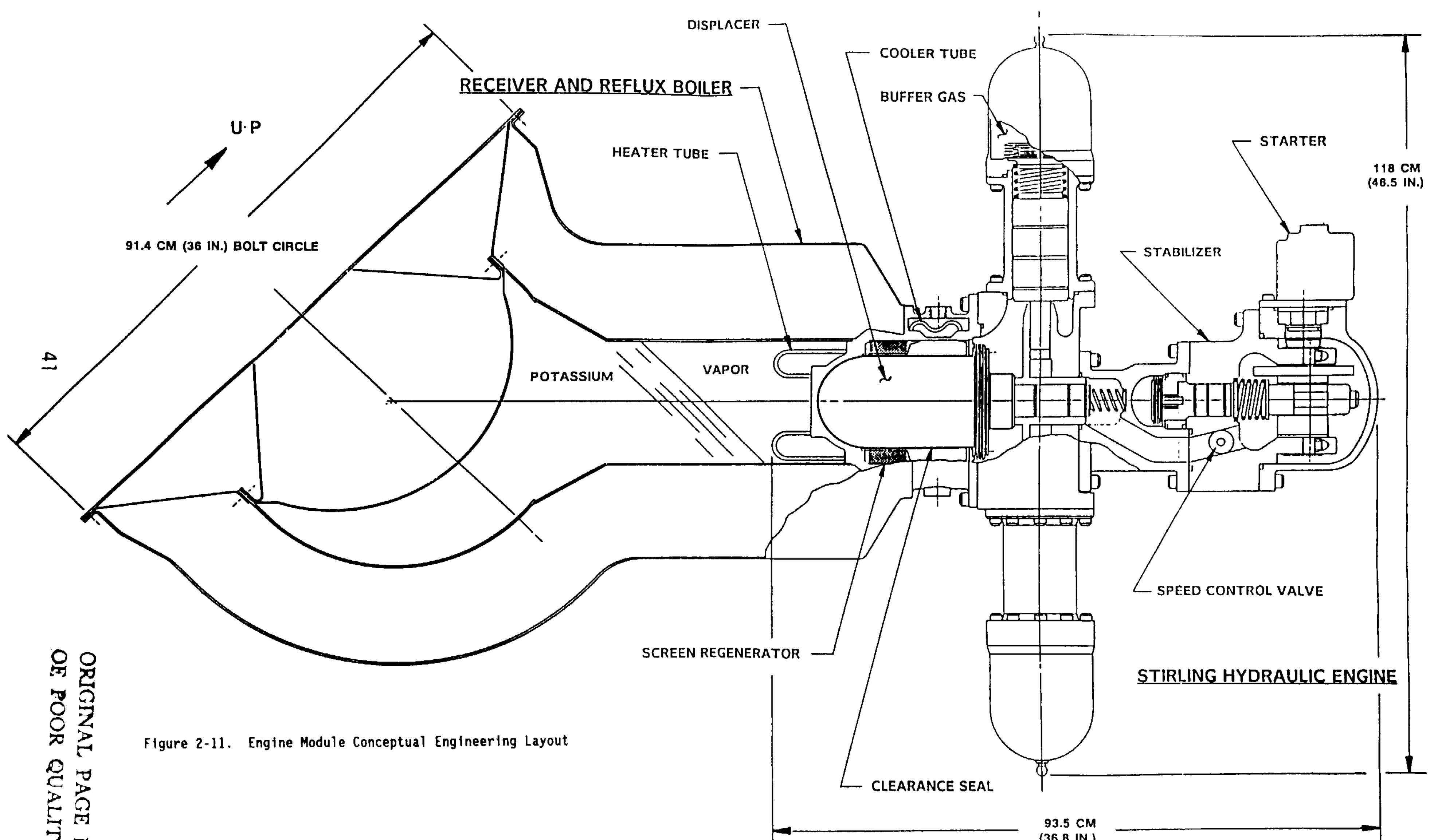


Figure 2-11. Engine Module Conceptual Engineering Layout

The heat exchangers and regenerator are discussed in detail in Section 2.2.4 on analysis. Care was taken in the design to achieve a simple U-tube heater geometry and a simple M-tube cooler geometry, with all heater tubes identical and all cooler tubes identical. Gas flow diffusers are cast into the heater head to reduce the velocity pressure of flow from the heater and cooler tubes impinging upon the regenerator matrix. The velocity pressure is reduced to a small fraction of the matrix pressure drop. This minimizes the potential for performance loss from regenerator flow irregularities. The baseline regenerator geometry for initial demonstration on the Test Bed Concentrator is 1 mil wire screens with 70 percent porosity. Foil regenerators were investigated and are favorable as a low cost advanced technology approach which is particularly insusceptible to flow irregularities and which packages well in the low cost annular regenerator arrangement of the present engine design.

Design details on the conceptual engine design are presented in Section 2.2.5.

2.2.3 Materials Evaluations

Early in the conceptual design a target temperature limit for the heat transport system was established at 800°C, primarily on the basis of objectives established by the RFP. Stress analysis of heater tubes and the heater head at 800°C indicated that the wall thicknesses would be larger than desirable, and that thermal stresses associated with thick walls might be a problem.

Information provided by G. D. Johnson, Manager of Materials Engineering at Westinghouse Hanford Company, also indicated that materials compatibility problems with liquid metals could be formidable at 800°C. The concern for materials compatibility with liquid metals was reinforced through discussions with materials experts on the staff of the NASA Lewis Research Center. On the basis of materials compatibility concerns, the heat transport system design temperature was reduced to 700°C. At 700°C, wall thicknesses become reasonable and liquid metals compatibility concerns, while not trivial, come into a range with a good experience base.

The high temperature alloys of choice are AISI 316 for the high temperature receiver and heat transport system components, CG-27 for heater tubes, and XF-818 for the heater head casting. The creep rupture and fatigue safety factors for engine components using these materials meet or exceed the value of 1.5 specified by NASA for design criteria. Potassium compatibility for a 60,000-hour life for these materials at 700°C (and probably any other materials) will require testing, the more prototypic the better. Successful tests with sodium for other applications offer a good likelihood of success.

Exterior surfaces of the heat transport system will be coated with nickel aluminide which is easily applied and becomes an Al_2O_3 oxidation barrier upon heating.

More detailed information on materials evaluations is provided in Appendices A and D.

2.2.4 Computer Simulation and Other Analyses

This section discusses computer simulation methods for analyzing engine dynamics and thermodynamics and computer analyses to establish basic engine design parameters. Also discussed briefly are other supporting analyses performed to verify that the performance of the engine will be high and to quantify overall system performance.

Computer Simulation of Engine Performance

This is a complex subject because the development of an engine design involves both synthesis and analysis, generally interwoven. An engine cannot truly be analyzed until it is specified, but conversely, an engine cannot be specified without analysis. This is therefore a trial and error process which requires much experience and has to factor in the realities of mechanical engineering analysis and of mechanical engineering design.

The computer codes used in synthesis and analysis of the basic engine parameters are SCALE and MCP, which are STC codes operated by STC personnel, and GLIMPS and SCALING, which are Gedeon Associates codes operated by David Gedeon.

The STC code SCALE is a simple but powerful algebraic computer code based upon a linearized isothermal model of a free-piston Stirling engine. Given trial values of important parameters and dimensionless ratios which are relatively invariant for free-piston Stirling engines, SCALE can generate a family of trial designs in the power, pressure, volume, and frequency range of interest. These designs can be checked for reality with hardware designers and an appropriate design selected for dynamic simulation. In the process of dynamic simulation, parameters are adjusted to arrive at a design which operates stably, controllably, and without collisions. This design can then be used as a dynamically correct input base case to SCALE to produce a large family of dynamically similar and dynamically correct designs at the power level of interest, exploring tradeoffs in pressure, volume, frequency, and the ratio of stroke to diameter.

In addition to the very fundamental information just mentioned, the list of candidate engine designs produced by SCALE includes parameters of major interest to both the design analyst and the mechanical design engineer. These include displacer and piston weights and dimensions, hydraulic flow inefficiencies, and stabilizer loads. Review of the output of SCALE leads to selection of a reference engine design which is stable and controllable, has low hydraulic flow losses, and has desirable mechanical dimensions. The ability of SCALE to produce dynamically correct designs has been checked by dynamically simulating designs produced by SCALE and verifying that the dynamic simulation output parameters match the input parameters.

The STC code MCP is an isothermal dynamic simulation method which has been used to design a number of hydraulic Stirling engines and analyze many more. Dynamic operating parameters of engines designed by MCP and tested in the laboratory agree well with the MCP predictions.

The Gedeon code GLIMPS is a remarkable fast running Stirling engine nodal thermodynamic analysis code that uses a personal computer to do Stirling engine simulations that would normally require a considerable amount of time on a mainframe computer. GLIMPS is a design point analysis code as opposed to an optimization code. Geometry, source and sink temperature, pressure, frequency, and sinusoidal piston motions were input from SCALE and MCP to GLIMPS. Engine thermodynamic performance was calculated. Parasitic heat loss and parasitic power output losses were calculated independently by STC and used to modify the GLIMPS results. More detail on GLIMPS is provided in Appendix E.

The Gedeon code SCALING allows the basic cycle power and the heat exchanger performance of a successful GLIMPS run to be constrained while changing some of the dimensions of the machine. SCALING was used in conjunction with GLIMPS to optimize the heat exchangers of the present design.

The final results of the engine synthesis and analysis work performed using SCALE, MCP, GLIMPS, and SCALING are well summarized by the subcontractor report prepared by David Gedeon, which is included as Appendix E. Referring now to Tables 3 and 4 of the appendix, Case 7.4 is designated the baseline design, with a 1 mil wire screen regenerator of 70 percent porosity. Case 7.1 is a foil regenerator engine which is comparable in performance to the baseline design.

A number of areas of concern in which studies by STC and Gedeon led to satisfactory conclusions are discussed at some length in the Gedeon report. These include:

- Validation of the GLIMPS code
- Selection of the heater concept
- Selection of the porous regenerator for the baseline design
- Selection of the volume allocation for heat exchangers
- Minimization of regenerator flow distribution problems
- Selection of operating pressure and frequency
- Optimization of the heater, regenerator, and cooler

Other Analyses

General analyses performed in support of the design effort include those listed below. The results of these analyses are reflected in the hardware designs presented in this report and submitted to Pioneer for cost evaluation, and in the engine and system performance projections made in this report. STC is satisfied with computational methods used to perform these analyses and believes the analyses provide sufficiently accurate estimates of performance for the purposes of the present design activity. In particular STC has done substantial theoretical and experimental investigation into flow losses associated with Stirling hydraulic engines and found the methods to be reasonably accurate.

GENERAL DESIGN ANALYSES

- Analysis of engine parasitic heat leaks
- Analysis of engine parasitic mechanical losses
- Analysis of stabilizer/controller mechanical losses
- Displacer clearance seal analysis
- Displacer gas spring sizing
- Buffer sizing
- Efficiency effect of heater tube-to-tube temperature differences
- Engine design power level requirement
- Analysis of power turndown capability
- Heat rejection system analysis
- Stabilizer bearing pressures
- System integrated annual energy production
- Bellows life analyses
- Stress analyses, including creep, fatigue and buckling
- Heat pipe analyses
- Weight analysis

Appendices A, B, and C provide details on analysis of heat pipe and other heat transport options. Stabilizer bearing pressure analysis results are presented in Appendix F. Heater tube stress analysis is presented in Appendix G.

2.2.5 Engine Design Details

A substantial amount of careful layout work and mechanical design analysis underlies the final concept design presented in this report. A conceptual layout of the heat transport system is included on the second page of Appendix A. A detailed conceptual layout of the 30 kW Stirling engine and selected detailed drawings prepared for cost evaluation purposes are provided in Appendix F. Also included in Appendix F is an automatic internal leakage makeup diagram, a diagram showing the very simple and easily produced components of the stabilizer, a table showing that the stabilizer bearing pressures are very light compared to automotive practice, and a table of engine and heat transport system suspended weights.

2.3 COMMERCIAL COMPONENTS, SYSTEM INTEGRATION AND CONTROLS

An important advantage of the selected approach is the use of commercial power generation components with zero development cost and well characterized performance, life and operating characteristics. This section of the report discusses the integration of the Stirling hydraulic engine and the commercial power generation components into an efficient, reliable, unattended, automatic power generation system.

2.3.1 System Schematic

A schematic for the integrated system is presented in Figure 2-12. The stand-alone system is shown for ease of representation and for simplicity of discussion. The engine module, Item 1, includes the receiver, the heat transport system, the Stirling hydraulic engine, and the engine speed control system, which regulates heater head temperature. The engine pump hydraulic fluid at a flow rate roughly proportional to engine heat input. The coolant radiator, Item 2, is similar to an automotive unit and includes a coolant pump and a fan. Two pulsation suppressors, Item 3, include one at the engine intake, and one at the engine discharge. The pulsation suppressors are small and inexpensive oil/gas accumulators with the oil separated from the gas by a simple rubber bladder. They produce steady flow in the external hydraulic circuit. Five micron filters, Item 4, are immediately upstream of the engine module and the hydraulic motor to keep the engine and motor clean, assuring low wear rates for very long life. Item 5 is a small priming and makeup pump with a discharge check valve. The pump is turned on and off by a diaphragm-type pressure switch. The automatic isolation valve, Item 6, closes to protect the engine bellows if the engine discharge hydraulic pressure drops to the 2600 psi engine charge pressure. A relief valve, Item 7, accepts engine flow if the engine is running when the generator is off line. The hydraulic motor, Item 8, includes a factory optional back pressure control unit, which adjusts hydraulic motor displacement to accept the flow rate of hydraulic fluid produced by the engine over the engine's entire power range. Motor torque is proportional to flow rate. The induction generator, Item 9, operates at line voltage and near synchronous frequency. Generator current is proportional to motor torque.

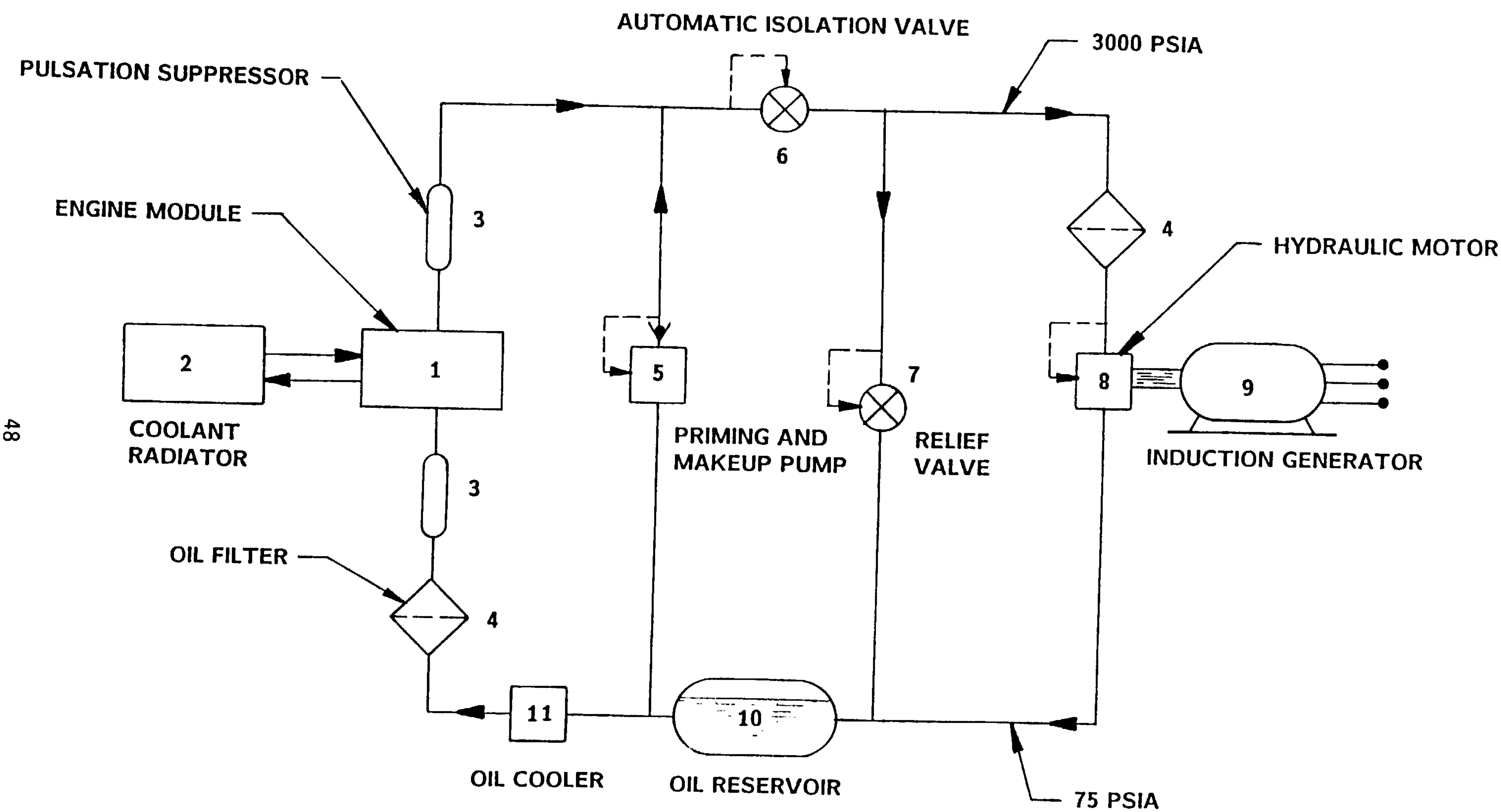


Figure 2-12. System Schematic

The oil reservoir, Item 10, is a simple, low-pressure tank containing hydraulic fluid pressurized by a volume of dry gas above the fluid. The oil cooler, Item 11, includes a fan to force air past the cooler.

The dashed lines in the system schematic represent pressure feedback for automatic unattended control of the priming and makeup pump, the relief valve, and the hydraulic motor.

For the array configuration, 20 concentrator-mounted engines are hydraulically connected in parallel to six sets of motors and generators. The method of connecting and controlling the array has been discussed in some detail in Section 2.1.1. In particular, it is useful to review the block diagram of the array shown in Figure 2-4. More details on the array hardware are included in an engineering schematic of the hydraulic circuit in Appendix F.

2.3.2 Control for Automatic Unattended Operation

Control concepts for the power generation system were discussed in Section 2.1.1 and were reviewed in the above System Schematic discussion. This subsection discusses sensors, control circuits, and final control elements for the power generation system and overall concentrator and generation system control for automatic unattended operation.

Engine Speed Control

The engine speed control, which simultaneously varies engine heat input and flow thoughput, operates to maintain constant temperature in the heat transport system. The sensing element will be either a thermocouple or a gas bulb thermometer, selected on the basis of life and reliability. The intermediate circuit will be a simple electronic circuit and two solenoid valves, as shown in Figure 2-9. Current flows in only the first solenoid if the potassium temperature is too low, in only the second solenoid if the potassium temperature is too high, and in neither if the potassium temperature is acceptable. The final control element for engine speed is the spool valve shown in the figure. The engine speed controller will be attached directly to the engine module for easy access to the sensing and final control elements.

Engine Starter Control

The engine starter is a small hydraulic motor powered by hydraulic fluid flowing through the motor from the high pressure pulsation suppressor to the low pressure pulsation suppressor. The starter engages through an extremely simple long life clutch which is operated by the same pressure which powers the starter. A logic circuit in the engine power control module momentarily engages the starter by opening a solenoid valve when the potassium temperature sensed by the engine speed control system enters the temperature control band during heat up.

Hydraulic Motor Control

The hydraulic motor control adjusts displacement of the constant frequency motors to increase or decrease fluid consumption thereby regulating hydraulic motor inlet pressure. This also establishes and regulates the engine outlet pressure. The process of regulating hydraulic inlet motor pressure also matches the hydraulic motor flow rate to the engine flow rate.

For the stand-alone system a factory control option provided with the hydraulic motor causes the motor to automatically regulate its own inlet pressure.

Control of the hydraulic motors in the array concept utilizes a motor-generator array controller discussed in Section 2.1.1.

Switchgear Control

The switchgear should be closed any time the pressure to the hydraulic motors is above the lower limit of the pressure control band of the hydraulic motor control system and opened any time the supplied pressure is below the lower limit. A limit switch on the controller of the variable displacement hydraulic motor will provide the information needed for switchgear control.

Concentrator Control

A possible control algorithm for the concentrator controller is to track-on when the sensed insolation exceeds a preset limit, and track-off when the sensed insolation falls below a preset limit. In addition, the concentrator should track-off if the temperature of the heat source exceeds a preset limit. The signal for track-off on overtemperature will be transmitted by wire from the engine speed controller to the concentrator controller.

2.3.3 Engine Cooling System

The engine cooling system parameters were selected to provide a 37°C nominal heat rejection temperature difference between the metal-to-gas interface of the Stirling cycle cooler and the ambient air temperature. The ambient air temperature was assumed to vary from -7°C to 33°C as specified in the request for proposal. The nominal ambient air temperature was taken to be 13°C. Adding the rejection temperature difference to the ambient temperature range gives an engine cold metal temperature ranging from 30°C to 70°C. A 50°C nominal cold metal temperature was used in engine performance simulations. The system performance given in this report takes into account reductions in electrical output of 1.85 kW per engine for cooling fan motor power and 70 watts for cooling pump motor power. The final heat rejection is through a commercial fan cooler similar to that depicted in Figure 2-13. The coolant is a 50/50 water glycol mixture.

2.3.4 Hydraulic Motors

The hydraulic motors selected for the baseline array design are the Volvo V11-250 variable displacement motor and the Volvo F11-150 fixed displacement motor. A fixed displacement motor is depicted in Figure 2-14. In the array, five fixed displacement motors and one variable displacement motor utilize the hydraulic power from 20 engines. The hydraulic motors are designed to operate at up to 5000 psig inlet pressure and up to 2500 rpm. The present design, at 3000 psig and 1800 rpm, provides a long operating life between bearing replacements. On the average, the hydraulic motors will require less than three replacements of main bearings in the 60,000 hour life of the plant. The labor required to exchange a hydraulic motor and replace its bearings is estimated to be 4 man-hours.

2.3.5 Commercial Induction Generators

The rotary induction generator is clearly the preferred machine for connection to a utility grid with established frequency and voltage. Induction generators provide the following advantages.

1. The induction generator has a cost advantage over the synchronous generator in sizes up to several megawatts. In the size range of present interest, the cost of an induction machine is about half that of a synchronous machine.

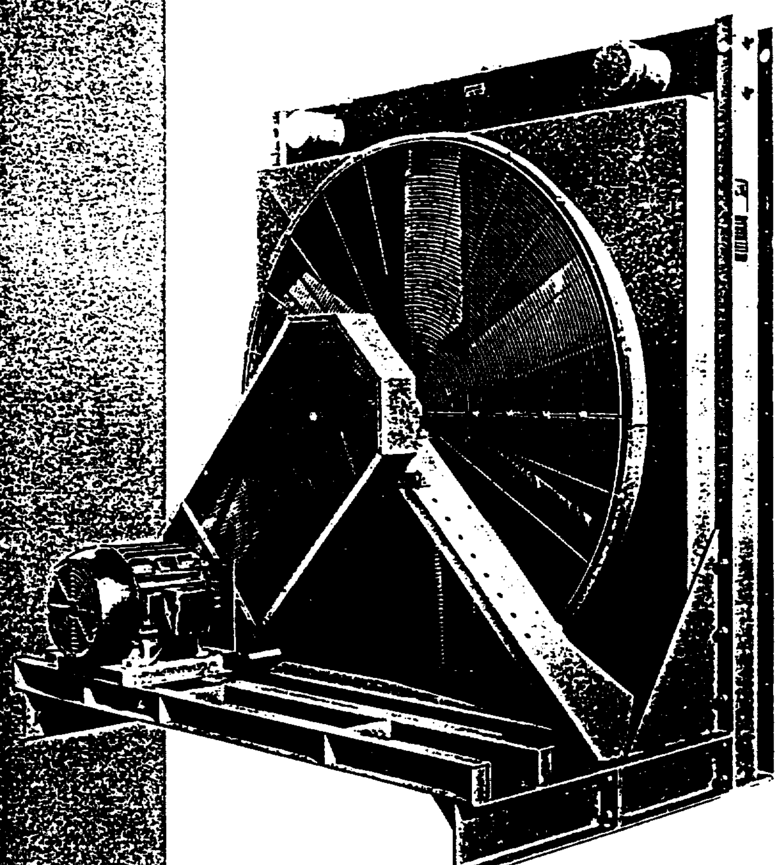
CATALOG 15 1986
Supersedes Catalog 15 1983

ENGINE COOLER

Young
RADIATOR COMPANY

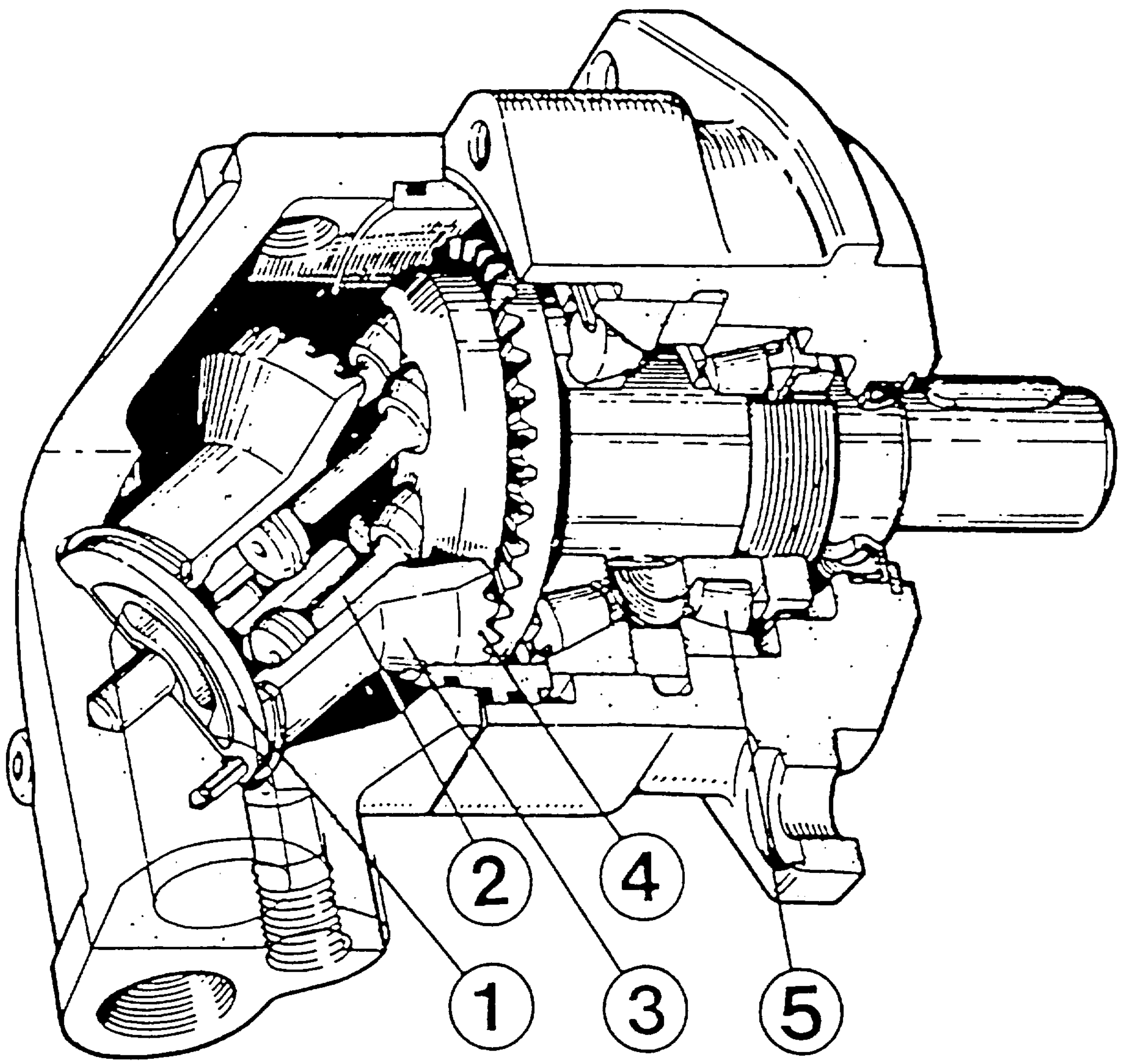
STANDARD MWC

WATER COOLING RADIATORS with OSHA guards



- 95 models in 10 frame sizes
- available from stock
- remote operation or external drive
- heavy duty industrial construction (100 mph 45 *m/s* wind)
- -Q quiet option
- "two circuit" variation

Figure 2-13. Fan Cooler



1 PORT PLATE
2 SPHERICAL PISTON
3 CYLINDER BARREL
4 GEARWHEEL SYNCHRONIZATION
5 TAPER ROLLER BEARING

Figure 2-14. Fixed Displacement Hydraulic Motor

2. Operation of an induction generator supplying power to a grid is the essence of simplicity. With an induction generator, the circuit breaker can be opened or closed at will at any time without concern for synchronization. This is not true for a synchronous generator operating into a grid. If an induction generator is below synchronous speed, it acts as a motor and draws power from the grid. At synchronous frequency no power flows. Above synchronous frequency power flows to the grid. At the 1,800 rpm synchronous frequency of the present design, the required slip for rated current is about 20 rpm. Thus, the hydraulic motor and the generator will operate at 1,820 rpm.

3. In the power range of interest the development, production, market base, availability, and price competition for rotary induction generators is incomparably larger that for competing equipment. For most intents and purposes, an induction motor is also an induction generator. There is little or no hardware difference between an induction motor and an induction generator. Therefore, for most purposes, a standard induction motor with specified parameters can be sold and put to work as an induction generator with the same specified parameters. This greatly broadens the commercial base.

4. In addition to its other advantages, the induction generator is efficient, simple (brushless), and rugged.

5. Polyphase rotating generators in general produce essentially zero harmonic distortion.

6. Commercial rotary induction generators in the size range of interest are three phase machines, and therefore are very compatible with a conventional three-phase grid.

7. The generators for the present design have an efficiency of 96% over a wide power generating range.

Generators are typically designed to provide eight to ten years of continuous operation at rated temperatures. Typical operation is below rated temperature, so the lifetime is typically greater than eight to ten years and thus well beyond the 60,000 hour design life specified for the reference design.

Drip-proof construction is considerably less expensive and slightly more efficient than totally enclosed fan-cooled construction. Drip-proof construction is routinely used in the open in severe environments such as oil fields and demonstrates no problems from blowing sand and driving rain. Flooding cannot be tolerated. Screens are sometimes provided in air flow passages to exclude insects and rodents. Drip-proof construction is expected to meet the requirements of the proposed application.

2.3.6 Generator Control and Power Factor Connection

There is no control required for the generator apparatus or the power output circuit with induction generators operating into a grid. Control is achieved through the automatic variation of hydraulic motor displacement, which is proportional to hydraulic fluid flow, hydraulic motor torque, and output current. Frequency and voltage are established by the grid.

The generator power factor for the array configuration is around 90% at high power, dropping to 85% at the full turndown condition. Therefore power factor correction is not required for the array configuration. For the stand-alone configuration, 94% of the energy generated is at a power factor above 0.85. The approach to be taken regarding the 6% of the available energy generation which is at a power factor below 0.85 will be resolved in the next phase. Primary alternatives include the following:

1. Addition of a small amount of capacitance to raise the power factor above 85% at all times,
2. A reduction in power revenue rates for the small amount of power generated at a power factor below 85%, or
3. A decision not to generate the 6% of the energy which would be at a power factor below 85%.

Section 3

RELIABILITY AND MAINTENANCE

All the components specified in the conceptual design have 60,000 hours of maintenance-free life expectancy with the exception of the hydraulic motor. On the average, the motor will require taper bearing replacement less than three times in the 60,000 hour life of the power generation system. Taper bearing replacement will require four man-hours per replacement including the time to exchange hydraulic motors and the time to replace the bearings.

The engine is designed for zero maintenance, but most engine parts are easily replaceable. A number of small engine configurations which are similar to the proposed concept have been developed and employed as artificial heart power sources. Two of these have been extensively life tested with outstanding results. Only one sample of each engine was tested, so the results are all the more impressive.

The System 4 engine operated unattended for 35,900 hours in one continuous run until retirement from service because of damage from a machining operation to remove a burned out electric heater.

The System 6 engine ran 22,700 hours in one continuous run finally interrupted by wear of a magnesium check valve poppet. In retrospect, the valve material was a questionable choice. On disassembly, all components except the valve were in excellent condition. The valve has been resurfaced and the engine reassembled for continuation of the tests. System 6 is expected to demonstrate many thousands of hours of additional life.

In assessing the remarkably low number of failures per hour in the tests cited above, it is important to remember that these are development tests intended to find the weakness in these designs. After development testing and related design improvements, much lower rates of failure can be expected.

As discussed in Section 2.3.5, rotary electric induction generators have maintenance-free lives on the order of 90,000 hours.

On the subject of reliability, it is important to note the high level of satisfaction commonly expressed by users of hydraulic equipment. Appendix H is a technical article entitled, "Plant Uses 53 Hydraulic Motors--No Motor Downtime in 2 1/2 Years."

Section 4

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1. M. A. White, "Conceptual Design and Cost Analysis of Hydraulic Output Unit for 15 kW Free-Piston Stirling Engine," Report No. DOE/NASA/0212-1, August 1982.

2. Peter Riggle, "Stability Characteristics of Stirling Engines," Proceedings of the 20th Intersociety Energy Conversion Engineering Conference, SAE, Miami Beach, Florida, August 1985, pp. 3.313-3.319.

3. Maurice A. White, S. Grant Emigh and Peter Riggle, "Practical Bellows Seals for Stirling Engines", Proceedings from the Society of Automotive Engineers, Inc., International Congress and Exposition, Detroit, Michigan, February 1987, SAE Technical Paper Series 870104.

4. P. Riggle, M. A. White, N. Lo, and J. E. Noble, "Stirling Engine Analysis Insights Derived from Sixteen Years of Artificial Heart Power Source Development", Proceedings of the 18th Intersociety Energy Conversion Engineering Conference, SAE, Orlando, Florida, August 1983, pp. 766-772.

Appendix A

RECEIVER AND REFLUX BOILER HEAT TRANSPORT SYSTEM - SELECTED SYSTEM

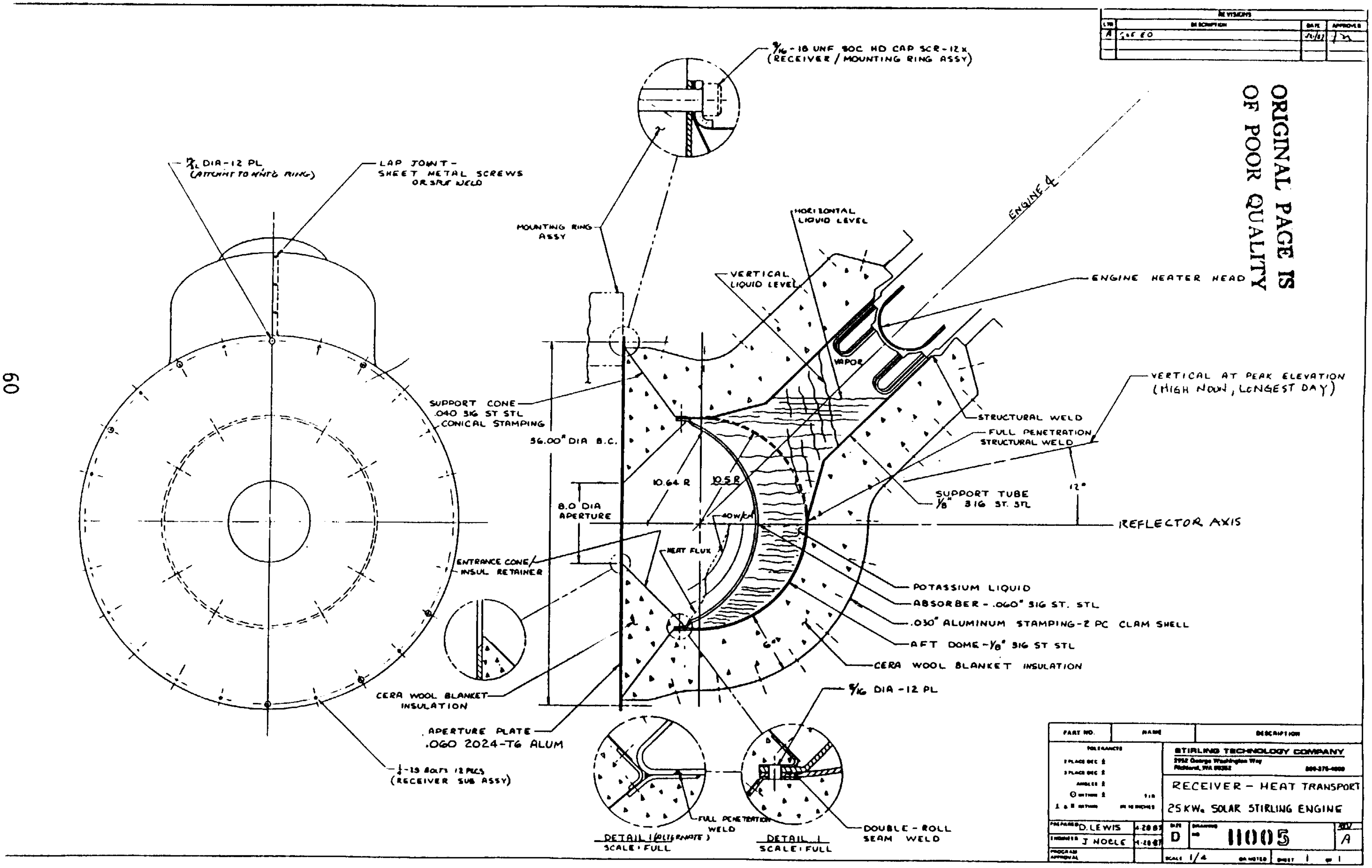
The system is shown in Figure A-1. Solar energy impinges upon the 10.6-inch radius spherical absorber which conducts heat to the potassium pool contained by the absorber and the 10.5-inch spherical radius aft closure. Nucleate boiling of the potassium at the back side of the absorber surface converts the incident energy which is delivered to the heater tubes by condensation of the vapor. The engine is supported by the support tube which is welded to the aft closure. The engine is welded to the support tube at the heater head providing a hermetically sealed system. Evacuation and fill ports (not shown) are provided. The assembly, weighing approximately 706 pounds, is supported by the support cone which attaches directly to the mounting ring at the 36-inch diameter bolt circle with twelve 9/16 diameter bolts. The aperture plate is sandwiched between the support cone and the mounting ring with its major support by the twelve 9/16 bolts.

The absorber surface is a 20-inch diameter 140° spherical segment with a peak solar heat flux of 46 watts/cm^2 (see Figure A-2), chosen because of its favorable geometric compatibility and an edge slope sufficient to preclude blanketing the lower extremity of the boiler surface with vapor. Lower heat fluxes for the same heat load can be realized with increasing the included angle and deeper cavities, but it is believed that this is a satisfactory flux and the choice an acceptable compromise. Further evaluation would likely result in a more optimum design. In Table A-1, the receiver energy losses are summarized.

The complete assembly is surrounded by cerawool insulation (Manville). The insulation is contained by a clamshell aluminum housing where the clamshells are joined by either spot welds or sheet metal screws. The potassium inventory is 54 lbs and the spacing between the spherical segments provides sufficient area for vapor flow and liquid return. Differential thermal expansion is accommodated by a slip joint between the aluminum insulation housing and the engine cylinder.

<u>Liquid Metal Containment</u>

The liquid metal container is made up of the absorber, the aft closure, and the support tube, all of 316 stainless steel. The absorber and aft dome are


REVISIONS
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APPROVED
A
$\frac{7}{32}$ DIA - 12 PL
(ATTACHMENT TO MNTG RING)
LAP JOINT -
SHEET METAL SCREWS
OR SPOT WELD
$\frac{5}{16}$ - 18 UNF SOC HD CAP SCR - 12X
(RECEIVER / MOUNTING RING ASSY)
MOUNTING RING
ASSY
HORIZONTAL
LIQUID LEVEL
VERTICAL
LIQUID LEVEL
ENGINE ℄
ENGINE HEATER HEAD
VAPOR
VERTICAL AT PEAK ELEVATION
(HIGH NOON, LONGEST DAY)
STRUCTURAL WELD
FULL PENETRATION
STRUCTURAL WELD
SUPPORT CONE
.040 316 ST STL
CONICAL STAMPING
56.00" DIA B.C.
10.64 R
10.5 R
12°
SUPPORT TUBE
$\frac{1}{8}$" 316 ST. STL
8.0 DIA
APERTURE
REFLECTOR AXIS
40 W/cm²
HEAT FLUX
ENTRANCE CONE/
INSUL RETAINER
POTASSIUM LIQUID
ABSORBER - .060" 316 ST. STL
.030" ALUMINUM STAMPING - 2 PC CLAM SHELL
AFT DOME - $\frac{1}{8}$" 316 ST STL
CERA WOOL BLANKET INSULATION
6"
CERA WOOL BLANKET
INSULATION
$\frac{5}{16}$ DIA - 12 PL
APERTURE PLATE
.060 2024-T6 ALUM
$\frac{1}{4}$ - 15 BOLTS 12 PLCS
(RECEIVER SUB ASSY)
FULL PENETRATION
WELD
DETAIL I (ALTERNATE)
SCALE: FULL
DETAIL I
SCALE: FULL
DOUBLE - ROLL
SEAM WELD
PART NO.
NAME
DESCRIPTION
TOLERANCES
STIRLING TECHNOLOGY COMPANY
RECEIVER - HEAT TRANSPORT
25 KWe SOLAR STIRLING ENGINE
D. LEWIS
J NOELE
D
11005
A
SCALE 1/4
SHEET 1 OF 1

Figure A-1.

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OF POOR QUALITY

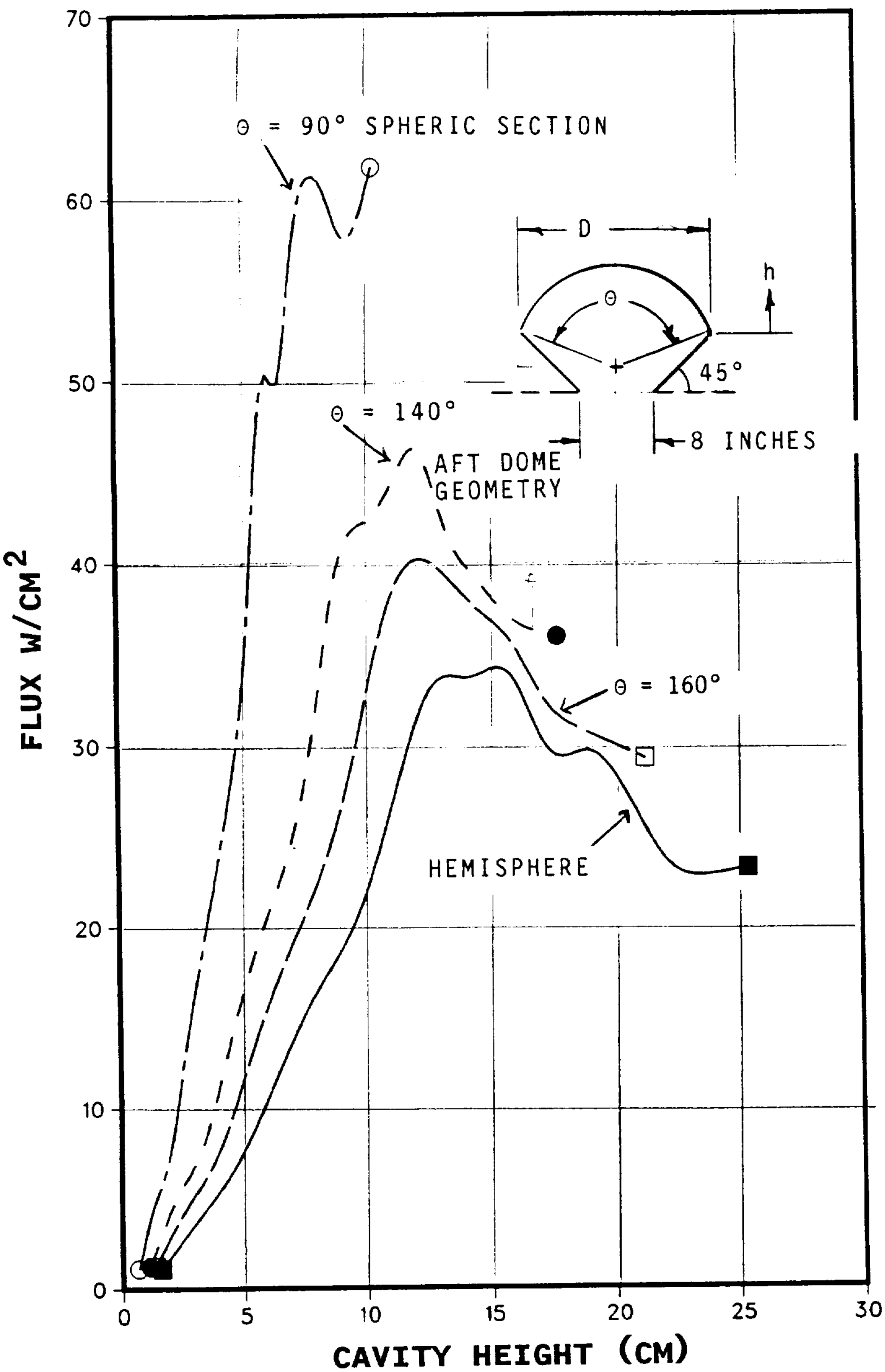


Figure A-2. Solar Flux Profile for 20 Inches Spheric Sections

Table A-1

RECEIVER ENERGY LOSS MECHANISMS

	% OF THERMAL INPUT 20-INCH DIAMETER 140 DEGREE SPHERICAL SECTOR
SHELL INSULATION	1.0
CAVITY RERADIATION	2.6
DISH SHADING	0.7
TRANSIENT STARTUP	2.0
CAVITY REFLECTION	2.3
CAVITY CONVECTION	1.0
NET RECEIVER EFFICIENCY (HORIZONTAL)	90.4

T_{∞} = 25 C

WIND SPEED = 6 MPH

TOTAL SOLAR ENERGY INPUT ON 950 W/M^2 DAY = 2.1×10^6 KJ

6 INCH INSULATION ON EACH DESIGN

REFLUX VAPOR TEMPERATURE = 730 C

formed spherical sectors joined by a double roll seam weld as shown in the inset detail. The major advantage of this type construction is its relative low cost. An alternate approach to joining is identified in the following section.

The support tube has a formed conical section and is welded to the aft closure. This intersection is circular which enhances ease of manufacturing.

Joint Construction and Welds

Two approaches to joining the absorber surface to the aft dome as shown in Figure A-3 are considered. The double roll seam weld could be made in two steps to minimize oxidation in the regions which will be in contact with potassium. The outside weld would be made first to form a seal. The joint can then be cleaned and baked out under vacuum as required, and a cover gas incorporated during the final inside seam weld. Some development and testing would be required to qualify this construction.

Because there is yet concern regarding the ability to sufficiently clean the region immediately adjacent to the inner seam weld and in the crevice, an alternate method of construction is shown. It is believed this joint will be more costly. A weld qualification program will be necessary and a method of inspection devised.

Welding of the support tube to the aft closure is by a full penetration fusion weld. As the intersection of the cone and the spherical segment is circular, it is possible to perform a second weld pass on the inner surface to assure integrity of the weld.

Welding of the engine cylinder head to the support tube will also be by fusion weld with care taken in the design to assure acceptable internal conditions after weld. Because with execution of this weld all significant sized openings will be closed, visual inspection is not possible and x-ray or ultrasonic inspection will be necessary.

Container Structural Loading

Stress levels in the container are in general minimal with sizing set by manufacturing and handling, and corrosion and material transfer. In the case

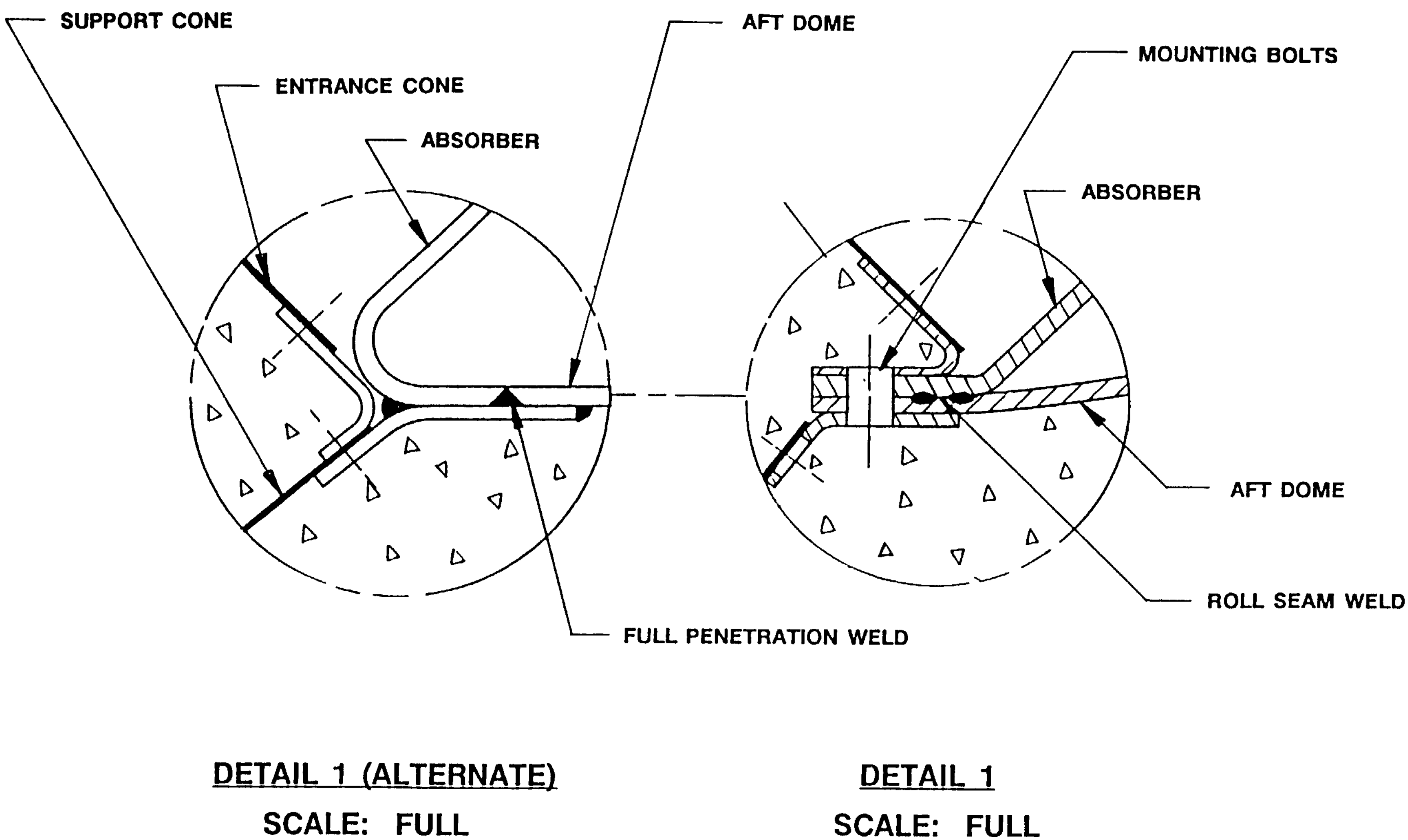


Figure A-3. Joint Configurations

of the absorber, thermal diffusivity is a consideration.

Loads are induced by differential pressure during loading and operation, and by gravity and inertial forces imposed by the engine and the receiver assembly itself. During loading and shutdown conditions, a vacuum exists in the container with atmospheric pressure external to it. During operation, potassium vapor pressure at 6.6 to 8.1 psi is the internal pressure with atmospheric pressure external. The absorber experiences cyclic thermal stress, a portion of which will relax with time.

For the absorber the nominal membrane stress is given by

$$\sigma = \frac{Pr}{2t}$$

where

P = pressure = 14.7 - 6.6 = 8.1 psi (max)

r = radius of curvature = 10.6 in.

t = material thickness = 0.06 in.

giving in operation

$$\sigma = 716 \text{ psi}$$

First cycle thermal stress is, for the nominal heat load

$$\sigma_T = 1/2\ \alpha\ \Delta T\ E$$

where

$$\Delta T = \frac{\phi t}{k} = \text{temperature differential}$$

ϕ = heat flux - 46 watts/cm^2

t = material thickness = 0.06 in.

α = coefficient of linear expansion = 10.6×10^{-6} in./in. °F

E = modulus of elasticity 20×10^6 psi

k = thermal conductivity = 1.03 BTU/hr-in. °F

which gives

$$\sigma_T = 6260 \text{ psi}$$

In reality, this is the thermal stress that the absorber would experience if it were completely restrained. As this is not the case, it is a qualitative measure of the severity of the effect of the heat load. Further evaluation of stress due to non-uniformity of incident heat flux and of localized stresses should be made in the next phase of design. In general, these stresses are considered sufficiently small to not warrant concern.

The aft closure and the support tube will experience external pressure during operation and shutdown and are subject to buckling. For the aft closure the allowable external pressure is

$$\sigma_{cr} = \frac{2Et^2}{r^2 \sqrt{3\left(1-\nu^2\right)}}$$

and for the support tube

$$\sigma_{cr} = \frac{E}{4\ (1-\nu^2)} \left(\frac{t}{r}\right)^3$$

where

t = material thickness

r = radius of curvature

ν = Poissons ratio = 0.3

E = modulus of elasticity

= 20×10^6 psi at 700°C } 316 stainless steel

= 30×10^6 RT

These relationships show allowable external pressures far in excess of those applied--the minimum being 49 atmospheres for the absorber.

Gravity and intertia loads were considered for the structure. A 1.5 dynamic load factor was applied to gravity forces to cover inertial effects, with an additional 1.5 overall factor safety.

Maximum stress in the support tube is 2670 psi and the maximum membrane load per unit length at the intersection of the support tube conical section and the aft closure dome is 155 lbs/in. giving a stress of 1240 psi. As the cone elements feed tangentially into the spherical aft closure dome, very little kick or lateral components exist. These stress and load levels are sufficiently low to be considered non-problems.

Support Cone

The support cone is of 0.040-inch 316 stainless steel, the thickness set by handling. For crippling, the required thickness is conservatively given by the relationship

$$\sigma_{cr} = 0.3\ E\ \frac{t}{r}$$

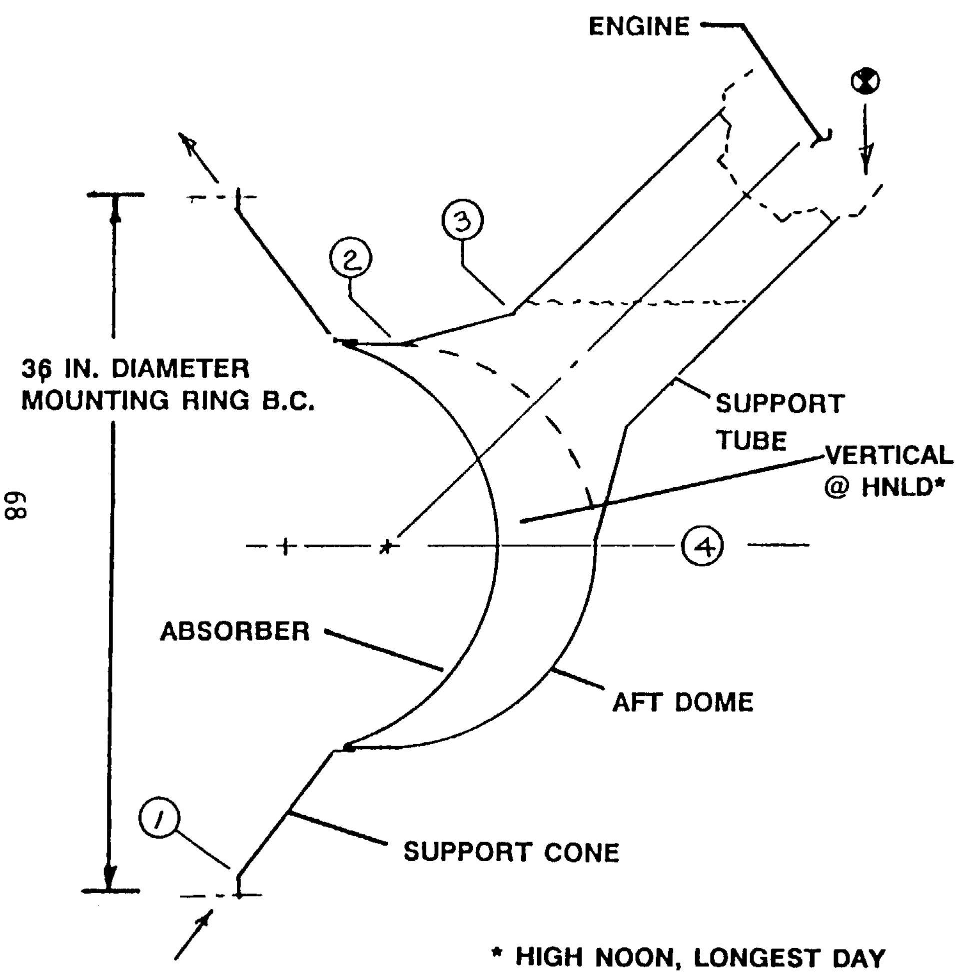
where the terms are as given earlier. The required thickness with a 1.5 dynamic load factor in addition to a safety factor of 1.5 is 0.014 inches. Design of this component is thus ultraconservative.

Heat loss by conduction to the mounting ring is estmated by first order analysis to be 151 watts. This could be reduced by reducing cone wall thickness. Longitudinal stiffness formed into the material to provide rigidity is transmitted to the mounting ring by twelve 9/16-inch bolts which are lightly loaded. A stiffener ring will spread the concentrated bolt loads to the thin conical shell.

The location of critical loads and stresses is given in Figure A-4 along with a summary in Table A-2 of loading and margins of safety. The structure can be further optimized for weight removal if the tradeoff with cost would warrant it.

Liquid Metal Working Fluid

Potassium was selected as the liquid metal heat transport fluid because of its favorable vapor pressure at 700°C. This pressure, 418 mm Hg (0.057 MPa), should be sufficient to assure steady nucleate boiling at the absorber back face. There is concern that the vapor pressure of sodium at 700°C (0.014 MPa) is insufficient to prevent unsteady boiling (bumping). Critical heat flux will be higher for potassium than sodium as the product of vapor density and heat of vaporization at this temperature is greater for potassium than for sodium resulting in a much lower volumetric vapor flow. A comparison of the properties of potassium and sodium at 700°C is in Table A-3.



MEMBER AND LOCATION	CRITERIA	INTENSITY	MARGIN OF SAFETY
SUPPORT CONE	BUCKLING	--	LARGE
ASORBER	CREEP THERMAL STRESS	0.8% 60,000 HRS	-- NOT CRITICAL
AFT DOME	BUCKLING CREEP	-- 0.4% 60,000 HRS	LARGE --
SUPPORT TUBE SECTION 2	BUCKLING	-- 1240 PSI (155 LB/IN.)	LARGE LARGE
SECTION 3		2670 PSI	LARGE
VAPOR/LIQUID FLOW PASSAGE	FLOODING	14,820 BTU/HR	2.6 HORIZONTAL 1.8 VERTICAL

Figure A-4. Critical Loads and Stresses

Table A-2

LIQUID METAL PROPERTIES

	MELTING POINT C	HEAT OF VAPORIZATION CAL/GM (BTU/LB)	VAPOR PRESSURE MPA	VAPOR DENSITY G/CC	LIQUID DENSITY GM/CC (700 C)	SURFACE TENSION DYNES/CM	KUTATELADZE FLOODING FIGURE OF MERIT
POTASSIUM	62.3	496 (876)	0.057	2.76×10^{-4}	.675	75	8.4
SODIUM	97.8	1005 (1827)	0.014	4.0×10^{-5}	.782	130	7.4

The potassium inventory is 54.0 pounds. This is mainly influenced by the size of the flow passage required for the liquid and vapor flow which is addressed in the next paragraph.

Flooding Limit

The required flow passage for vapor flow to the heater tubes and liquid return is given by the Kutateladze relationship.

$$\phi = \frac{h_v\, c^2 \, [\rho_v{}^2\, g\, (\mathrm{Sin}\ \alpha)\, \gamma\, (\rho_l - \rho_v)]^{0.25}}{[1 + (\rho_v/\rho_l)^{0.25}]^2}$$

where

h_v = heat of vaporization

ρ_v = vapor mass density

ρ_l = liquid mass density

γ = surface tension

α = angle of flow to horizontal

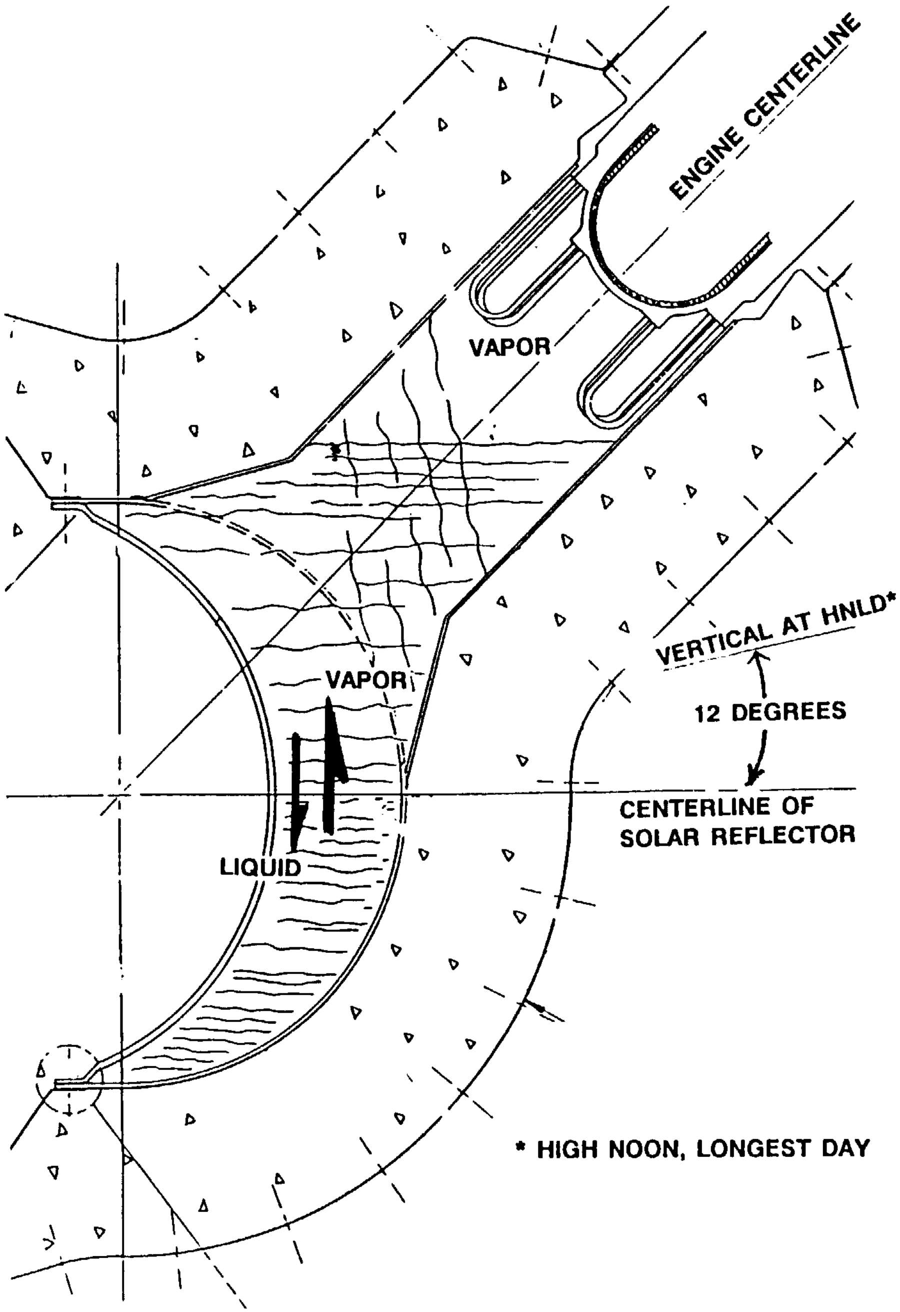
g = gravitational constant

c^2 = dimensionless constant = 3.2

ϕ = allowable heat flux

For the design presented, the critical region appears to be at the midplane for the system in both the horizontal and vertical orientations (see Figure A-5). By the above relationship, for horizontal operation (α =90°) the allowable heat flux is 6.42×10^4 BTU/hr-ft^2. For 29,610 BTU/hr (86.75 kW) total heat load the required flow area is

$$A = \frac{1/2\ (29{,}610)}{6.42 \times 10^4}\ (12)^2 = 33.2 \text{ in.}^2$$



ALLOWABLE HEAT FLUX = 86,400 BTU/HR-FT^2

PEAK HEAT FLUX = 24,000 BTU/HR-FT^2

KUTATELADZE FLOODING LIMIT

$$\phi = \frac{h\, c^2 \left[\rho_v{}^2\, g\, (\sin \alpha)\, \gamma\, (\rho_l - \rho_v)\right]^{0.25}}{\left[1 + (\rho_v/\rho_l)^{0.25}\right]^2}$$

ϕ = ALLOWABLE HEAT FLUX

h = HEAT OF VAPORIZATION

ρ_v = VAPOR MASS DENSITY

ρ_l = LIQUID MASS DENSITY

γ = SURFACE TENSION

α = FLOW ANGLE TO HORIZONTAL

g = GRAVITATIONAL CONSTANT

c^2 = DIMENSIONLESS CONSTANT = 3.2

Figure A-5. Flooding Limit

The current design has a flow area of 88.8 in.^{2} giving a factor of safety

$$M.S. = \frac{88.8}{33.2} = \underline{\underline{2.67}}$$

For vertical orientation (reflector axis 12° off zenith at Albuquerque, New Mexico, longest day at high noon) α =12° giving an allowable heat flux of 4.33×10^4 Btu/hr-ft^2 resulting in a factor of safety of 1.8. In a subsequent design phase (the angle between the engine centerline and solar reflector centerline could be reduced along with an increase in diameter of the conical section of the support cone) a minor modification of the geometry would increase the margin of safety for the vertical orientation.

Materials and Material Compatability

The materials selected for the heat transport system are discussed below. While cost and ease of manufacture is a fundamental and driving consideration, reliability and life is an overriding factor.

- POTASSIUM CONTAINER--The absorber, aft closure dome, and support tube are of AISI 316 stainless steel. This is a molybdenum alloyed 18-8 stainless steel of relatively good strength with extremely good performance in liquid metal environment and with a large amount of substantiating data and service experience. With potassium of reasonable purity, loss on the order of 0.001 inches/year could be expected based upon data presented in Appendix D.

- HEATER TUBES AND CYLINDER HEAD ASSEMBLY--This assembly completes the potassium containment. The cylinder head is of XF-818, an iron-based cast chrome/nickel/molybdenum alloy. The nickel content is moderate (19%) with no titanium or aluminum. It would therefore be expected to behave well in the potassium environment, although as yet, substantiating data has not been found.

The heater tubes are tentatively of CG-27, an iron-based chrome/nickel/molybdenum wrought alloy with significant amounts of titanium and aluminum. The nickel content is quite high (38%) which gives rise to concern regarding material transfer. Appendix D, Table D-3 shows for a very similar alloy (D66) a rather high rate of intergranular attack but moderate material transfer (0.0006 inches/year). An 8,000-hour test conducted by Stirling Thermal Motors (STM) of Ann Arbor, Michigan, of CG-27 coupons in a sodium heat pipe, however, reportedly showed little material degradation. The testing reported in Appendix D on D66 was for sodium flowing in an open loop and it is noted that CG-27 in a closed system may react quite differently. On the other hand, the validity of the STM test has been subject to some question because the coupons were reportedly loose (not electrically connected and therefore without the conductive path to form an electrochemical cell, and were located in the evaporator section. Further considerations are that 1) the tube OD which is exposed to potassium is not quite as highly stressed due to pressure as the ID because of the thick/thin cylinder effect, and 2) thermal stresses will be compressive at the OD. These factors tend to be beneficial. However, with relaxation of thermal stress, the tube OD will go into tension during shutdown which would give rise to concern regarding low cycle fatigue, although it is unlikely this would be critical.

With such conflicting and questionable information regarding compatibility of CG-27 in a potassium environment, it would appear premature to discard it as a prime candiate for use as heater tube material. Its exceptionally high strength (45,000 psi stress rupture at 60,000 hours at 700°C), potential low cost, and absence of strategic materials makes it ideal. Of all other materials considered, only Udimet 700 (15-20% CO) exceeds it in strength. Rene 41 compares well (39,000) but is costly and of high nickel content. Inconel 625 has sufficient strength but very high nickel content and high rates of material transfer are reported. Alloy D979 has very good strength but is similar to CG-27 in chemical constituency. Multimet N-155 (21 Cr, 20 Ni, 20 CO) and other cobalt alloys are likely candidates as cobalt tends to behave well in the alkali metal environments.

Stress rupture vs. temperature at 60,000 hours and stress vs. percentage creep in 1000 hours is shown in Figures A-6 and A-7 for some of the materials considered.

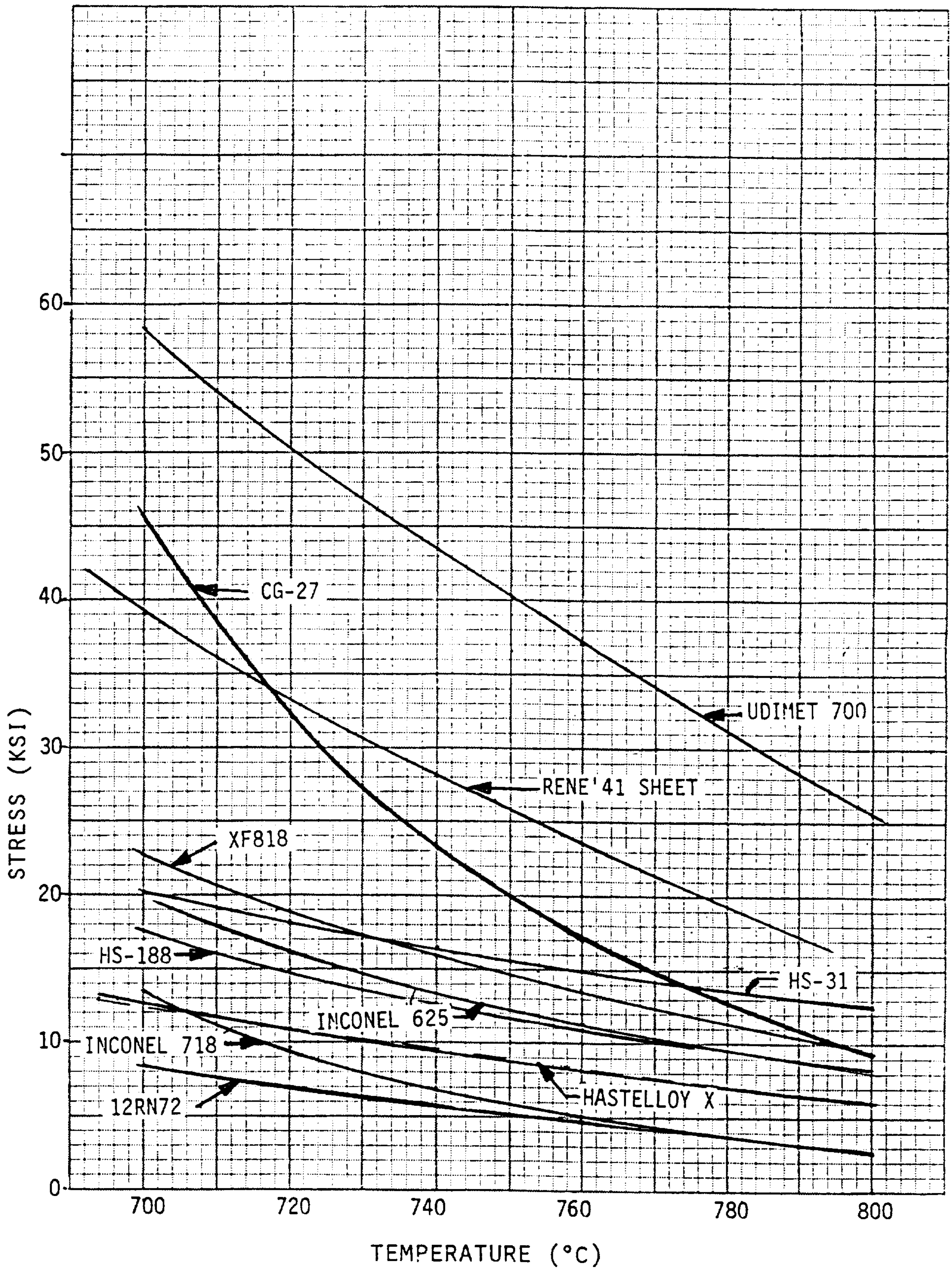


Figure A-6. Stress Rupture vs. Temperature for 60,000 Hours

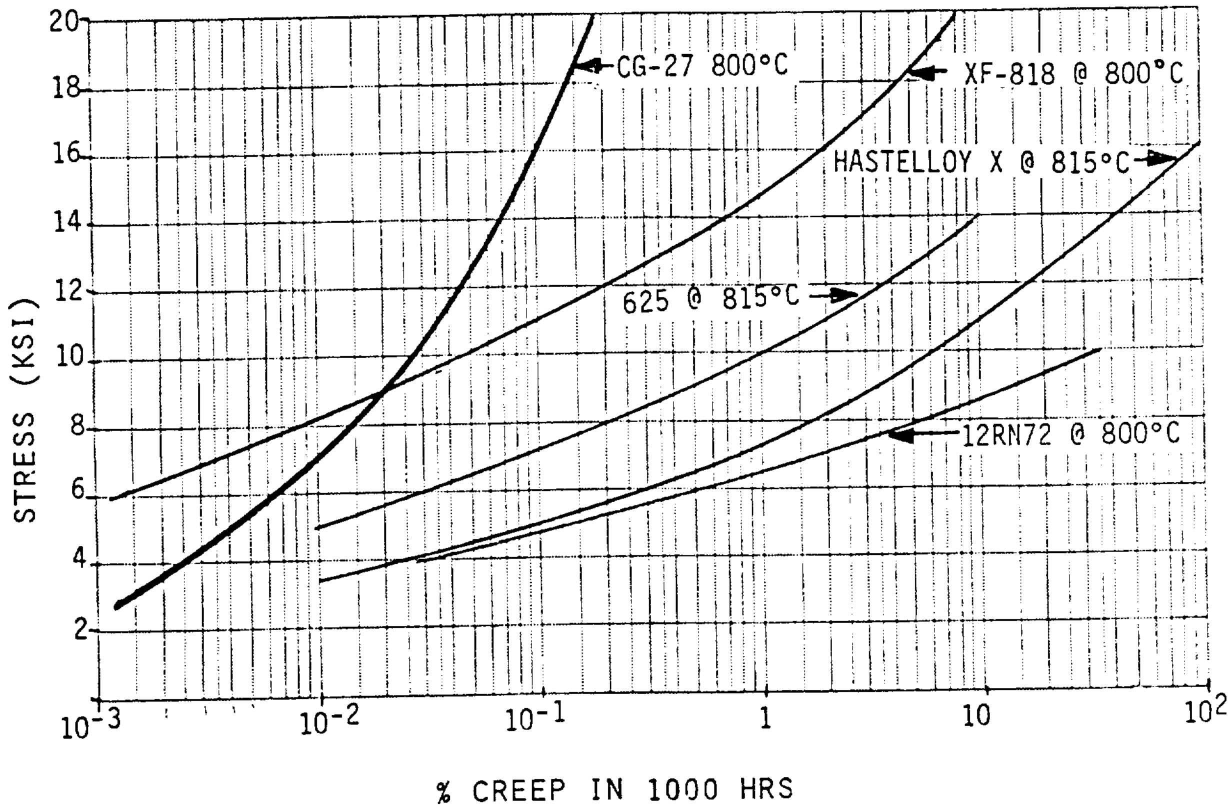


Figure A-7. Stress vs. Percentage Creep in 1000 Hours

The end conclusion regarding heater tube material at this point in the preliminary conceptual design phase is to retain CG-27 as the prime material candidate contingent upon compatibility testing. A true evaluation can only be made in a closed system which operationally and chemically closely simulates the actual configuration. Material transfer is not only dependent upon the alloys and their combination but upon the relative surface area exposure and the thermal conditions.

Appendix B

CAPILLARY HEAT PIPE

This Appendix was developed for sodium heat transport systems at 800°C. The conclusions reached are generally valid for potassium at 700°C.

Introduction

In the following study the effect of varying absorber diameter upon superheat in the wick of a sodium heat pipe is evaluated. The one dimensional flow model used represents the most vertical element of the circular absorber/evaporator and is shown in Figure B-1. It is a circular sector of small included angle cut from the absorber/evaporator. The wick is fed at its upper extremity by a feed artery of sufficient size that the flow loss in it is negligible, and at its lower extremity by the liquid pool. It is assumed that the flow in the model wick is symetric, that is, the flow upward equals the flow downward. As flow is on demand from the evaporator and solar flux is nominally symetric, this is not an unreasonable assumption as a first approximation. Cross flow from other portions of the absorber/evaporator which would feed liquid to the wick is ignored. The absorber surface is taken as a flat disk for simplification of analysis.

Evaporation from the wick is assumed to be uniform. If this is the case, flow of liquid in the wick across any radius will be proportional to the area bounded by that radius. For uniform pressure gradient in the liquid flow stream it can be shown that a wick that varies linearly in thickness from maximum at the outer diameter to zero at the center satifies this requirement. While this may not be economically or physically practical, it was taken as a baseline case and is presented for completeness. The study is mainly for wicks of uniform thickness.

Failure of a heat pipe is generally by burnout in the wick. Thix occurs when a sufficient supply of liquid is unable to reach the evaporator heat transfer surface. This can result from a clogged wick or by vapor in the wick preventing flow of liquid to the heat transfer site. Once nucleate boiling has commenced and a bubble formed, it is difficult to recover stable operation. Nucleate boiling is promoted by excessive superheat in the liquid.

In general there are two components which contribute to superheat in the wick, that due to the temperature gradient required to transfer heat through the wick to the liquid vapor interface (designated herein as ΔT_T), and that

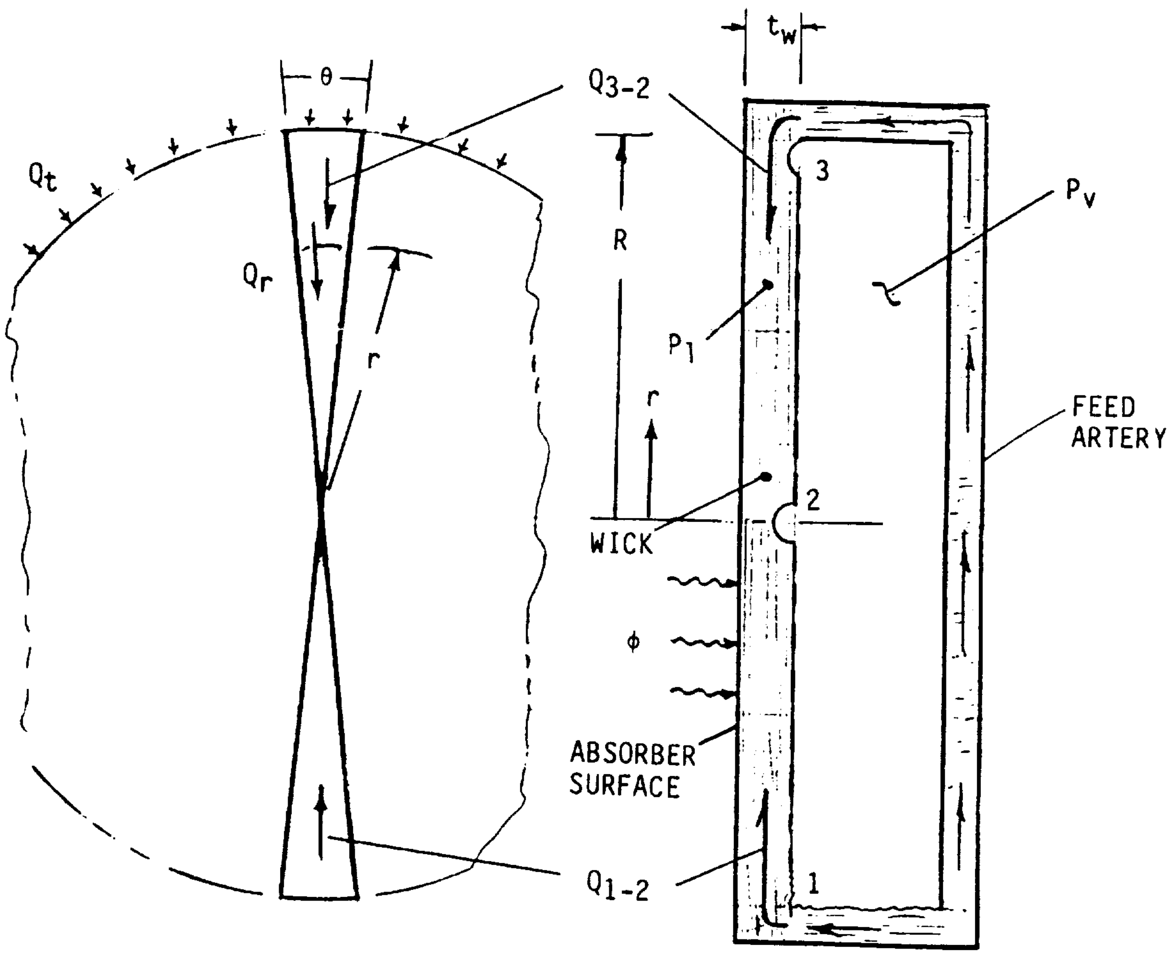


Figure B-1. Heat Pipe One-Dimensional Model due to reduced pressure in the wick relative to the equilibrium vapor pressure (designated ΔT_P). In the presentation of the data a factor of 2.0 is applied to the superheat resulting from thermal gradient to account for localized variation in solar heat flux incident on the absorber surface which reportedly can vary by that amount.

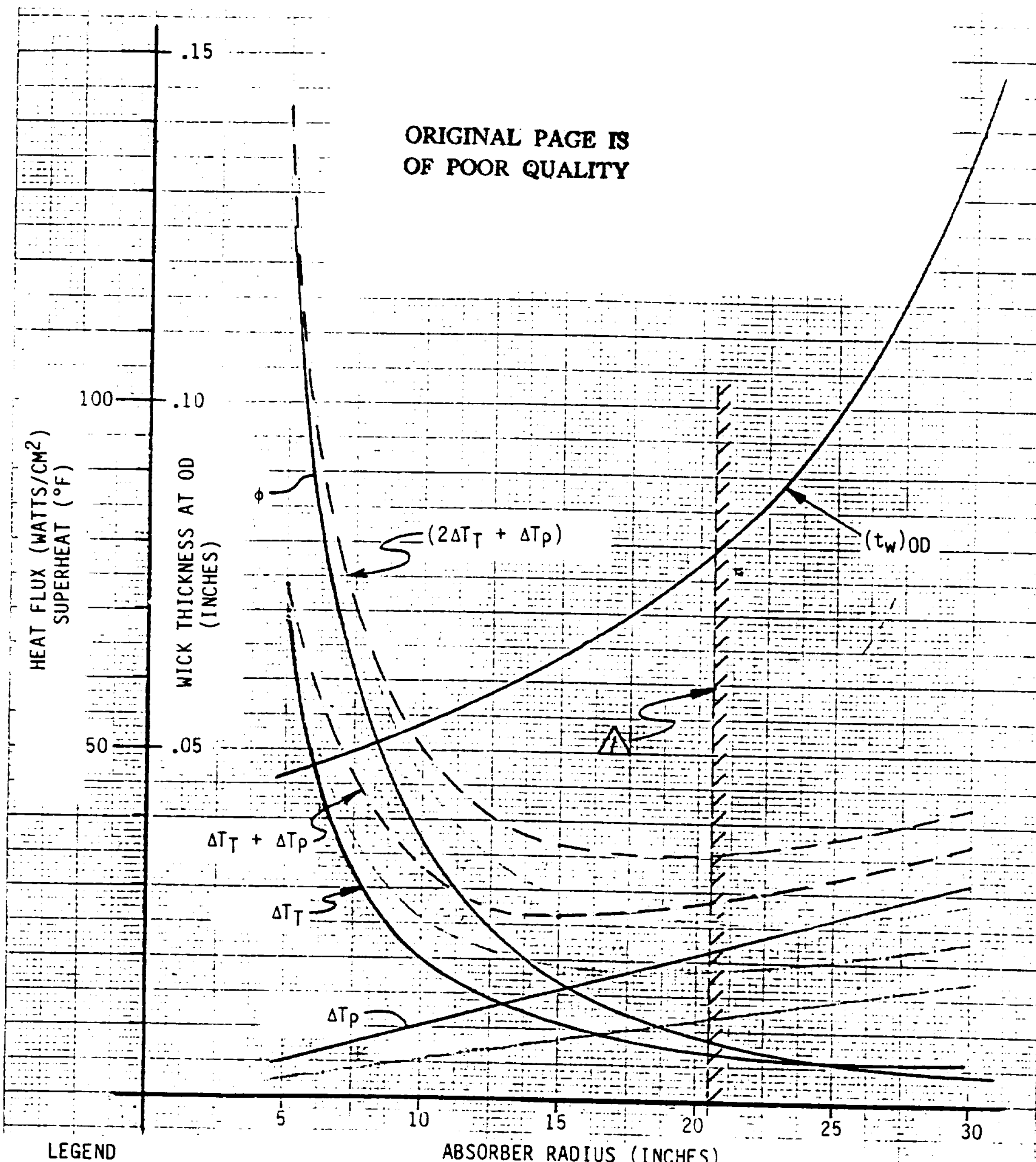
Some margin is necessary in the pumping pressure required to distribute the liquid relative to the capillary pumping pressure available. If the maximum pressure differential which the liquid surface tension will support is developed at the liquid-vapor interface, small perturbations such as dynamic forces can cause catastrophic failure. Pumping ratio is therefore defined as the ratio of capillary pressure used to drive the fluid system relative to the maximum available based upon the capillary pore size and the liquid surface tension. A pumping ratio of 0.5 means that one-half the available capillary pressure is used to move the fluid through the wick.

Results and Presentation

In Figures B-2 through B-6, the following parameters are shown.

- Incident solar heat flux ϕ
- Superheat due to thermal gradient in wick (ΔT_T)
- Superheat due to pressure differential across meniscus (ΔT_P)
- Total max superheat ($2\Delta T_T + \Delta T_P$)
- Wick thickness
- Capillary pumping limit

Figure B-2 is a plot of the above parameters for wicks that taper from zero at the center to maximum thickness at the outer edge. It is seen that total maximum superheat ($2\Delta T_T + \Delta T_P$) is minimum at a value of 35°F for an absorber of 17-inch radius. In this case the full capillary pumping pressure is used with no margin ($\eta = 1.0$). It is clear that at this superheat and with no pumping margin, there are no satisfactory absorber proportions. The capillary pumping limit shown indicates the maximum absorber radius imposed by the capillary pumping limit without regard for superheat in the wick.



LEGEND

t = WICK THICKNESS AT OUTER EDGE

ΔT_T = SUPERHEAT DUE TO THERMAL GRADIENT THROUGH WICK AT OUTER EDGE

ΔT_P = SUPERHEAT AT OUTER EDGE DUE TO PRESSURE DIFFERENTIAL ACROSS MENISCUS (RESULT OF STATIC + DYNAMIC LOSS)

ϕ = ABSORBER HEAT FLUX

NOTE: AT CENTERLINE $\Delta T_T = 0$
$\Delta T_P = 22.1°F$

⚠ LIMIT IMPOSED BY CAPILLARY PUMPING OF 400 MESH SCREEN

Figure B-2. Superheat at Upper Edge, Wick Thickness at Outer Edge, and Heat Flux vs. Absorber Radius--Tapered Wick Model Per $P_{WORKING} = P_{CAPILLARY}$ (n = 1.0)

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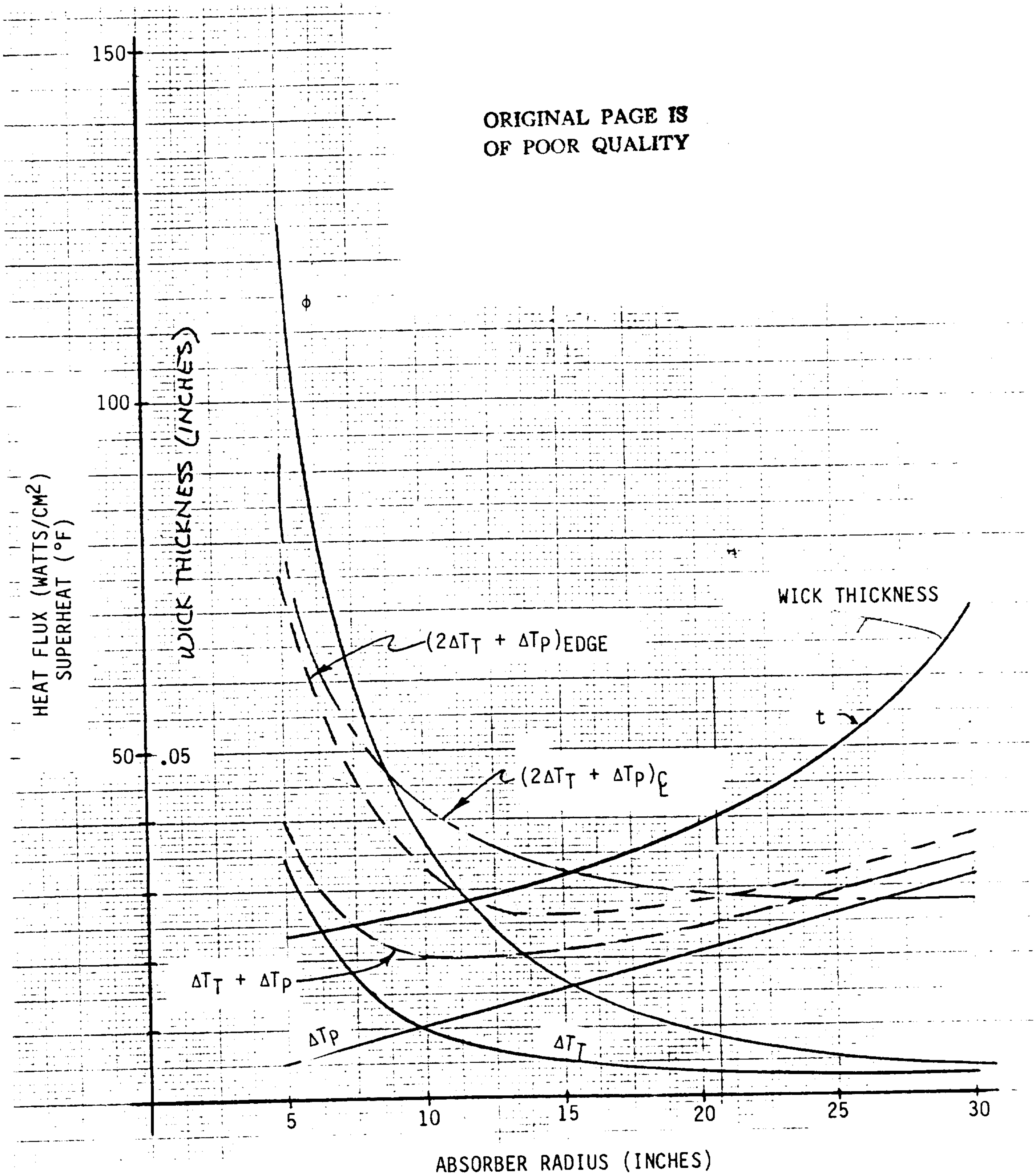

Figure B-3. Superheat Wick Thickness and Heat Flux vs. Absorber Radius-Uniform Wick Model Per Working Pressure = $P_{CAPILLARY}$ ($n = 1.0$)

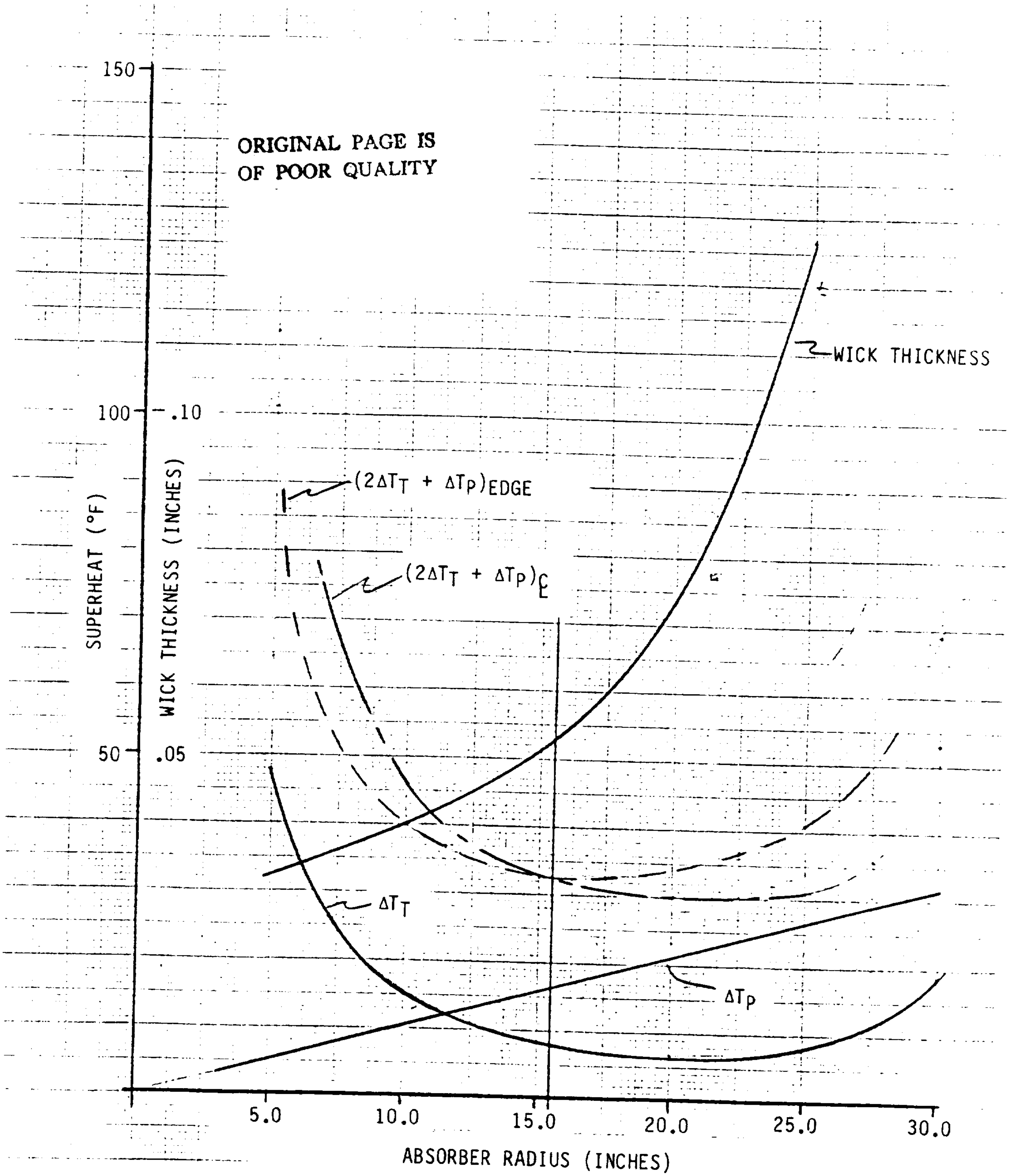

Figure B-4. Superheat and Wick Thickness vs. Absorber Radius--Uniform Wick (n = 0.75)

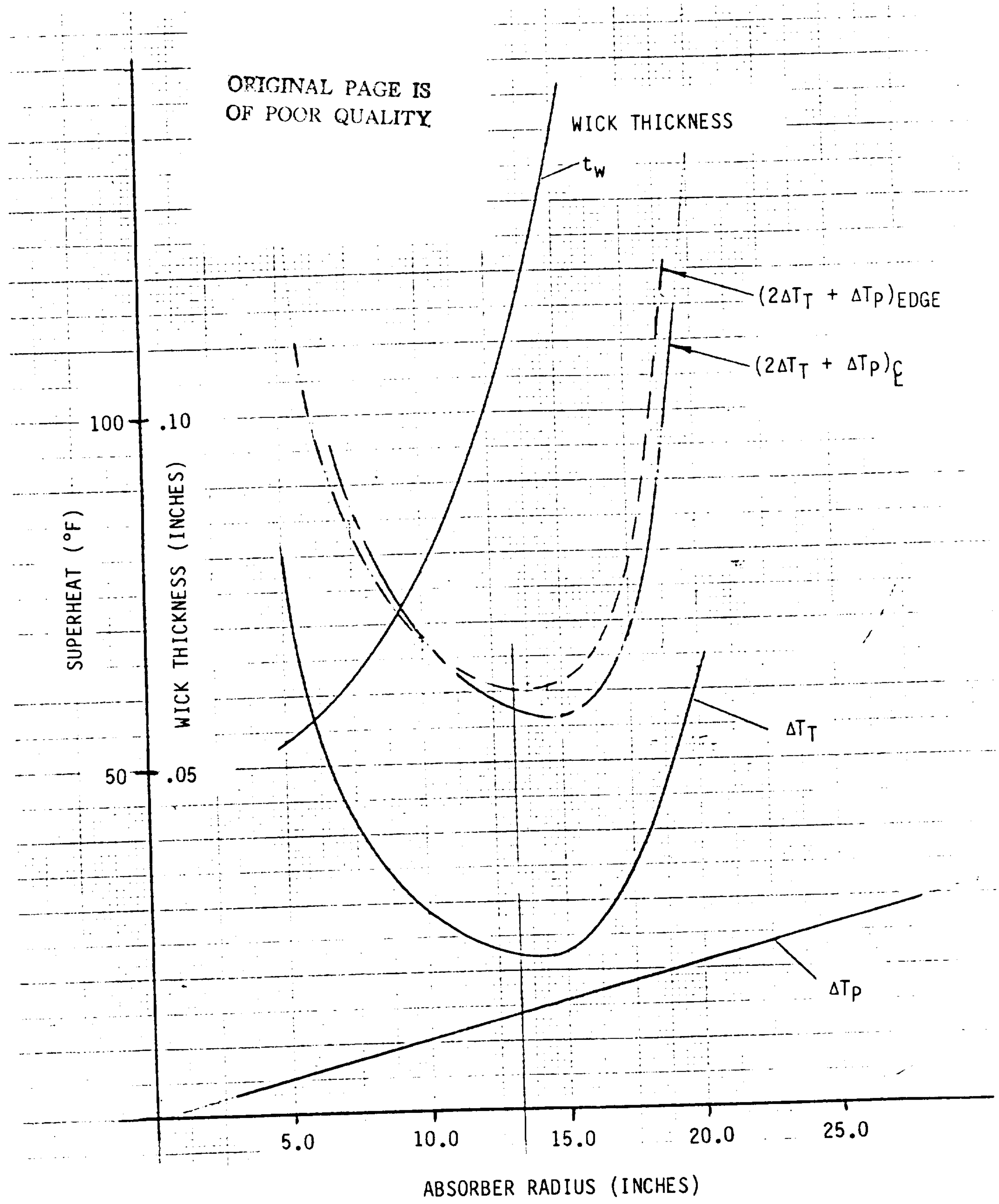

Figure B-5. Superheat and Wick Thickness vs. Absorber Radius--Uniform Wick (η = 0.5)

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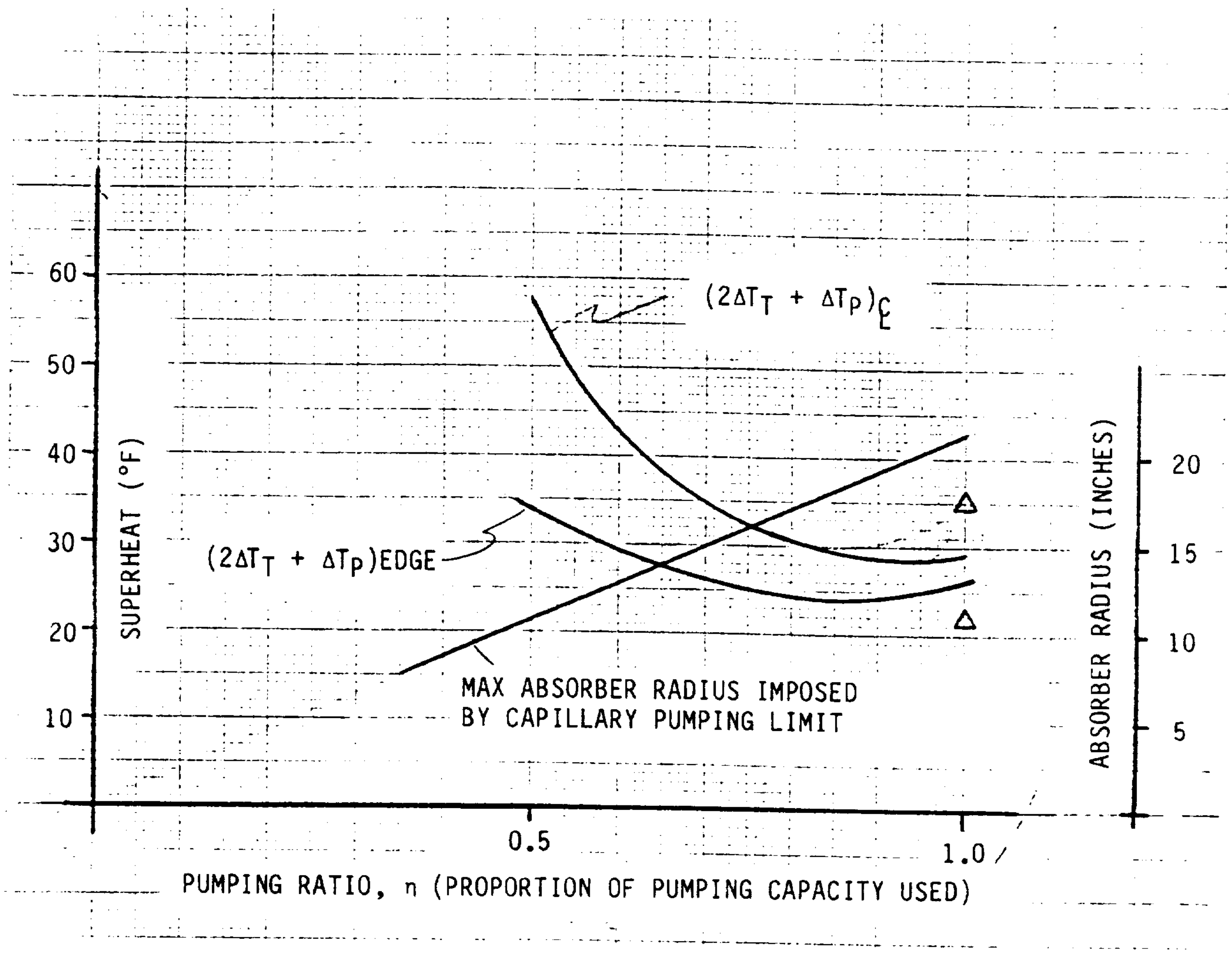

Figure B-6. Superheat at Optimum Absorber Diameter and Max Absorber Radius vs. Pumping Ratio (Uniform Wick Thickness)

Figures B-3 through B-5 present the same data for wicks of uniform thickness for pumping ratios of η = 1.0, 0.75, and 0.5. Total maximum superheat is shown for center and edge locations.

Figure B-6 is a summary plot of Figures B-3 through B-5 showing superheat at the optimum absorber radius as a function of proportion of pumping capacity used (for wicks of uniform thickness). Absorber radius for capillary pumping limit is also shown.

Conclusions

Tolerable superheat is dependent upon factors that are difficult to access or control. For the reliability required of the proposed system, a superheat of not more than 10 to 15°F with a pumping ratio not greater than 0.5 would be a minimum requirement. Figure B-6 shows the sensitivity of superheat to the percentage of capillary pumping capacity which is committed to distributing the liquid in the wick. It is seen that for η = 0.5, superheats at the outer edge and particularly at the center are above what would be considered safe limits. Superheat due to thermal gradient can be reduced by increasing wick permeability and thus reducing wick thickness. Permeability is increased, however by increasing the sub-surface pore size. This reduces the capacity of the wick to sustain its charge under dynamic conditions or to recover from impending burnout.

A further overriding factor prevails. There is no apparent way to sense a wick burnout until catastrophic failure occurs. The margins required to accommodate this risk would be prohibitive.

While it may not be impossible to build a wicked system to provide the required function, it is doubtful that it can be done economically with the reliability required for the proposed 30 year/60,000 hour lifetime.

Analytical Approach

The heat pipe model was described earlier and is shown in Figure B-1. The nomenclature used is as follows where the numerical values for sodium are taken at 800°C.

P_v = vapor pressure

P = static pressure in general

P_c = max capillary pumping pressure $= \frac{2\gamma}{r_p}$

η = pumping ratio (ratio of capillary pumping pressure to maximum available)

R = absorber/evaporator radius

S = absorber surface area

q = heat flow (4266 Btu/min at 75 kW level)

ϕ = solar heat flux

Q = volumetric flow

A = flow area

μ = absolute viscosity = 2.39×10^{-8} lb-sec/m^2

w = weight density = .0266 lb/in.^{3}

K = wick permeability = 1.20×10^{-7} in.^{2} (for 200 mesh St Stl screen)

r_p = pore radius

γ = surface tension = 7.02×10^{-4} lb-in.

h = heat of vaporization = 1812 Btu/lb

t_w = wick thickness

k_e = effective thermal conductivity = 2.22 Btu/hr in °F
(for sodium and St stl wick of 67% porosity)

Subscripts

v = vapor

l = liquid

1, 2, 3, etc. = locations noted

0 = unit value

w = wick

T = temperature

t = total

OD = outside diameter

In general, for flow in a wick the viscous flow loss is given by

$$\frac{dp}{dx} = \frac{\mu}{KA} Q$$

The total liquid flow to the wick for uniform heat flux is

$$Q_t = (C_1 \phi) \pi R^2$$

$$\text{where } C_1 = \frac{1}{hw}$$

or $$Q_t = \frac{q}{hw}$$

The flow across any radial element of the model (Figure B-1) is

$$Q_r = [\frac{Qt}{2\pi R^2}] r(r\theta)$$

The driving pressure in the wick for flow from Station 3 the Station 2 is

$$\left.\frac{dp}{dr}\right|_{3-2} = \frac{(P_l)_3 - (P_l)_2}{R} + w$$

and for flow from Station 1 to Station 2

$$\left.\frac{dp}{dr}\right|_{1-2} = \frac{(P_l)_1 - (P_l)_2}{R} - w$$

Liquid pressures in the wick at Stations 1, 2, and 3 are

$$(P_l)_1 = P_v$$

$$(P_l)_2 = P_v - \eta P_c$$

$$(P_l)_3 = P_v - 2wR$$

Substituting these values gives driving pressures in the wick

$$\overline{\Delta P}_{3-2} = \overline{\Delta P}_{1-2} = \eta P_c - wR$$

For a tapered wick where the wick thickness varies linearly from zero at the center to maximum at the edge (OD)

$$\frac{dp}{dr} = \frac{\mu}{K}\frac{Q}{A}$$

where $A = t_w(r\theta)$

$$t_w = \frac{(t_w)_{OD}}{R} r$$

giving

$$\frac{dp}{dr} = \frac{\mu}{K}\left[\frac{Q_t}{2\pi R}\right]\frac{1}{(t_w)_{OD}}$$

and the total pressure drop is

$$\overline{\Delta P}_{3-2} = \overline{\Delta P}_{1-2} = \int_0^R \left(\frac{dp}{dr}\right)dr = \frac{\mu}{K}\frac{Q_t}{2\pi(t_w)_{OD}}$$

Equating pressure drop and driving pressure for steady flow gives the required wick thickness

$$(t_w)_{OD} = \frac{\mu Q_t}{K2\pi}\frac{1}{nP_c - wR}$$

where

$$Q_t = \frac{q}{h} = \frac{4266\ \text{Btu/min}}{(60\text{sec/min})(1812\ \text{Btu/lb})(.0266\ \text{lb/in.}^3)} = 1.47\ \text{in.}^3/\text{sec}$$

(for 75 kW input)

For a wick of uniform thickness it can be shown that

$$t_w = \frac{1}{2}(t_w)_{OD}$$

Superheat in the wick due to thermal gradient is given by

$$q = \frac{k_e \Delta T_T \, S}{t_w}$$

$$\Delta T_T = \frac{q t_w}{k_e \pi R^2} \quad ; \; q = 4266 \text{ Btu/min}$$

Superheat in the wick due to pressure differential across the meniscus is:

at outer edge

$$(\Delta T_P)_3 = \frac{P_v - P_{l(3)}}{(dP/dT)_v}$$

at center

$$(\Delta T_P)_2 = \frac{P_v - P_{l(2)}}{(dP/dT)_v}$$

where $(dP/dT)_v$ = vapor pressure/temperature gradient for sodium at the operating temperature = 0.0919 Psi/°C from Figure B-7.

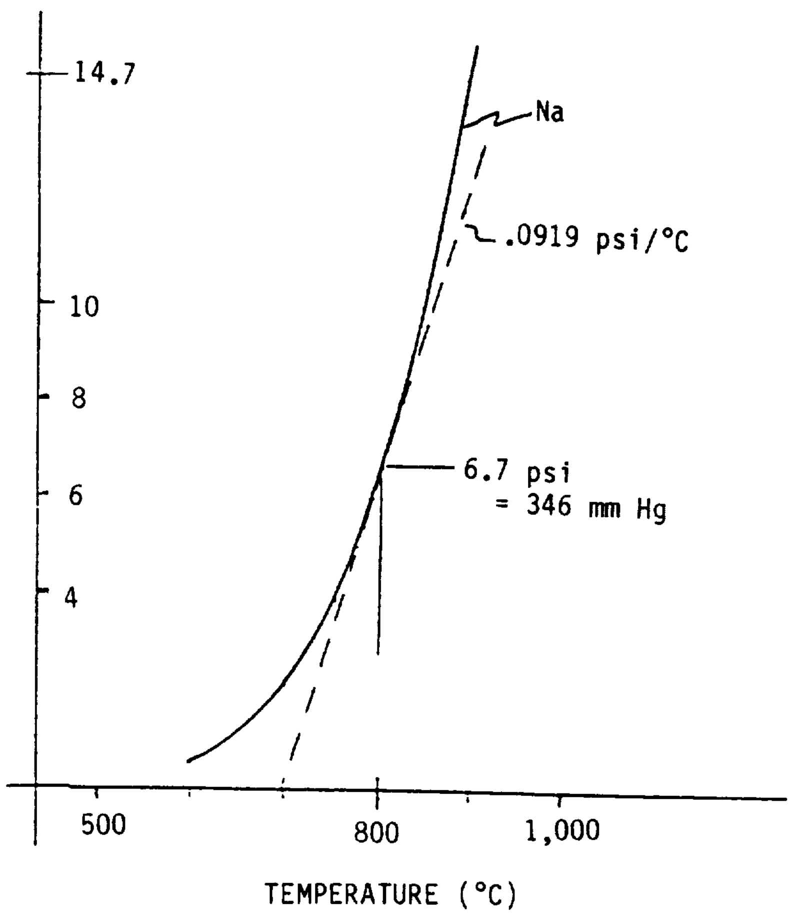

Figure B-7. Sodium Vapor Pressure vs. Temperature

Appendix C

This Appendix was developed for sodium heat transport systems at 800°C. The conclusions reached are generally valid for potassium at 700°C.

HEAT TRANSPORT SYSTEM OPTIONS

The chosen approach of a conventional tubular heater head involves multiple tubes connecting the upper end of the regenerator to the cylinder head, with the tubes passing through a plenum region where heat is picked up from the absorber. Five passive systems and one active system employing a pumped loop were considered as means to deliver the heat to the tubes. The alternative approaches are described briefly below.

A. Wicked Heat Pipe (Self-Priming)

This approach utilizes a conventional non-nucleate boiling wicked heat pipe. The absorber back surface is covered with wicking material and constitutes the boiler or evaporator. Sodium vaporizes from the surface and condenses on the heater tubes delivering heat to the engine. Condensate returns by gravity to the small liquid pool. Liquid is delivered to the upper regions of the boiler by a circumferential feed wick. For this system to be self-priming in all orientations, it was determined that a feed wick with a cross-section in excess of four square inches is required.

B. Wicked Heat Pipe (Non-Self-Priming)

This system is identical to A above except that a smaller feed wick is used and the all-orientation self-priming feature is foregone.

Wicked heat pipes are discussed in Appendix B.

C. Pool Boiling

This approach is a nucleate boiling reflux capsule. It utilizes nucleate boiling at the absorber back surface with the boiler cavity completely submerged in liquid sodium. An open area allows the heated vapor to circulate and condense on the heater tubes which are located above the pool. Condensate returns to the pool by gravity. The engine and its heater tube array is mounted so that it is above the sodium pool for all dish orientations.

D. Pool Immersion

A fifth system would have the heater tubes in the boiler cavity with the cavity filled with liquid sodium and pressurized to approximately one atmosphere to supress boiling. Convection cells would develop allowing heat transport by free convection in the liquid sodium pool. Atmospheric pressure would be maintained by argon or another inert gas with pressure regulation or a membrane interface with the atmosphere.

E. Two-Phase Boiling

This approach would have the tubes located in the boiler cavity which is partially filled with sodium. Under steady state conditions, violent boiling at the absorber back surface would cause the cavity to be filled with a two-phase liquid-vapor system which would wash both the absorber back surface and the heater tubes providing the required heat transport. Startup may be a problem requiring auxiliary heat.

F. Pumped Loop

This system employs an electromagnetic pump to supply liquid to the boiler surface in place of the capillary driven feed wick. Liquid sodium is picked up at the sump and delivered to circumferential feed tubes which supply the boiler surface. Sufficient flow and perhaps external cooling would be required to prevent premature vaporization of the liquid as it leaves the feed tubes. The boiler wick could be of minimal thickness to minimize superheat. Delivery of heat to the engine heater tubes by condensation will assure the uniform temperature requirement of the heater system.

Appendix D
MATERIALS

The following brief report on materials and material compatability is included as a supplement to the earlier discussion on materials and material compatability provided in the discussion of the receiver and reflux boiler in Appendix A. The report was submitted by G. D. Johnson, Manager of Materials Engineering for the Westinghouse Hanford Company.

The report focuses to a large extent on CG-27 as a candidate material for the heater tubes. While no data involving CG-27 explicitly is available, there is data for similar alloys.

Two concerns were addressed. One is the credibility of the extrapolation of creep rupture data for CG-27 at 60,000 hours. The other is the compatability of CG-27 with potassium in a closed system at 700°C.

Data for alloy D-979, which is very similar to CG-27, were generated in 60,000-hour tests and compare well with the extrapolated data for CG-27. This verifies to a first order the extrapolated data of Figure A-6, Appendix A.

Corrosion data were reported for similar alloys tested in an open plumbed sodium loop at 700°C. The most similar alloy to CG-27, D-66, did not perform too well. However, Johnson points out that CG-27 may perform extremely differently in a closed system. As compatability tests are proposed as part of the next design phase, it is considered premature to discard CG-27 in view of the advantages of high strength and low projected cost. CG-27 has therefore been retained as the primary choice of material for the heater tubes.

MATERIALS ISSUES FOR THE RECEIVER AND HEAT PIPE OF A 25 KW(e) SOLAR ELECTRIC ADVANCED STIRLING POWER SYSTEM

DESIGN REQUIREMENTS

The basic design requirements that directly impact the choice of materials for the receiver and heat pipe system are listed below.

Required Life:	30 years (calendar)
	60,000 hours (at temperature)
Startup/Shutdown Cycles:	20,000
Operating Temperature:	700°C
Heat Pipe Fluid:	Potassium

MATERIAL PROPERTIES

Selection of a material for the high temperature components of the Solar Electric Advanced Stirling Power System requires information on mechanical properties, corrosion behavior, weldability, fabricability, etc. During the conceptual design phase, the most important properties of concern are creep strength, stress rupture life and resistance to liquid metal corrosion. As the program progresses from a conceptual to a preliminary design, other properties need to be evaluated. They are low cycle fatigue, creep-fatigue interaction, oxidation behavior and weldability.

For the conceptual design study, the material to be used is CG27. The choice of this alloy was based on its excellent high temperature properties for application in automotive Stirling engines.(1) This alloy was produced by the Crucible Steel Corp. of Syracuse, New York. Since this alloy is no longer produced, other superalloys of similar composition should be considered for the purpose of providing material property data when such data are not available for CG27. Table 1 provides a listing of a few alloys that are similar in composition to CG27.

Creep and Stress Rupture Behavior

Information presented at the Preliminary Design Review in February 1987, included stress rupture and creep curves for several high temperature alloys. The stress rupture curves were for a time of 60,000 hours. In the short time spent on reviewing materials, it was not possible to locate the actual data base for CG27. In many cases correlations are provided for creep and stress rupture properties where actual test times are much less than those being considered for a given application. In an investigation conducted on alloy D-979,[2] data were generated to times of 60,000 hours at temperatures in the range of 1000-1500°F. Figures 1 and 2 show the properties of D-979 in comparison to CG27. This provides verification from actual data that the expected creep and stress rupture properties can be achieved for the conceptual design.

Liquid Metal Corrosion

A brief review of the literature did not provide any data on the corrosive interaction between potassium and superalloys (iron or nickel base). The behavior of alloy CG27 in potassium at 700°C can be inferred only from data for iron and nickel base alloys obtained with sodium loop tests. Numerous alloys were evaluated by several DOE contractors from 1975 to 1978 for the purpose of screening potential fuel pin cladding materials for use in liquid metal reactors. Most of these tests were conducted by Westinghouse at the Hanford Engineering Development Laboratory and the Advanced Reactors Division. Sodium compatibility tests were performed at 600°C and 700°C with 1 ppm oxygen. Sodium velocities ranged from 2.4 to 4.8 m/s and the Reynolds Number ranged from 2.28×10^4 to 4.5×10^4. Tests were conducted on commercial alloys and developmental alloys ("D"). Alloy compositions are given in Tables 1 and 2. Sodium exposure of tubular samples generally resulted in mass loss, wall thinning, chemically depleted zones, subsurface attack and intergranular attack (IGA). The depleted zone in D9 and AISI 316 is a uniform layer of ferrite. Subsurface porosity was observed in PE16, Inconel 706 and D68. Intergranular attack was observed for D9, D21, D25, D66 and PE16. Results of tests at 700°C are summarized in Table 3. The worst behavior was exhibited by alloy D66. This alloy is very similar to CG27. Sodium wastage correlations were developed for each alloy. The calculated performance for each alloy at 625°C for a time of 18,000 hours is shown in Table 4. Although these conditions do not apply to the heat pipe-Stirling engine system, these values are consistent with those in Table 3 with respect to ranking the alloys for comparative purposes. Solute strengthened austenitic alloys (AISI 316, D9) and ferritic alloys (HT9) show superior sodium corrosion behavior compared to the precipitation hardened alloys. It should be pointed out that the formation of a ferrite layer does not imply a total loss in strength in the layer. From a strength consideration, a porous depleted zone and intergranular attack would have a more deleterious effect. If these results from sodium loop tests indicate what might occur in a potassium heat pipe system, then alloy CG27 would appear to be a poor choice for 60,000 hour service at 700°C. On the other hand, a case could be made that the potassium heat pipe system is quite different than the sodium loop. The only way to evaluate CG27 for this case would be to run a series of refluxing capsule tests at 700°C. Destructive examination of the capsules at various times would be required to assess the long term behavior.

Low Cycle Fatigue-Creep Interaction

The evaluation of steady state creep and rupture life are valuable in a preliminary scoping evaluation. As the design becomes more rigid, it will be necessary to evaluate the interaction between creep and low cycle fatigue. The ASME design procedures for fatigue at high temperatures are based on Nuclear Code Case N47. The procedure for damage evaluation is a linear summation of creep and fatigue damage. "Safety factors" are applied to the stress (factor of 2) or the life (factor of 20). Many people use a strain range partitioning approach and several good papers on this subject can be found in Reference 3. In addition, several papers on this subject are listed in the literature survey, which is included as an attachment to this report.

Oxidation Resistance

The surface of the receiver will be subject to oxidation. Reaction between air and the alloy can be minimized with a protective coating. Nickel aluminide coatings are commonly used for this purpose and are applied by a diffusion process. A protective layer of Al_2O_3 is formed when the component is heated.

It has been reported[4] that oxidation can be enhanced under conditions of low cycle fatigue strain. During cycling, regions of intense local oxidation develop. This arises because of repeated rupture of the oxide film by the localized fatigue. Also, it has been shown that some of the degradation observed in low cycle fatigue-creep tests is really due to an interaction with the environment.[5] Thus, future evaluations of low cycle fatigue (LCF) data should pay careful attention to the test environment. If the receiver is to be coated, then the LCF data should be for tests run in vacuum or in an inert atmosphere.

Weldability

Most of the superalloys can be welded by conventional techniques. Since these alloys obtain their strength from a multiple step heat treatment, the welding will result in a very weak zone of material. A post-weld heat treatment would be required to maintain the high creep strength of the alloy.

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2. J. R. Wood, "Long-Time Creep-Rupture Properties and Elevated Temperature Stability of D-979," in Superalloys: Metallurgy and Manufacture, Ed. B. H. Kear, et al., AIME, 1976, pp. 227-235.

3. Low-Cycle Fatigue and Life Prediction, ASTM-STP-770, C. Amzallay, et al., Editors, 1982.

4. C. P. Sullivan and M. J. Donachie, Jr., "Microstructures and Mechanical Properties of Iron-Base Superalloys," Metals Engineering Quarterly, November 1971, pp. 1-11.

5. L. A. James, "Some Questions Regarding the Interaction of Creep and Fatigue," J. of Eng. Matls. and Tech., Trans. ASME, Vol. 98, July 1976, pp. 235-243.

Table D-1

ALLOY COMPOSITIONS

Alloy	Composition, Wt.%													
	Fe	Ni	Cr	Mo	Ti	Al	C	W	V	Nb	Co	Mn	Zr	Si	B
CG27	38.24	38	13.25	5.5	2.5	1.6	0.05	---	---	0.85	---	---	---	---	0.009
Alloy 901	34	42.7	13.5	6.1	2.5	0.25	0.05	---	---	---	---	0.45	---	---	0.015
Inconel 706	39.23	40	16	---	1.7	0.3	0.02	---	---	2.75	---	---	---	---	0.004
Hastelloy X	18.5	47.3	22	9	---	---	0.1	0.6	---	---	1.5	0.5	---	0.5	---
A-286	54.55	25	15	1	2.15	0.15	0.05	---	0.2	---	---	1.3	---	0.6	---
PE16	35	42.8	16.4	3.4	1.24	1.1	0.05	---	---	---	---	---	---	---	0.003
D-979	28	45	14.6	4.3	3.2	1.0	0.039	3.8	---	---	---	---	---	0.02	0.011

Table D-2

ALLOY COMPOSITIONS

Alloy	Composition, Wt.%														
	Fe	Ni	Cr	Mo	Ti	Al	C	W	V	Nb	Co	Mn	Zr	Si	B
D66	41.9	40	10.5	2	3.3	1.7	0.03	---	---	---	---	---	0.05	0.05	0.005
D68	48.3	34	12	---	1.7	0.3	0.03	---	---	2.9	---	---	0.04	0.05	0.005
D21	59.2	25	8.5	1	3.3	1.7	0.05	---	---	---	---	0.04	0.05	0.2	---
D9	Bal.	15.5	13.5	1.65	0.25	---	0.04	---	---	---	---	2.0	---	0.75	---
AISI 316	Bal.	13.5	17.5	2.5	---	---	0.05	---	---	---	---	1.75	---	0.6	---

Table D-3

SODIUM CORROSION OF VARIOUS ALLOYS AT 700°C
AND ONE-YEAR EXPOSURE

ALLOY	MASS LOSS $mg\text{-}dm^{-2}$	WALL THINNING μm	DEGRADED ZONE μm	FERRITE LAYER μm[a]	I.G.A. μm
Inconel 706	2600	6.0	48.5	--	0
Inconel 706	2880	12	55	--	0
PE16	2000	5.6	33.5	--	100
M-813 (Arc Cast)	2000	31	31	--	51
AISI 316	1442	18	--	12	0
D9	1442	18	--	22	0[c]
D21	2160	5[b]	58	--	30
D25	2000	5[b]	45	--	45
D66	4320	15	37	--	75.4

[a] Applicable to AISI 316 and D9 where ferritic layer is solid. Note that degraded zone is also ferritic, but it is porous and follows a linear time dependence.

[b] Estimated Values.

[c] No IGA seen in the short exposure time in ITF. IGA was observed at longer times in STCL, however.

Table D-4

SODIUM CORROSION OF ALLOYS AT 625°C FOR 18,000 HOURS

Alloy	Total Damage Depth (microns)
HT9	13
AISI 316	14
D9	29
D68	32
D25	39
IN 706	39
D21	46
PE16	47
D66	56

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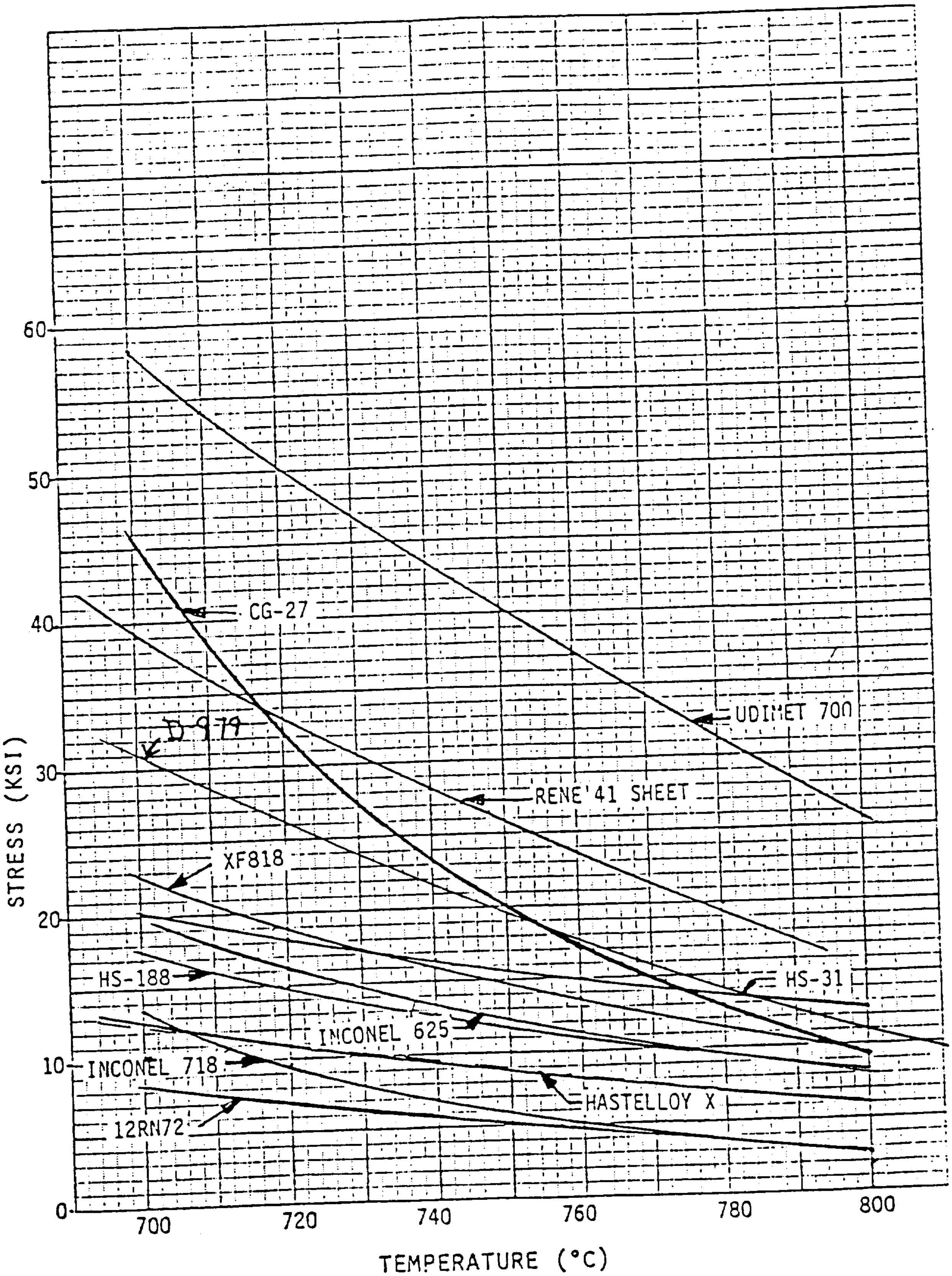

Figure D-1. Stress Rupture vs. Temperature for 60,000 Hr

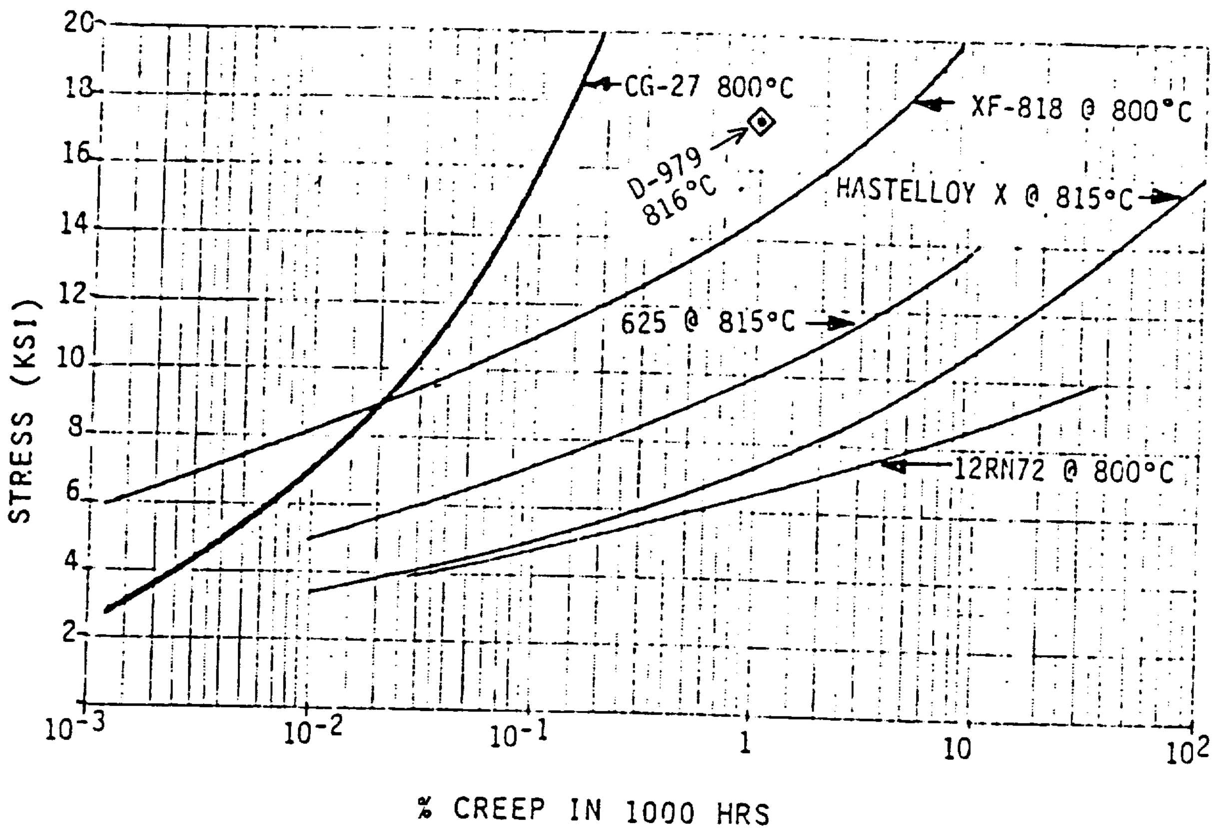

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Figure D-2. Stress vs. Creep in 1,000 Hrs

Appendix E

GEDEON ASSOCIATES REPORT

Supporting analysis for a Solar Electric Advanced Stirling Power System

Subcontractor Report

June 1987

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Reference: NASA LeRC contract DEN3-371

1 Background

This report covers work performed by Gedeon Associates for Stirling Technology Co. (STC) on a 25 kW solar electric Stirling power system. Under an overriding contract with NASA Lewis Research Center, STC subcontracted with Gedeon Associates to perform computer simulations of the Stirling thermodynamic cycle and to help with detailed heat exchanger design. Details of the solar absorber, hydraulic system, alternator and overall mechanical design are not covered in this report.

2 Method of Approach

Two key computer programs were used in the course of this work: A nodal-type Stirling thermodynamic analysis and a linearized scaling analysis. The GLIMPS (GLobally IMPlicit Simulation) nodal analysis [3] allowed fast and accurate simulation of trial designs while scaling analysis allowed designs to be varied to achieve performance trade-offs and design changes.

GLIMPS

This program is described in great detail in reference [3] and will not be re-described here. However, suffice it to say that GLIMPS solves the gas energy, momentum and continuity equations and is thereby better able to resolve some of the subtle effects of Stirling thermodynamics than isothermal or adiabatic analyses. For example, the problem of incomplete flow through heat exchangers (gas-element tidal excursion less than heat exchanger length) is resolved by GLIMPS but not by simpler analyses.

There is, however, at least one limitations of GLIMPS. Since the model assumes one-dimensional flow, it does not assess flow distribution or flow non-uniformity problems in heat exchangers. In spite of new 2-dimensional software currently under development (discussed in more detail later on), the best alternative remains to follow steady-flow manifold design guidelines and try to minimize flow non-uniformities as much as possible. For example, side-inlet manifolds feeding large frontal-area heat exchangers should be avoided.

Validating GLIMPS is an ongoing process, but the results look good so far and there is every reason to expect good accuracy with the type of engine chosen for the STC solar design. In an independent test of GLIMPS on Re-1000 engine data by Roy Tew at NASA "the code predicted 1.19 kW power at 31.2% efficiency" compared to "0.94 kW power and 27.2% efficiency" for the experimental data. These results are fairly good considering no parasitic losses (leaks, appendix loss, etc.) were simulated.

Lately there have been some concerns that GLIMPS (as well as most other Stirling analysis programs) have trouble matching data for low temperature-ratio space-power type engines. This type of engine is tough to simulate for several reasons. First, because of low temperature ratios, errors in gas-to-wall temperature drops in heat exchangers are more significant than for more conventional engines. Second, because of low tidal-amplitude-to-volume ratios in the heat exchangers, heat transfer tends to concentrate near the regenerator inlets. Third, the low phase angles between pressure wave and piston position mean that small errors in pressure phase angle result in relatively large changes in power. In contrast, the STC solar engine has

1. A high temperature ratio ($T_h/T_c = 3$ vs $T_h/T_c = 2$ for space engines).
2. High tidal-amplitude ratios in the heat exchangers (swept tidal volume / heat exchanger volume ≈ 3.2 in the cooler and ≈ 2.1 in the heater).
3. A very large pressure phase angle (≈ 50 degrees vs ≈ 10 degrees for space engines).

for all these reasons, GLIMPS is expected to do well when modeling the STC solar design.

Scaling

Scaling is based on linearized isothermal analysis. Stated broadly, the goal in scaling is to constrain certain dimensionless groups related to basic cycle power and heat exchanger performance while simultaneously changing some dimensional aspect(s) of the machine. The methodology of scaling is described in reference [4]. More detail on the specific problem of scaling the Solar engine can be found in section 7 of this report.

STC did not allow Gedeon complete freedom for the thermodynamic engine design but rather provided temperature, stroke and volume specifications and constraints to insure compatibility with their hydraulic system, free-piston dynamics and general mechanical design. Each specification set — of which there were several over the course of the work — consisted of values for the following:

- Power.
- Hot and cold metal temperatures.
- Pressure.
- Frequency.
- Piston and displacer strokes.
- Piston, displacer and drive-rod diameters.
- Heat exchanger volume ratios (compared to expansion space swept volume).

Within the guidelines of each data set was the freedom for Gedeon to chose heat exchanger details and displacer phase angle as needed to achieve the power determined by dynamic analysis at STC. There was also the freedom to reapportion volumes among the heat exchangers so long as the total gas mass within the system remained constant.

GLIMPS simulations and scaling were used together in a hands-on pseudo-optimization process. Upon receipt of the first data set, heat exchanger variables were chosen using rules of thumb and previous design experience, and a nodal simulation was run. Performance deficiencies were noted and corrected by scaling the design of the heat exchangers with appropriate constraints. The scaled design was then re-simulated and the entire process repeated until acceptable performance was achieved.

3 The Design Search

This section discusses the logic behind some of the design trade-offs made in the evolution of the thermodynamic design.

Pressure and frequency are two important design parameters confronting any Stirling engine designer. Fortunately, high pressure and moderate frequency designs seemed best to both STC and Gedeon. From STC's point of view, these designs tended to have low hydraulic losses. From Gedeon's point of view, the scaling trends indicated that high pressures and moderate frequencies would help to keep heat-exchanger tube counts low. Although moderate frequency runs somewhat counter to current aerospace practice, it makes sense in a solar application because weight is not a critical factor — particularly for the STC design, which has a ground-mounted electric generator.

Regenerator type was another important choice. The idea of using a rolled foil regenerator was originally suggested by STC in light of its potentially low cost and similarity to the simple annular gap regenerators (between the displacer and its cylinder) used in their artificial heart engines. STC also concluded that the absence of radial porosity would eliminate radial flow irregularities; and that there was potential for eliminating circumferential flow irregularities too by inserting longitudinal partitions between foil layers. Gedeon was quickly won over; especially after a review of the work done by Andy Ross suggesting that foils down to 25 micron (1 mil) thickness could be used. Also, foils sidestep the issue of enhanced axial gas conductivity which is a problem in flow through porous materials that has only recently been recognized by the Stirling community. More on this later.

Offsetting the cost and analytically perceived thermodynamic advantages of the rolled foil regenerator is the fact that experience with rolled foil matrices is quite limited. In consideration of this point, made by many of the reviewers at the preliminary design review meeting held at NASA Lewis Research Center, the wire screen matrix was eventually designated the baseline design with the foil regenerator as an advanced-technology backup. Prior to this decision, however, considerable effort had gone into designing a foil based machine. So instead of starting all over, we looked into the possibility of simply substituting a screen type regenerator in the existing design. The conclusions regarding foil- and screen-regenerator engines are discussed separately in sections 4 and 5.

The heater design was another open question. Early on, the proprietary heater design initially proposed by STC was judged to be too risky and the consensus was that a tubular design should be used. After a brief discussion between Gedeon and STC during which a straight-tube package in a shell-and-tube arrangement was considered, we quickly decided that, for packaging reasons, a hairpin-tube design was best.

Another issue was heat exchanger swept-to-dead volume ratio which was considered in great detail. Current problems in some of the NASA sponsored aerospace engines are, in part, directly traceable to the relatively small volumetric flow excursions in the heat exchanger volumes. Fortunately, Seume and Simon [10] have recently tabulated values for swept volume ratios in the heat exchangers of a wide variety of Stirling engines — some successful, some not. We decided to design the solar engine with its heater and cooler swept volume ratios well into the range of the most successful Stirling engines developed to date.

Regenerator flow distribution problems were yet another important issue to consider. We did our best to avoid potential problems although exact analysis was not possible. A compact, low void/swept volume design was generally helpful in avoiding flow distribution problems. The interface between the heat exchanger tubes and the regenerator was examined in minute detail and some suggestions for flow distribution improvements in this area are discussed later on.

Another issue was hot-end temperature. For the 800C hot-end temperature initially specified by NASA, the low allowable thermal stresses in hot components made for relatively thick pressure walls. At 800C, a maximum stress of somewhere between 70 and 100 mPa (10,000 and 15,000 PSI) was appropriate based on creep-rupture and desired lifetime critera. This led to the investigation of lower temperature 700C designs.

4 Final Designs for Foil Matrix Engines

This section documents the final — and best — foil-regenerator design points that were achieved. Run 3A.3 represents a 193 bar (2800 psi) design with somewhat-optimized heat exchangers and is the direct ancestor to all the other designs in this section — that is, all the other designs were obtained by simple scalings starting with 3A.3. To avoid confusion, a complete discussion of all the preliminary designs that led up to run 3A.3 are not reported. Essentially, 3A.3 evolved from heat exchanger geometric optimization and volume re-apportionment starting from a set of dynamically consistent temperatures, strokes and volumes provided by STC. The progression from run 3A.3 to 4A.2 was

3A.3 → 4A.1 Pressure reduced to 138 bar (2000 psi); volumes increased to hold power fixed.

4A.1 → 4A.2 Reduced heater tube count from 104 to 60 by scaling NTU by 0.83 but fixing heater volume and pressure drop.

Around the time run 4A.2 was completed, it was found that the 800C heat source temperature was too high for structural integrity and liquid metal compatability. Also, to generate power at the system *survival* heat input (rather than shut down) required an increased engine power output capablity.

STC then provided scoping set 5 for further thermodynamic analysis — a 700C, 138 bar (2000 psi), 35 kW design. The most notable effect of reducing the heat source temperature was a slight increase in the ratio of displacer swept volume to power piston swept volume. The ratios of heat exchanger volumes to displacer swept volume were maintained at the values selected for scoping sets 3 and 4. The progression from run 4A.1 to run 5A.3 was

4A.1 → 5A.1 Reduced hot-end temperature from 1045K to 973K increased power (volumes) slightly to provide a buffer above nominal power rating.

5A.1 → 5A.3 Reduce heater tube count from 171 to 75 by scaling NTU by 0.75 but fixing heater volume and pressure drop.

The remaining two runs, 5A.2 and 5A.4 are half-power points obtained by reducing frequency, piston stroke and displacer phase angle of runs 5A.1 and 5A.3 in accordance with STC dynamic analysis. Tables 1 and 2 summarise the key engine specifications and performance of the selected designs.

At this point the preliminary design review meeting took place, resulting in the following conclusions regarding the direction the engine design should take:

1. Use 700C as the design heat source temperature.

2. Freeze the displacer and power piston stroke volumes.

3. Increase the design mean pressure to achieve the engine power needed for operation at survival levels of insolation.

4. Use a screen or screen-like matrix in the baseline design.

Design points which reflect these conclusions are discussed in the following section.

Run ID number	3A.3	4A.1	4A.2	5A.1	5A.2	5A.3	5A.4
Mean pressure (mPa)	19.3	13.8	13.8	13.8	13.8	13.8	13.8
Frequency (Hz)	50	50	50	50	38.5	50	38.5
Gas type	He	He	He	He	He	He	He
Piston amplitude (mm)	5.08	5.08	5.08	5.08	4.09	5.08	4.09
Displacer amplitude (mm)	10.2	10.2	10.2	10.9	10.9	10.9	10.9
Displacer Phase (deg)	45	45	45	45	34	45	34
Displacer appendix gap (mm)	0.13	0.15	0.15	0.17	0.17	0.17	0.17
Displacer length (mm)	150	150	150	150	150	150	150
Displacer diameter (mm)	128	151	151	174	174	174	174
Displacer rod diameter (mm)	44	52	52	58	58	58	58
Piston diameter (mm)	153	182	182	208	208	208	208
Cooler wall temperature (K)	323	323	323	323	323	323	323
Cooler tube length (mm)	100	100	100	100	100	100	100
Cooler tube number	96	122	122	170	170	170	170
Cooler tube internal diameter (mm)	2.60	2.74	2.74	2.73	2.73	2.73	2.73
Regenerator length (mm)	59	59	59	58	58	58	58
Regenerator canister area (cm^2)	78	102	102	147	147	147	147
Regenerator foil thickness (μm)	25.4	25.4	25.4	25.4	25.4	25.4	25.4
Regenerator foil gap (μm)	32.0	37.8	37.8	36.7	36.7	36.7	36.7
Heater wall temperature (K)	1045	1045	1045	973	973	973	973
Heater tube length (mm)	194	194	215	184	184	214	214
Heater tube number	82	104	60	171	171	75	75
Heater tube internal diameter (mm)	3.18	3.34	4.18	3.16	3.16	4.42	4.42
CS volume at mean Xp and Xd (cm^3)	182	255	255	356	356	356	356
ES volume at mean Xd (cm^3)	171	240	240	335	335	335	335

Table 1: Key engine parameters for final designs. Displacer appendix gap was chosen for minimal efficiency loss considering shuttle heat transfer and appendix power losses.

Run ID number	3A.3	4A.1	4A.2	5A.1	5A.2	5A.3	5A.4
POWER							
GLIMPS Total PV power (kW)	34.36	34.55	33.85	42.04	21.43	40.65	20.93
Displacer rod power (kW)	3.64	3.63	3.25	3.36	1.30	2.40	0.93
Appendix gap PV loss (kW)	0.17	0.13	0.13	0.17	0.17	0.17	0.17
Piston PV (kW)	30.55	30.79	30.47	38.51	19.96	38.08	19.8
EFFICIENCY							
Piston indicated (%)	51.5	51.7	51.6	49.9	51.8	50.0	51.9
HEAT INPUT							
Basic heat input to heater (kW)	59.30	59.55	59.03	77.24	38.54	76.19	38.16
Shuttle heat transfer (kW)	0.17	0.17	0.17	0.17	0.04	0.17	0.04
Other thermal losses	NA	NA	NA	NA	NA	NA	NA
PRESSURE							
CS pressure amplitude (mPa)	2.99	2.13	2.13	2.00	1.35	1.99	1.35
CS pressure phase (deg wrt piston)	-49.3	-49.8	-49.3	-50.4	-73.3	-49.8	-72.7
COOLER							
Flow friction dissipation (kW)	0.56	0.51	0.51	0.81	0.30	0.81	0.30
Available energy loss (kW)	1.97	1.96	1.98	2.70	0.95	2.75	0.96
Mean gas-wall JT (K)	23	23	24	23	16	23	16
REGENERATOR							
Flow friction dissipation (kW)	3.02	3.04	3.05	4.22	2.35	4.25	2.36
Available energy loss (kW)	1.68	1.70	1.68	2.26	1.39	2.23	1.38
Enthalpy transport (kW)	1.39	1.57	1.58	2.37	1.60	2.36	1.61
HEATER							
Flow friction dissipation (kW)	1.26	1.19	1.49	1.59	0.66	2.21	0.92
Available energy loss (kW)	1.40	1.41	1.50	1.81	0.51	2.01	0.56
Mean gas-wall JT (K)	43	43	52	38	21	49	27

Displacer rod power is the power transmitted by the displacer rod required for overcoming mechanism losses outside the Stirling working gas; rod power = GLIMPS total PV power − GLIMPS piston PV power.

Appendix gap PV loss is an external calculation not included in GLIMPS.

Basic heat input to heater does not include parasitic losses.

Available energy loss is a *Second-law* power loss due to irreversibility associated with gas-to-wall heat flow across finite ΔT. Roughly, the net work a Carnot engine could deliver in a reversible process with equivalent heat flows across the gas-to-wall ΔT.

Table 2: Performance of final designs predicted by GLIMPS version 1.1.

5 Final Designs for Screen Matrix Engines

Several months elapsed between the simulations of this section and those of section 4. During this time a new version of GLIMPS was developed. A major difference in the new version is that gas + solid-mode axial conductivity in the regenerator is now built into the simulation rather than added in later as a parasitic calculation. The values of heat input tabulated in this section reflect this fact. To bridge the gap between this and the previous section, run 4A.2 was re-simulated with the new version of GLIMPS and the results tabulated as run 4A.2*. Based on this comparison, the new version of GLIMPS predicts lower efficiency by about 2.5%, of which less than 1% can be accounted for by the added axial conductivity. Tables 3 and 4 summarize run 4A.2* and the other results of this section.

STC suggested that target power might be satisfactorily met at 700C by simply increasing the pressure of the 4A.2 design from 138 bar (2000psi) to 179 bar (2600psi). This is run 6.1 although pressure was actually simulated as 174 bar due to a data entry error. Run 6.1 still has the same heat exchangers and foil regenerator as run 4A.2. The results appeared promising although no attempt was made to insure validity of the free-piston dynamics of run 6.1.

STC then provided scoping set 7 to re-establish the validity of the free-piston dynamics at 179 bar and 700C. The volume ratios and stroke ratios are the same as scoping set 5. Run 7.1 continues to use the same heat exchangers and foil regenerator as run 4A.2. A volume adjustment was made in the compression space to keep the total gas mass content consistent with the STC scoping guidelines. Compared to runs 5A.1 and 5A.3, run 7.1 shows about 4% lower efficiency, although only about half of this is due to thermodynamic effects, the remainder being due to the new version of GLIMPS. With the compact design of case 7.1, it is a strong contender for the baseline foil-regenerator design.

Screen-regenerator simulations begin with run 7.2 which represents an effort to package a screen regenerator into the same canister as in run 4A.2. Some hand optimization was done on wire diameter and porosity. A wire diameter of 50.8 μm (2 mil) and a porosity of 70% was chosen. Unfortunately, flow friction dissipation for the regenerator was not constrained and it turn out about 3kW higher than expected. Therefore run 7.2 is not consistent with the STC dynamic analysis and, as a result, the piston indicated efficiency is meaningless (it implies a small power input through the displacer rod). However, the gross indicated efficiency (ratio of total PV power to heater heat input) is meaningful; unfortunately, it was down about 4% due to large enthalpy flux and axial conduction the regenerator.

Another attempt to package a screen regenerator in the foil canister of 4A.2 is run 7.5. In this case the wire diameter was constrained at 25.4 μm (1 mil) and the regenerator pumping dissipation to about 3kW for consistency with the STC dynamic analysis. This case was a dismal failure. The porosity consistent with the above constraints was about 95%, giving very large values for enthalpy

Run ID number	4A.2*	6.1	7.1	7.2	7.4	7.5
Mean pressure (mPa)	13.8	17.4	17.9	17.9	17.9	17.9
Frequency (Hz)	50	50	50	50	50	38.5
Gas type	He	He	He	He	He	He
Piston amplitude (mm)	5.08	5.08	5.08	5.08	5.08	5.08
Displacer amplitude (mm)	10.2	10.2	10.9	10.9	10.9	10.9
Displacer Phase (deg)	45	45	45	45	45	45
Displacer appendix gap (mm)	0.15	0.15	0.15	0.15	0.15	0.15
Displacer length (mm)	150	150	150	150	150	150
Displacer diameter (mm)	151	151	152	152	152	152
Displacer rod diameter (mm)	52	52	51	51	51	51
Piston diameter (mm)	182	182	182	182	182	182
Cooler wall temperature (K)	323	323	323	323	323	323
Cooler tube length (mm)	100	100	100	100	100	100
Cooler tube number	122	122	122	122	122	122
Cooler tube internal diameter (mm)	2.74	2.74	2.74	2.74	2.74	2.74
Regenerator length (mm)	59	59	59	59	27	59
Regenerator canister area (cm^2)	102	102	102	102	190	102
Regenerator foil thickness (μm)	25.4	25.4	25.4	NA	NA	NA
Regenerator foil gap (μm)	37.8	37.8	37.8	NA	NA	NA
Regenerator wire diameter (μm)	NA	NA	NA	51	25	25
Regenerator porosity (%)	60	60	60	70	70	95
Heater wall temperature (K)	1045	973	973	973	973	973
Heater tube length (mm)	215	215	215	215	215	215
Heater tube number	60	60	60	60	60	60
Heater tube internal diameter (mm)	4.18	4.18	4.18	4.18	4.18	4.18
CS volume at mean Xp and Xd (cm^3)	255	255	295	295	295	295
ES volume at mean Xd (cm^3)	240	240	260	260	260	260

Table 3: Key engine parameters for final designs. Displacer appendix gap was chosen for minimal efficiency loss considering shuttle heat transfer and appendix power losses.

flux and axial conduction and large temperature swings in the regenerator gas — in other words, poor thermal performance.

Run 7.4 is the result of an effort to optimize a screen regenerator in its own canister. An intermediate result (run 7.3) is not tabulated. The length and frontal area of the regenerator were allowed to vary while the void volume was kept the same as run 4A.2. Regenerator pumping power was again constrained to about 3kW. The results of this case look pretty good showing about the same power and efficiency as run 7.1. Run 7.4 was designated the baseline STC solar design.

Run ID number	4A.2*	6.1	7.1	7.2	7.4	7.5
POWER						
GLIMPS Total PV power (kW)	34.34	39.93	42.15	36.12	42.20	22.23
Displacer rod power (kW)	3.06	4.57	3.43	-0.30	3.88	1.10
Appendix gap PV loss (kW)	0.13	0.17	0.18	0.18	0.18	0.18
Piston PV (kW)	31.15	35.19	38.54	NA	38.14	21.05
EFFICIENCY						
Piston indicated (%)	49.0	44.9	45.9	NA	45.7	28.2
HEAT INPUT						
Basic heat input to heater (kW)	63.56	78.33	83.88	78.03	83.43	74.67
Shuttle heat transfer (kW)	0.17	0.19	0.22	0.22	0.22	0.22
Other thermal losses	NA	NA	NA	NA	NA	NA
PRESSURE						
CS pressure amplitude (mPa)	2.23	2.70	2.70	2.56	2.70	2.00
CS pressure phase (deg wrt piston)	-47.6	-43.5	-48.1	-47.6	-47.6	-33.2
COOLER						
Flow friction dissipation (kW)	0.29	0.39	0.49	0.51	0.49	0.60
Available energy loss (kW)	2.44	3.38	3.72	3.68	3.65	5.63
Mean gas-wall JT (K)	27	28	29	29	29	36
REGENERATOR						
Flow friction dissipation (kW)	3.09	2.95	3.45	6.44	3.05	3.29
Available energy loss (kW)	1.67	2.52	3.13	3.00	1.78	8.60
Enthalpy transport (kW)	2.70	4.00	5.18	4.46	1.69	7.54
Axial conduction (kW)	0.94	0.82	0.82	3.08	3.45	12.0
HEATER						
Flow friction dissipation (kW)	1.07	1.34	1.73	1.69	1.72	1.55
Available energy loss (kW)	1.60	2.18	2.21	1.89	2.22	1.79
Mean gas-wall JT (K)	59	58	57	53	57	51

Displacer rod power is the power transmitted by the displacer rod required for overcoming mechanism losses outside the Stirling working gas; rod power = GLIMPS total PV power − GLIMPS piston PV power.

Appendix gap PV loss is an external calculation not included in GLIMPS.

Basic heat input to heater does not include parasitic losses. It does include regenerator axial conduction which was not included in table 2.

Available energy loss is a *Second-law* power loss due to irreversibility associated with gas-to-wall heat flow across finite ΔT. Roughly, the net work a Carnot engine could deliver in a reversible process with equivalent heat flows across the gas-to-wall ΔT.

Axial regenerator conduction includes gas molecular + eddy + solid-mode conduction.

Table 4: Performance of final designs predicted with GLIMPS version 2.0. Run 4A.2* is a rerun of case 4A.2 for comparison. Case 7.4 is the baseline.

6 Detailed Discussion of Components

Heater

The heater is arranged as a tube bundle, each tube having a hairpin bend at about the midpoint — tube count and length were held to reasonable values within the constraints of geometry and allotted volume. Scaling trends indicate that high pressure and moderate frequency were important in the success of the design.

Heater volume was carefully considered to avoid the problem of low swept/void volume ratio. As the tidal excursion in a heat exchanger is reduced, the distribution of heat flux becomes concentrated near the regenerator end, with adverse consequences on gas-wall temperature difference and efficiency. The distribution of heat flux along the heater length in the 3A.3 design point varies by only about $\pm 20\%$ as simulated by GLIMPS — which is good.

Regenerator

Although woven-screen regenerators are more widely used than foil regenerators the latter have some unique advantages. With the present lack of good empirical heat transfer and pressure drop data for screens, foil enjoys the luxury of exact solutions in the laminar flow regime of interest. Even for oscillating flow, laminar parallel flow between parallel plates is well understood [5]. Foil is inherently cheaper to manufacture than screens, requiring only a rolling process rather than a wire drawing process followed by weaving. Foil regenerators of 25.4 micron (1 mil) foil thickness have been successfully used in Stirling engines (Andy Ross); the spacing between layers being maintained by an array of raised dimples on the foil. With foil of this thinness it is possible to achieve surface-area-to-volume ratios comparable to the best screen regenerators.

About the only potential disadvantage to foil is its unbroken metallic conduction path from the hot end to the cold end. At least this conduction can be accurately calculated in contrast to the current state of affairs in screen regenerators where metallic conduction is known only approximately and enhanced gas conduction (due to microstructure of flow field) is potentially significant. In the various foil regenerators simulated in the 25 kWe solar engine, metallic conduction is acceptably low and enhanced gas conduction in laminar flow is not a problem.

Enhanced axial conductivity in flow through porous materials has been well documented in the chemical engineering literature. Molecular conduction is not really enhanced; the effect amounts to an accounting adjustment in one-dimensional models. In porous flow there are microscopic fluid eddies which, in the presence of an axial temperature gradient, exchange heat with each other in a sort of shuttle heat transfer mode. This results in an increased axial energy flow not accounted for by the enthalpy of the bulk flow. A correlation given in reference [2] shows that for Reynolds numbers R_e above about 10, the enhancement factor N_k is roughly $N_k \approx 1.3R_e$. With Reynolds numbers of 10–100 common in Stirling engines, axial conductivity enhancements are potentially serious losses.

There is always the potential for flow maldistribution in the regenerator — in this case the flow jets emerging from the adjacent heat exchanger tubes were thought to represent a potential problem. A jet is essentially a boundary-layer separation phenomenon that occurs when a fluid stream encounters an abrupt area increase — as when flow enters from a tube into the regenerator. For flow in the opposite direction no separation occurs. Therefore in the vicinity of tube inlets net flow circulations result over the course of a cycle.

An obvious question is: How do these flow circulations affect regenerator performance? An attempt was made to model flow jetting with the two-dimensional flow program (MANIFEST) under development by Gedeon Associates for NASA Lewis. Unfortunately MANIFEST was not able to model jets successfully, presumably as a result of the necessary viscous terms required for boundary layer separation effects having been ignored in its momentum equation. MANIFEST was shelved for the time being until funding becomes available to continue its development. Meanwhile the jet problem was addressed from a more simplified point of view.

Clearly the ratio of velocity head ($Q = \rho v^2/2$) in the heater and cooler tubes compared to pressure drop (ΔP) across the regenerator is important. A value of $Q/\Delta P = 1.0$ would clearly represent a problem while $Q/\Delta P = 0.01$ would probably be OK. For the typical design 3A.3 the ratio for both the cooler and heater turned out to be $Q/\Delta P = 0.1$, which is probably in the range where further analysis would be wise. It is possible that jet-induced flow circulations might have some detrimental effects near the entrances of the regenerator matrix.

One way to reduce the ratio $Q/\Delta P$ is to reduce Q, by incorporating tapered diffusing sections at the cooler and heater tube entrances adjacent the regenerator. The tapered diffuser feature is incorporated into the baseline design of the heater and cooler tubes by incorporating a 7.5 degree taper into the casting for the housing. The taper is sufficient to double the flow area of the heater and cooler tube exits, reducing $Q/\Delta P$ to .025 for design 3A.3, which should go a long way toward elimination of potential jetting problems.

Under ordinary conditions, the total included angle of a conical diffuser should be limited to about 8 degrees to avoid boundary- layer separation in the diffuser itself [7]. However, the presence of the regenerator matrix downstream of the diffuser will probably allow included angles significantly greater than 8 degrees. Successful flow-fillings of conical diffusers having included angles of 28 and even 90 degrees are reported in reference [9] by use of discrete screens within the diffuser, although a pressure drop penalty (on the order of Q) is paid. A general principle seems to be that complete flow- filling can be achieved at any diffuser angle by canceling the ideal static pressure recovery with appropriate flow resistance within the diffuser. After all, it is the adverse pressure gradient of decelerating flow that causes boundary-layer separation in the first place. Perhaps this idea is patentable in the context of Stirling engines? Anyway, if placing flow restrictions in the diffusers is too difficult, there are still beneficial effects to be had [9] by merely having the regenerator flow resistance at the extreme downstream end of the diffuser; although no quantitative guidelines have been located to date.

If further analysis indicates that there is indeed a jetting problem in the regenerator then it seems likely that the problem can be solved by flow diffusers. However, to obtain precise diffuser design information will require either a comprehensive literature search and some luck or some careful analysis. Perhaps the modeling of the diffuser-regenerator flow field might eventually fall within the capabilities of the MANIFEST computer program.

Cooler

The cooler is a tube-bundle design with the tubes in the form of a shallow U. Again, tube count and length are reasonable and the swept volume ratio low enough that the distribution of heat flux is uniform along the length within ±35% in design 3A.3.

At one point a finned annular cooler configuration was considered. In such a design the cooler is the finned inner surface of the pressure vessel and heat passes through the pressure vessel wall by conduction to the finned outer wall and eventually to the coolant. The main advantage of this design is mechanical integrity — the cooler can be a monolithic casting and the possibility of leaks at tube braze joints is eliminated. However, the Achilles heel proved to be the problem of getting the rejected heat through the pressure wall without excessive temperature drop and associated thermal stresses. As it turned out it could not be done within the constraints of the high power density solar engine design.

Displacer Gap

There is great confusion in the Stirling literature regarding the losses in the displacer gap, however, they can be broadly classified as thermal (heat-flow) losses and PV power loss. Heat-flow losses were approximated using the shuttle-heat-transfer formula given by Rios in reference [8], and the PV loss using a simple formula derived by Berchowitz in reference [1]. A proper analysis of the losses in the displacer gap is even more complicated than the analysis for the main Stirling cycle itself. Most recently, Huang and Berggren [6] have developed a computational solution to the appendix problem, however, no analytic formulae are available from that work and besides, they seem to lump thermal and PV losses together. At any rate, all displacer gap analyses start out by assuming the presence of a linear temperature gradient along the length of the displacer and that the displacer has a seal element at the cold end of the gap.

In the shuttle mechanism heat is carried by the moving displacer and alternately picked-up and deposited to the cylinder wall across the gap. So long as the longitudinal temperature gradient can be maintained the shuttle loss varies inversely with the gap. Shuttle heat transfer is relatively easy to estimate and the following formula was used.

$$Q_{shuttle} = \frac{\pi D k_g}{2L\delta} \frac{1+\lambda}{1+\lambda^2} X_d^2 \Delta T$$

where

D	=	Displacer diameter
k_g	=	gas conductivity
k_s	=	solid conductivity
L	=	Displacer length
X_d	=	Displacer amplitude (1/2 stroke)
δ	=	Radial gap
ΔT	=	Temperature difference
λ	=	$1 + (k_g/k_s)\sqrt{\alpha_s/(\omega\delta^2)}$
α_s	=	$k_s/(\rho c)$; Solid thermal diffusivity
ω	=	angular frequency (rad/s)

For the solar engine the λ parameter is very close to 1 which simplifies the loss calculation.

The appendix PV power loss is due to a sort-of minature Stirling cycle in opposition to the main cycle. The motion of the displacer along the cylinder causes the gas in the gap to alternately heat-up and cool-down roughly 180 degrees out of phase with the main Stirling cycle. Meanwhile, a compression and expansion of the gas in the gap occurs because of the pressure wave imposed by the expansion space. The result is a small heat pumping cycle in the gap which tends to pump heat from the cold to the hot end (in opposition to the shuttle loss) and develops a negative PV power. Berchowitz [1] derived the following formula for the PV power loss, however, the great number of simplifying assumptions used suggest that it is valid only for an order of magnitude estimate.

$$\text{PV Loss} = H - P_{seal}$$

where

$$P_{seal} = 1/2(\pi D\delta)P_{amp}(\omega X_d) sin(\phi)$$
$$H = -1/2(C_p M \omega r X_d T_x)(1/\pi - \cos(\phi_m)/4)$$

and

C_p	=	Gas specific heat
M	=	Mean gas mass in appendix
P_{amp}	=	Expansion space pressure amplitude
r	=	$\sqrt{(r_p\cos(\phi) - r_t)^2 + (r_p\sin(\phi))^2}$
r_p	=	P_{amp} / mean pressure
r_t	=	$X_d/(2L)$
T_x	=	Temperature gradient in appendix
ϕ	=	Expansion space pressure phase $-$ X_d phase
ϕ_m	=	$\arctan(\frac{r_s\cos(\phi)-r_t}{r_p\sin(\phi)})$

The results of Tables 1 and 2 reflect the choice of an optimal displacer gap as far as shuttle heat transfer and the appendix loss go. A small computer program was written to calculate the shuttle loss and appendix PV power loss over a range of gaps. The overall effect on efficiency was noted and the optimal gap selected. In all cases a displacer active length of 150mm (6 in.) was assumed. It turned out that the optimal gap was always about $0.001D$.

7 Detailed Discussion of Scaling

The scaling program used for the solar design is a general purpose tool which can be customized to a particular application. Essentially, the scaling program allows the user to define an arbitrary number N of scaling variables that correspond to scaling ratios for the basic Stirling variables of interest. The user then defines N arbitrary equality constraints which are functions of the scaling variables. Each constraint C is restricted to the log-linear form ($\log(C)$ is linear in the variables $log(X_i)$)

$$C = X_1^{\gamma_1} X_2^{\gamma_2} \dots X_N^{\gamma_N}$$

where the X_i represent the scaling variables and the γ_i arbitrary exponents. Scaling is accomplished by solving for the scaling variables that simultaneously satisfy all equality constraints.

For the case of the solar engine the scaling variables were the ratios of the following parameters.

Ve	=	Expansion space volume
P	=	Pressure
F	=	Frequency
Te	=	Hot temperature
τ	=	Cold/hot temperature ratio
κ	=	Compression/expansion space swept volume ratio
α	=	Expansion – compression space phase angle
σ	=	Expansion space/unswept volume ratio
Ar	=	Regenerator flow area
Gr	=	Regenerator foil gap
Lr	=	Regenerator length
α_r	=	Regenerator fill factor
Nc	=	Cooler tube number
Dc	=	Cooler tube diameter
Lc	=	Cooler length
Nh	=	Heater tube number
Dh	=	Heater tube diameter
Lh	=	Heater length

and the scaling constraints were ratios of

Ve	=	Expansion space volume
P	=	Pressure
F	=	Frequency
Te	=	Hot Temperature
τ	=	Cold/hot temperature ratio
κ	=	Compression/expansion space swept volume ratio
α	=	Expansion - compression space phase angle
σ	=	Expansion space/unswept volume
W	=	Power
VrVe	=	Regenerator volume/Ve ratio
NTUr	=	Regenerator NTU
DPr	=	Regenerator flow dissipation/Power ratio
α_r	=	Regenerator solid/gas heat capacity ratio
VcVe	=	Cooler volume/Ve ratio
NTUc	=	Cooler NTU
DPc	=	Cooler flow dissipation/Power ratio
VhVe	=	Heater volume/Ve ratio
NTUh	=	Heater NTU
DPh	=	Heater flow dissipation/Power ratio

Note that the variables and constraints are grouped into the categories of (1) general Stirling, (2) regenerator, (3) cooler and (4) heater. It turns out that due to the simplified form used for the constraints that each grouping forms an independent scaling sub-problem – that is it can be solved independently of the other groupings.

The general Stirling category is somewhat special. Because of the design guidelines provided by STC all of the variables in the general Stirling group appear again as constraints in the *constraints* list. An extra constraint *power* also appears. Since there are only 8 general Stirling variables, only 8 of the 9 constraints were used at any one time, although the selection varied depending on the needs of the problem under consideration.

Scaling Trends

Scaling trends were established to quantify the effects on scaling variables of independent variations in scaling constraints. These trends help to answer questions like: What happens to heat exchanger tube count and diameter if pressure is increased? Although the trends are not universally valid, they are probably relevant for any Stirling engines having turbulent-flow tubular heat exchangers, laminar flow foil-type regenerators, and similar volume ratios and displacer phasing to the STC solar engine.

	Pressure	Frequency	Vdead	NTU	Flow loss
Ve	0.91	0.91	1.08	1.0	1.0
P	1.10	1.0	1.0	1.0	1.0
F	1.0	1.10	1.0	1.0	1.0
Te	1.0	1.0	1.0	1.0	1.0
τ	1.0	1.0	1.0	1.0	1.0
κ	1.0	1.0	1.0	1.0	1.0
α	1.0	1.0	1.0	1.0	1.0
σ	1.0	1.0	1.10	1.0	1.0
Ar	0.91	1.0	1.12	1.05	0.95
Gr	0.95	0.95	1.05	0.95	1.0
Lr	1.0	0.91	1.06	0.95	1.05
α_r	1.10	1.0	0.91	1.0	1.0
Nc	0.93	1.18	1.03	1.32	0.89
Dc	0.99	0.92	1.05	0.89	1.03
Lc	1.00	0.91	1.05	0.95	1.05
Nh	0.93	1.18	0.02	1.32	0.89
Dh	0.99	0.92	1.05	0.89	1.03
Lh	1.00	0.91	1.05	0.95	1.05

Table 5: Scaling ratios for isolated multiplication by 1.1 (10% increase) of: (1) pressure, (2) frequency, (3) heat exchanger and regenerator volume/Ve ratios, (4) heat exchanger and regenerator NTU and (5) heat exchanger and regenerator flow friction dissipation. Power is constant in all cases.

Table 5 below gives some important scaling trends in a condensed format. The body of the table contains scaling ratios for the indicated variables. Each column pertains to the isolated variation of some constraint(s) by a scaling factor of 1.1 – all other constraints are constant (multiplied by 1.0). In all cases Ve of the general Stirling category is allowed to float (not constrained). Column 1, for example, answers the question: What happens to the scaling variables when pressure is multiplied by a factor of 1.1 (increase by 10%).

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[3] D. Gedeon, *A Globally-Implicit Stirling Cycle Simulation*, Proc. 21st IECEC, American Chemical Society, pp. 550–554, 1986.

[4] D. Gedeon, *Scaling Rules for Stirling Engines*, Proc. 16th IECEC, pp. 1929–1935, 1981.

[5] D. Gedeon, *Mean-Parameter Modeling of Oscillating Flow*, ASME Journal of Heat Transfer, Vol. 108, pp. 513–518, Aug. 1986.

[6] S.C. Huang and R. Berggren, *Evalutaion of Stirling Engine Appendix Gap Losses*, Proc. 21st IECEC, pp. 562–567, 1986.

[7] R.H. Norris, F.F. Buckland, N.D. Fitzroy, *Fluid Flow Data Book*, General Electric Co., 1970.

[8] P.A. Rios, *An Approximate Solution to the Shuttle Heat-Transfer Losses in a Reciprocating Machine*, Journal of Engineering for Power, pp. 177–182, 1971.

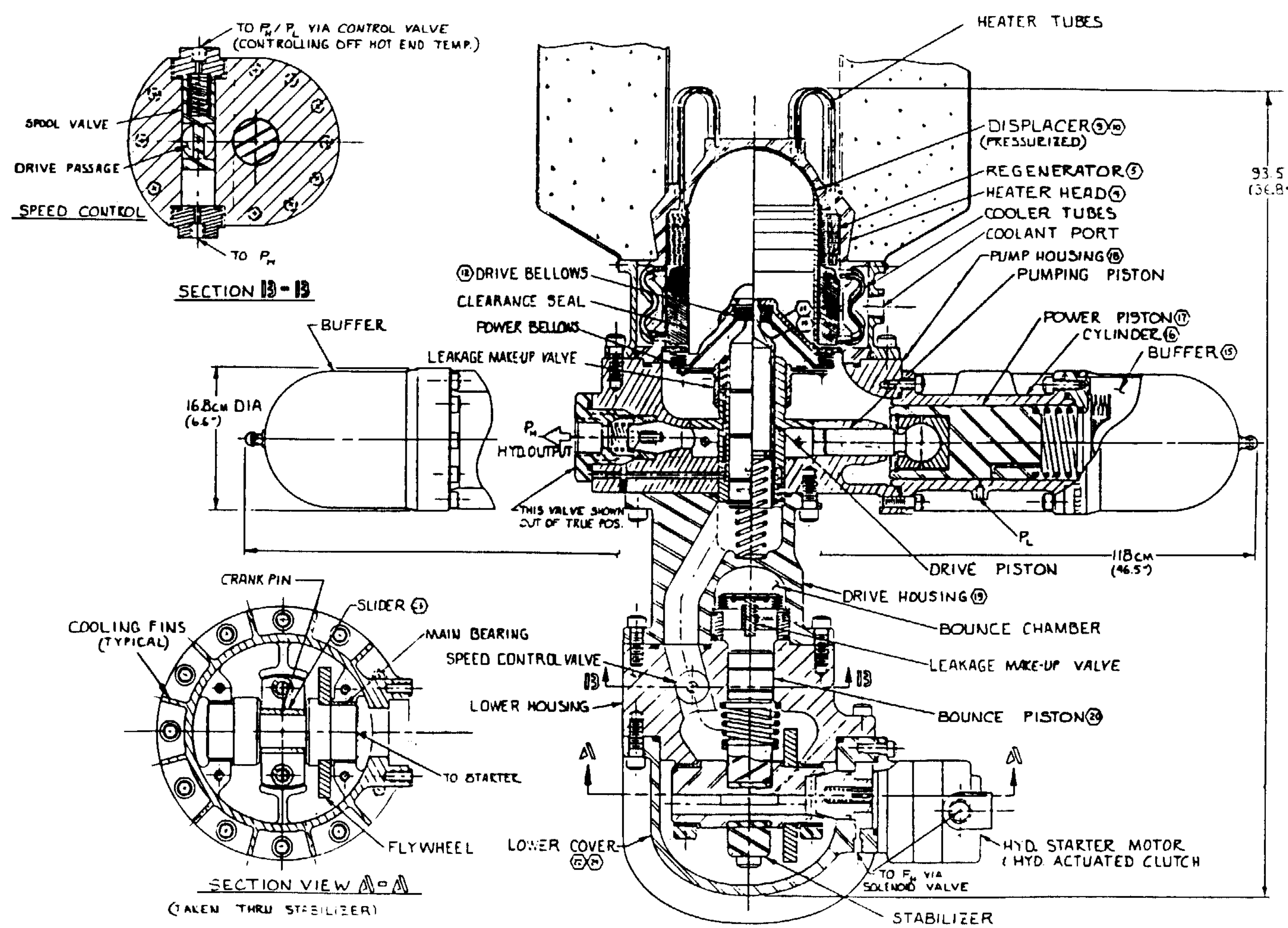
[9] G.B. Schubauer and W.G. Spangenberg, *Effect of Screens in Wide-Angle Diffusers*,NACA TN 1610, July 1948.

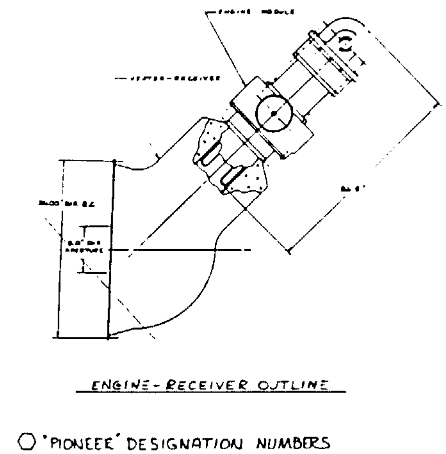
[10] J.R. Seume and T.W. Simon, *Oscillating Flow in Stirling Engine Heat Exchangers*, Proc. 21st IECEC, pp. 533–538, 1986.

Appendix F

DESIGN DETAILS

25 KW(E) SOLAR STIRLING HYDRAULIC ENGINE
CONCEPTUAL DESIGN LAYOUT





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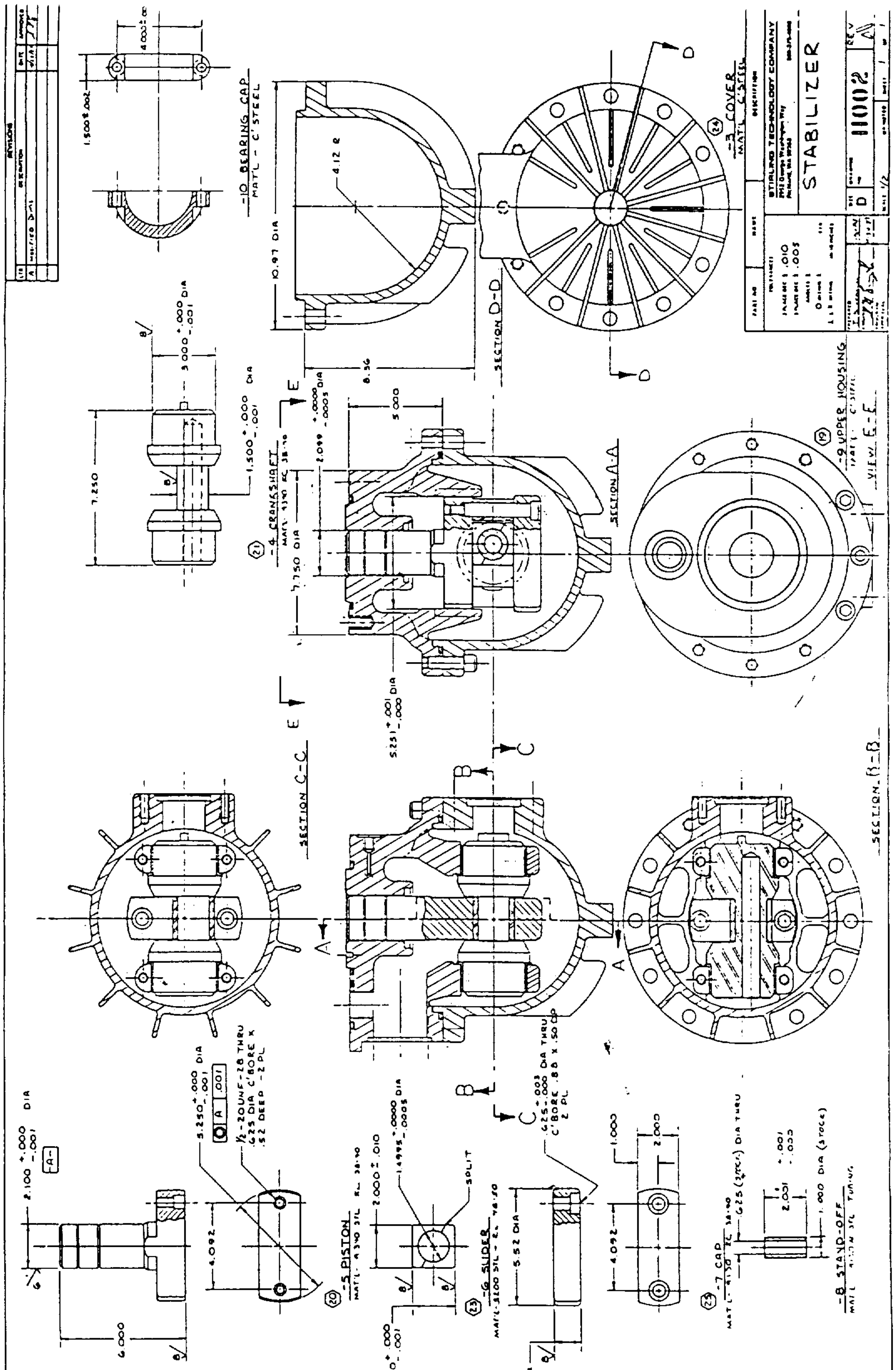

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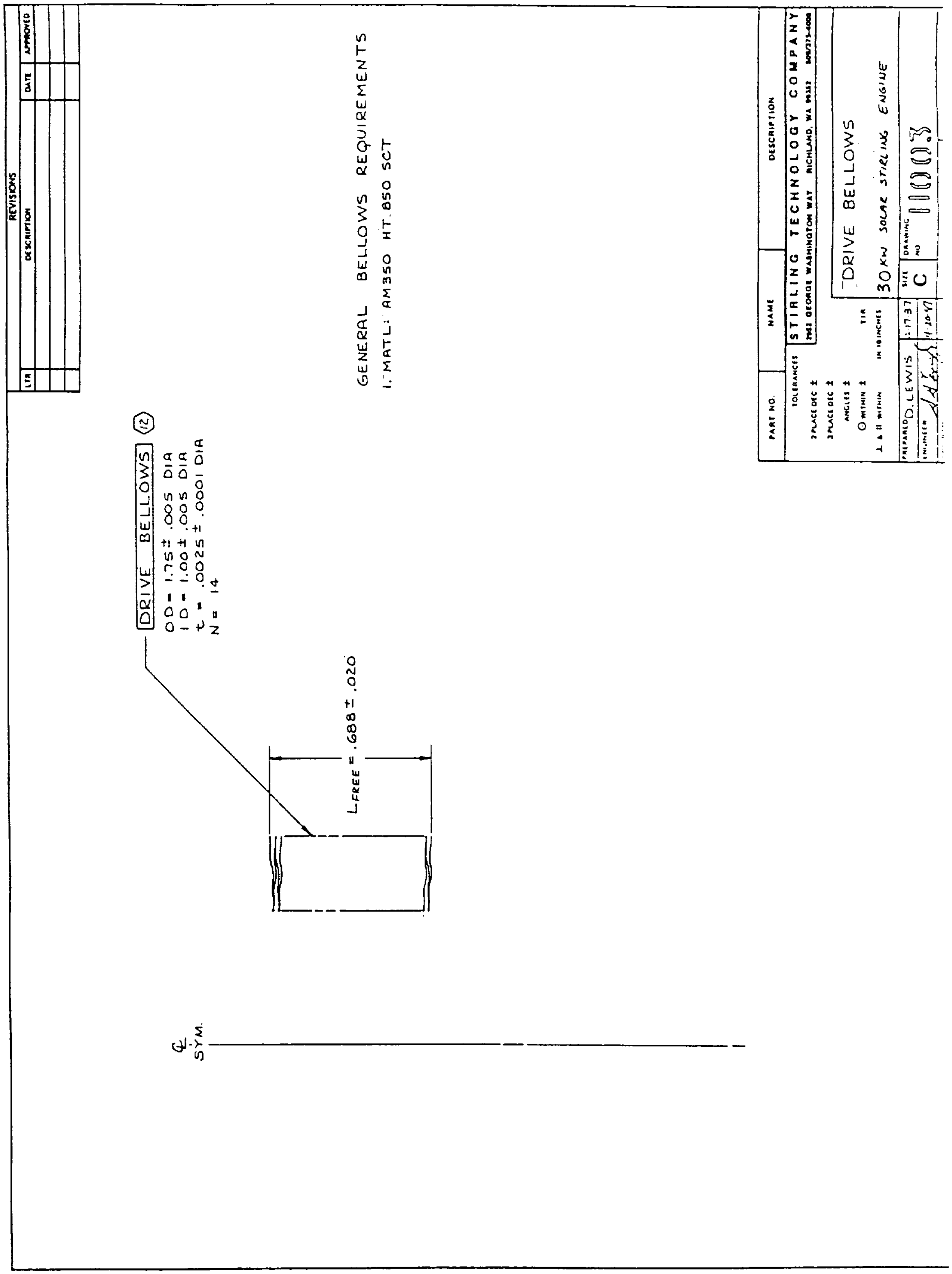

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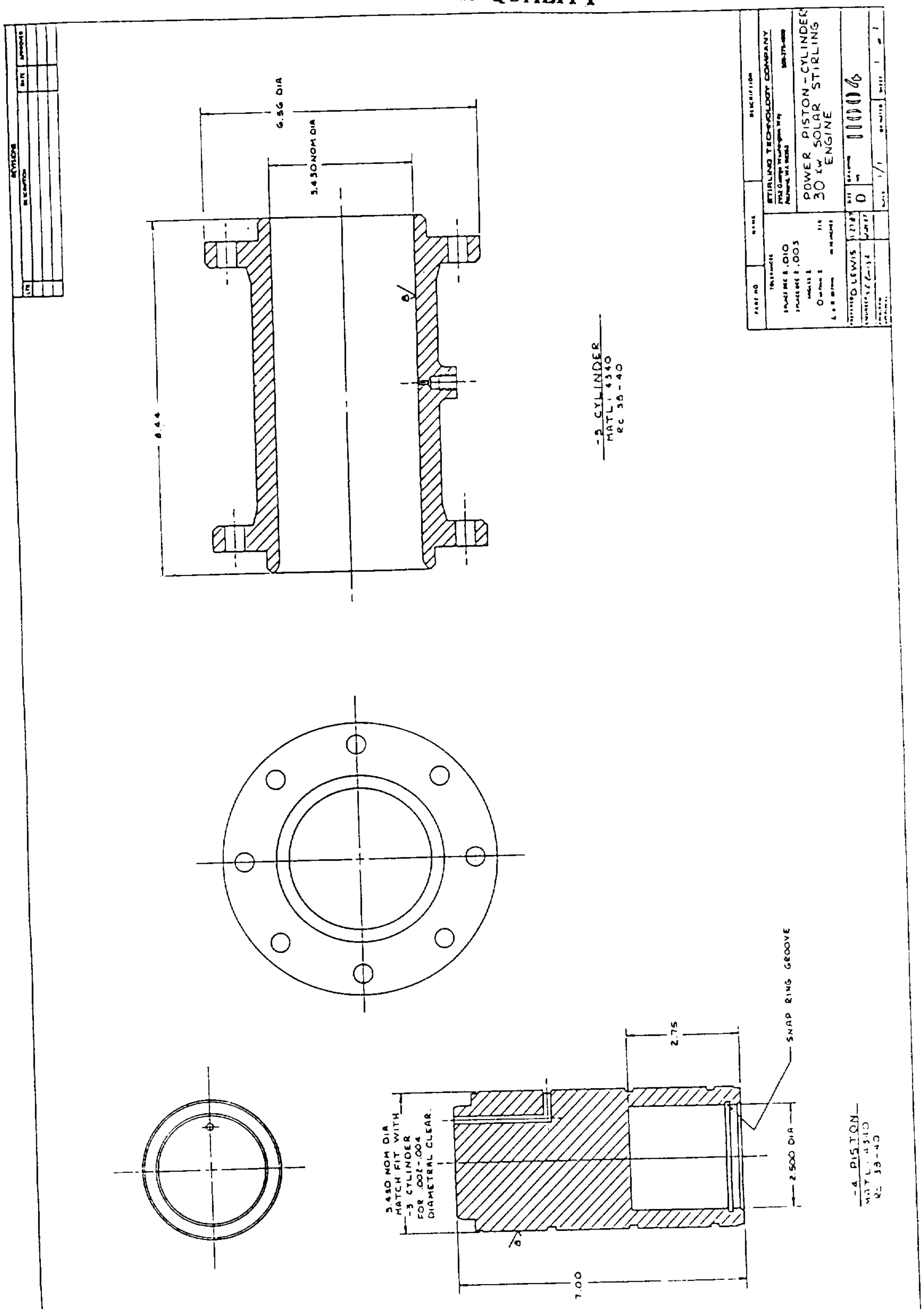
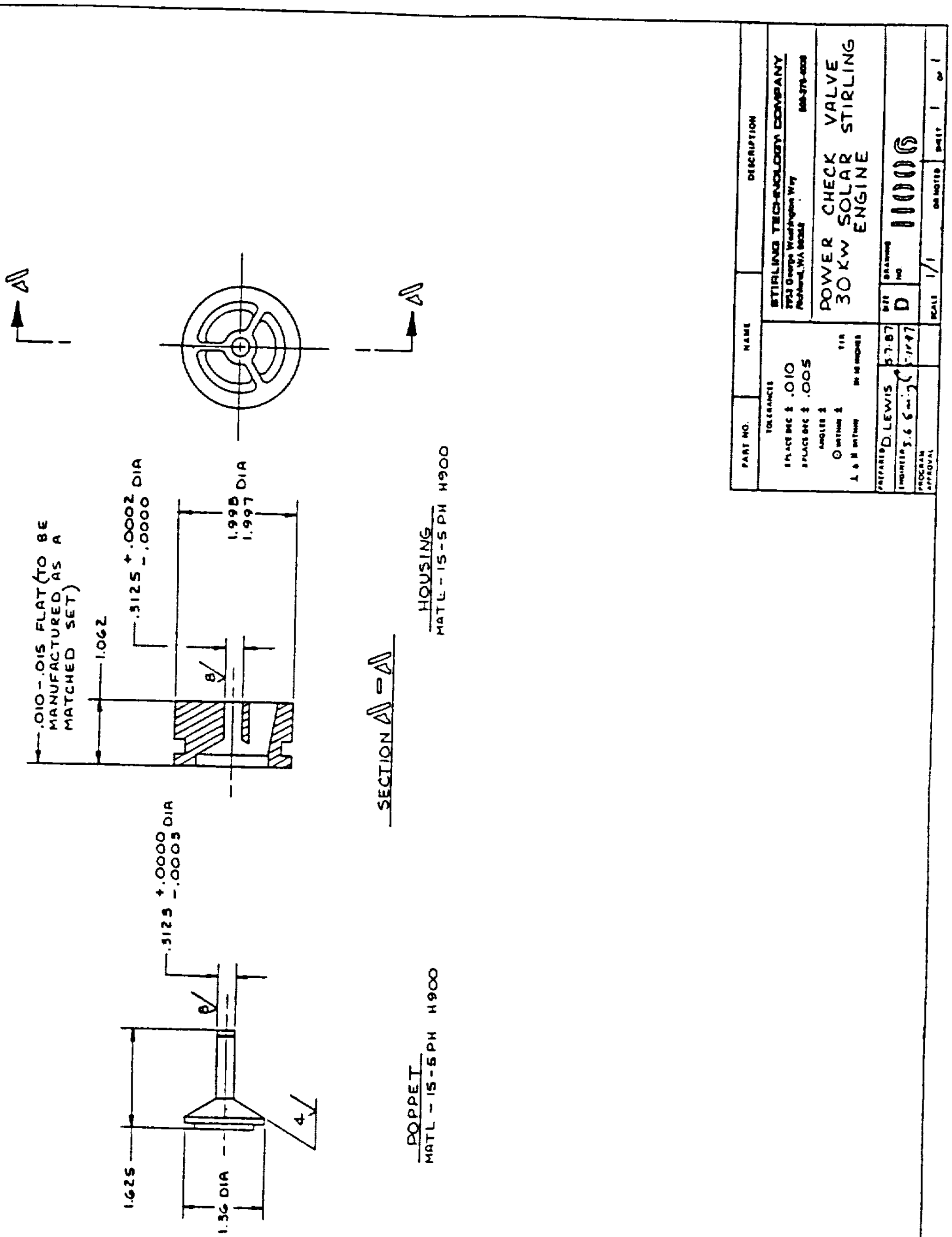


F-5



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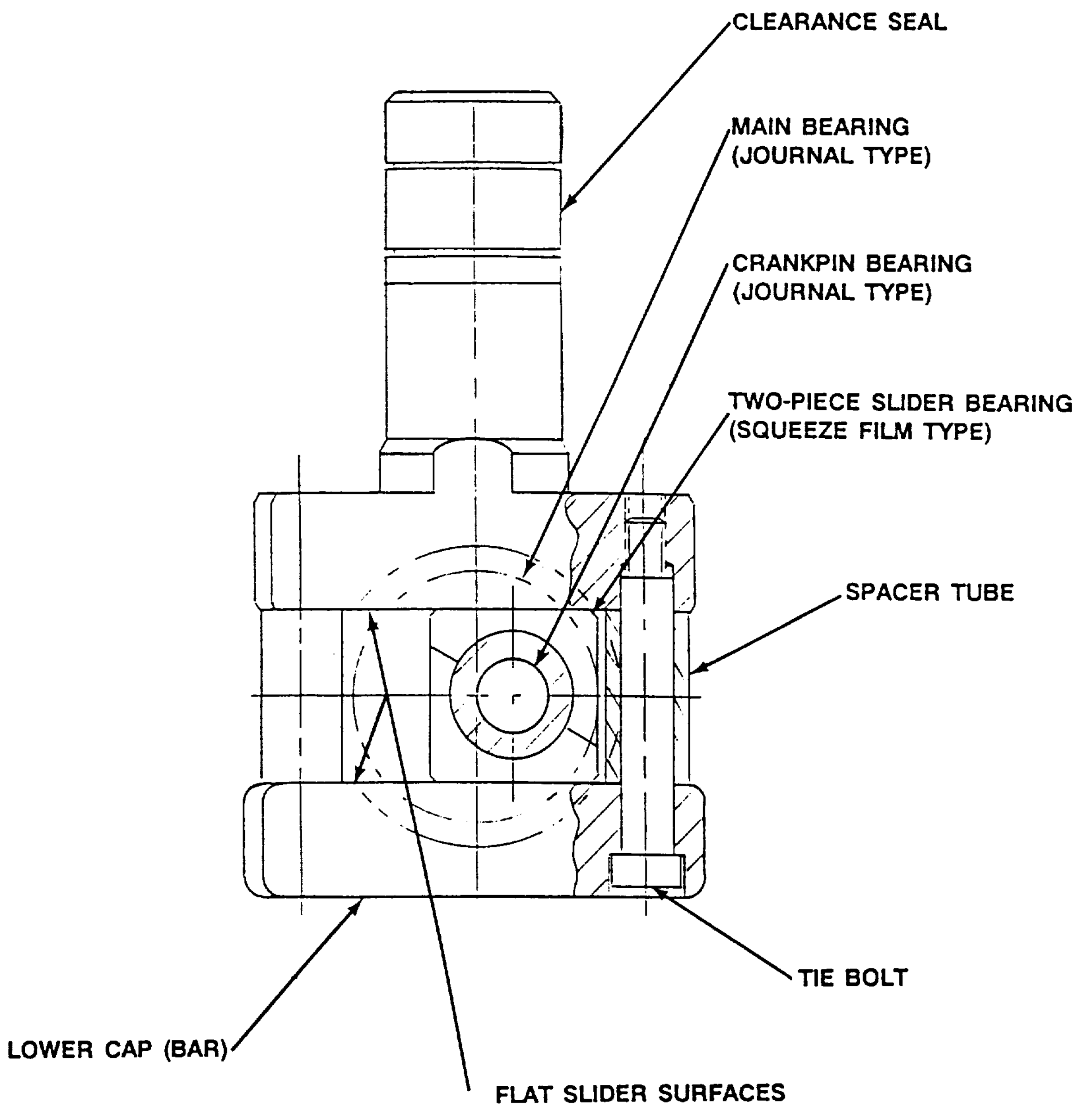
STABILIZER BEARING PRESSURES VS. CONVENTIONAL ENGINES, STRESS--MPa (PSI)

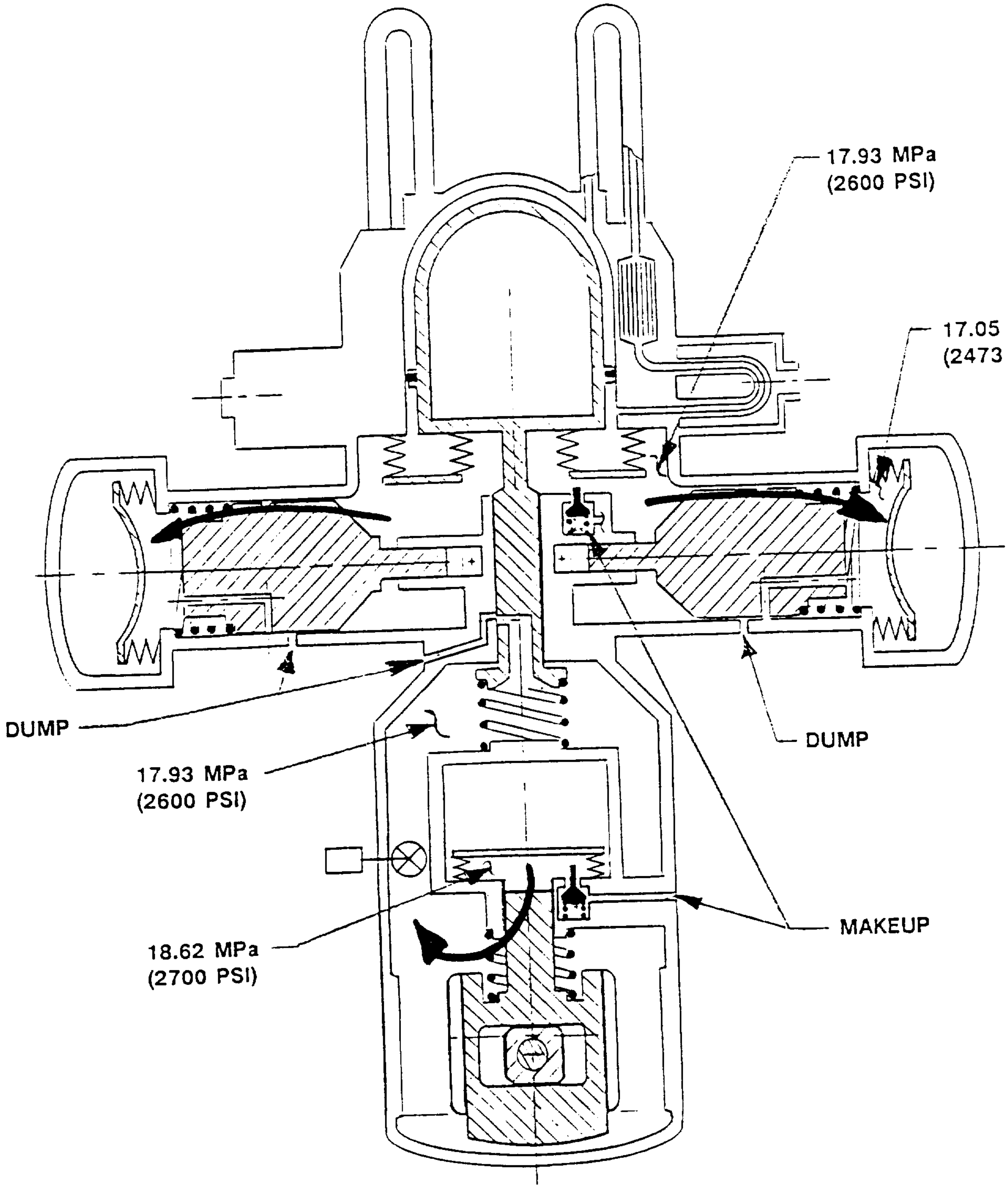
APPLICATION	MAIN	CRANK PIN	SLIDER/WRIST PIN
STC 1-2-397 ENGINE	2.28 (330)	3.79 (550)	2.84 (412)
AUTOMOTIVE ENGINES	7.93 (1150)	17.2 (2500)	25.2 (3650)
DIESEL ENGINES	6.55 (950)	11.0 (1600)	13.8 (2000)

STABILIZER COMPONENTS

4130 STEEL
WEIGHT = 7.04 KG (15.5 LB)



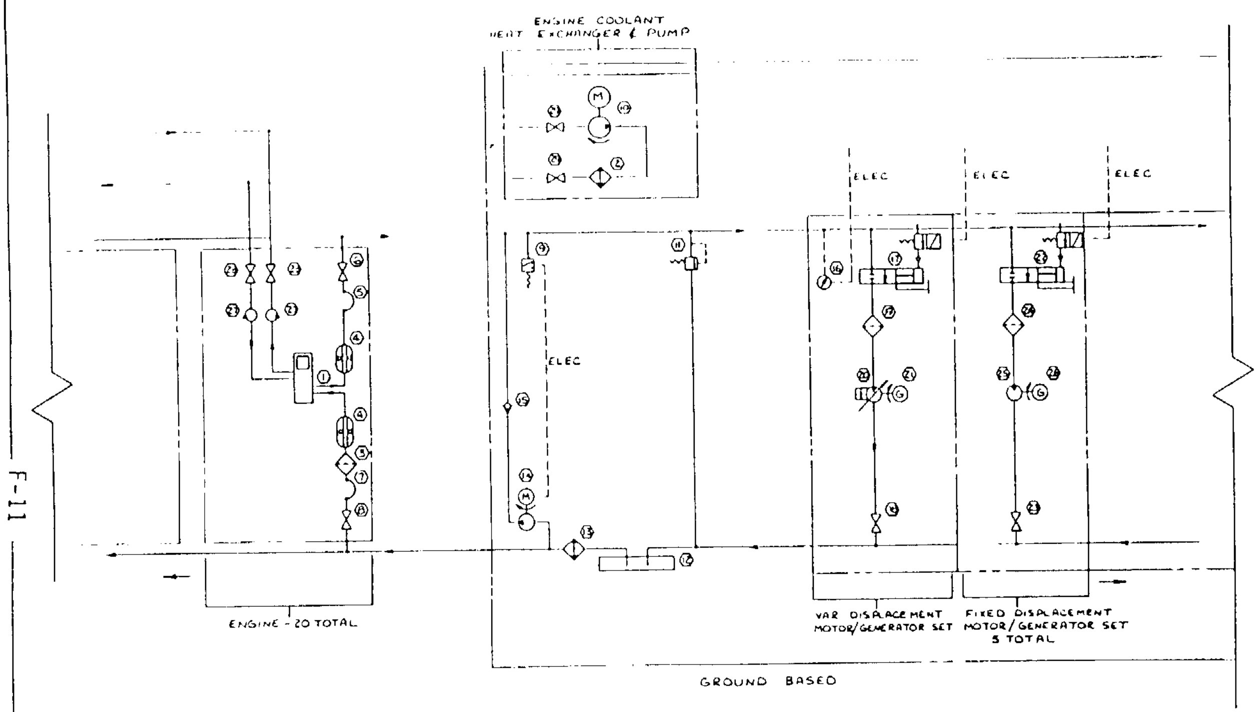
INTERNAL HYDRAULIC LEAKAGE MAKEUP



ENGINE MODULE WEIGHTS

NO.	COMPONENT	MATERIAL	PRELIM. WEIGHT		FINAL WEIGHT	
			KG	(LB.)	KG	(LB.)
1	HEATER TUBES	CG-27	1.27	(2.8)	.81	(1.8)
2	HEATER HEAD	XF-818	12.42	(27.4)	7.33	(16.2)
3	REGENERATOR/SPOOL	"300" ST. STL.	2.59	(5.7)	2.59	(5.7)
4	COOLER TUBES	LOW C. STL.	.23	(.5)	.30	(.7)
5	COOLER HOUSING	CAST STEEL	14.78	(32.6)	13.30	(29.3)
6	DISPLACER/DRIVE PISTON	INC 625/15-5PH	4.90	(10.8)	7.04	(15.5)
7	MOVING COLD PLATE	AM-350 ST. STL.	.36	(.8)	.36	(.8)
8	MAIN HOUSING	CAST STEEL	53.84	(118.7)	53.84	(118.7)
9	POWER PISTONS (2)	4340/1020 STL.	16.00	(35.2)	20.16	(44.4)
10	POWER HOUSINGS (2)	CAST STEEL	18.00	(39.6)	22.68	(49.9)
11	BUFFER ASSEMBLIES (2)	4130 STEEL	11.65	(25.7)	13.10	(28.8)
12	DRIVE HOUSING	CAST STEEL	16.43	(36.2)	12.32	(27.2)
13	LOWER HOUSING/COVER	CAST STEEL	27.83	(61.3)	27.83	(61.3)
14	SPEED CONTROL	15-5PH STL.	1.87	(4.1)	1.87	(4.1)
15	SCOTCH YOKE/BALANCE	4340 STL.	4.90	(10.8)	7.04	(15.5)
16	BOLTS	STEEL	8.01	(17.7)	10.41	(22.9)
17	SPRINGS	STEEL	1.48	(3.3)	1.92	(4.3)
18	FLYWHEEL	CAST STEEL	-	-	4.08	(9.0)
19	STARTER	STEEL/ALUMINUM	-	-	3.20	(7.0)
20	HYDRAULIC FLUID	MIL H-8446/ATF-F	-	-	3.22	(7.1)
21	PULSE DAMPERS (2)	STEEL/RUBBER	-	-	7.25	(16.0)
22	FILTER	ST. STL.	-	-	4.54	(10.0)
		ENGINE SUBTOTAL	196.56	(433.2)	225.19	(496.2)
23	SUPPORT CONE/RING	"300" ST. STL.	-	-	2.41	(5.3)
24	ENTRANCE CONE	"300" ST. STL.	-	-	6.36	(14.0)
25	ABSORBER DOME	"300" ST. STL.	-	-	5.13	(11.3)
26	AFT DOME	"300" ST. STL.	-	-	11.35	(25.0)
27	SUPPORT TUBE	"300" ST. STL.	-	-	10.21	(22.5)
28	INSULATION	CEREWOOL (8 PCF)	-	-	31.10	(68.5)
29	INSULATION SHELL	ALUMINUM	-	-	4.09	(9.0)
30	LIQUID METAL	SODIUM/POTASSIUM	36.32	(80.0)	24.52	(54.0)
		HEAT TRANSPORT SUBTOTAL	-	-	95.17	(209.6)
		GRAND TOTAL			320.36	(705.8)

REV	DESCRIPTION	DATE	APPROVED
A	SEE E.O.	7-1-87	DCL



QTY	ITEM NO.	NAME	DESCRIPTION
2	29	ISOLATION VALVE	MANUAL, 30 PSI, 320 GPM 8" UTILITIES SUPPLY CORP BEFF PVC 300
10	28	ISOLATION VALVE	MANUAL, 30 PSI, 16 GPM, 2 1/2" BALL, UTILITIES SUPPLY CORP, [illegible]
40	27	HOSE	30 PSI, 6.2 GPM, 2 1/2 ID AEROQUIP FC319-40X44"
5	26	INDUCTION GENERATOR	3 PHASE, 109 HP/81 KW, 1800 RPM, GE QUOTE #300-43-25.05 REV 0 ITEM 2
5	25	HYDRAULIC MOTOR	FIXED DISPLACEMENT, 3000 PSI 117 HP, VOLVO F11-150-HF-[illegible]
5	24	FILTER	5 MICRON, 3000 PSI, 72 GPM FAIREY 374A-3 35-TL305
5	23	ISOLATION VALVE	MANUAL, 60 PSI, 72 GAL/MIN BALL, WORCHESTER 2 1/2" 4546TSW
5	22	CONTROL & ISOLATION VALVE	AUTOMATIC, 3000 PSI, 72 GPM [illegible] WORCHESTER [illegible] ACTUATOR & MS032 MTN KIT
1	21	INDUCTION GENERATOR	3 PHASE, 168 HP/125 KW, 1800 RPM GE QUOTE #300-43-25.05 REV 1 ITEM 2
1	20	HYDRAULIC MOTOR	VAR DISPLACEMENT 3000 PSI [illegible] VOLVO [illegible]
1	19	FILTER	5 MICRON, 3000 PSI, 115 GAL/MIN FAIREY 374A-3 35-TL305
1	18	ISOLATION VALVE	MANUAL 60 PSI, 115 GAL/MIN BALL, 3" PIPE WORCHESTER 3" 4546TSW
1	17	CONTROL & ISOLATION VALVE	AUTOMATIC, 3000 PSI, 115 GAL/MIN WORCHESTER [illegible] ACTUATOR & MS032 MTN KIT
1	16	PRESSURE INDICATOR	3000 PSI ELECTRONIC OUTPUT
1	15	CHECK VALVE	3000 PSI, 2.3 GPM, 3/8 PIPE, REXROTH 5GA10/5
1	14	AUXILIARY PUMP AND MOTOR	3000 PSI, 2.3 GPM, VOLVO F11-5 5 HP, 3 PHASE, 1750 RPM, GE [illegible]
1	13	RADIATOR	21.5 KW, 460 GPM, 60 PSI 78.7 BTU/MIN °F, YOUNG OCS-2300
1	12	RESERVOIR	50 GAL, 60 PSI, 6" FLANGE INLET & OUTLET, SIGHTGAUGE, FILLPORT & DRAINPORT
1	11	PRESSURE RELIEF VALVE	OPEN @ 3300 PSI, 460 GPM REXROTH DB82F2-13/3X115U/5
1	10	COOLANT PUMP	124 GPM, 3 HP, 50 FT HD TEEL, GRAINGER 3PG05
1	9	PRESSURE SWITCH	CLOSES @ ≤2750 PSI, OPENS @ 2950 PSI, ASCO #SB11A/TNICA21
20	8	ISOLATION VALVE	MANUAL, 60 PSI, 23 GPM BALL 1 1/4" PIPE WORCHESTER 1 1/4" 446YBSW
20	7	HOSE	60 PSI, 23 GPM, 1 1/2 ID AEROQUIP FC316 [illegible]
20	6	ISOLATION VALVE	MANUAL, 3000 PSI, 23 GPM BALL 1" PIPE WORCHESTER 1" 446YBSW
20	5	HOSE	3000 PSI, 23 GPM, 1 1/4 ID X 8' AEROQUIP FC336-20 W/CRIMP FIT
40	4	PULSATION SUPPRESSOR	23 GPM, 50 HZ NACOL #3210-02
20	3	FILTER	10 MICRON, 23 GPM, 60 PSI FAIREY [illegible]
1	2	COOLANT RADIATOR	832 BTU/MIN °F YOUNG, #MWC256V30
20	1	ENGINE	STC 25 KW SOLAR STIRLING

COOLANT LOOP

PIPE SIZE (PVC SCH 80)	LENGTH (FT)
2 1/2"	2944
3"	448
4"	296

HYDRAULIC CIRCUIT

PIPE SIZE (STEEL)	LENGTH (FT)
1" SCH 80	1472
1 1/4" SCH 40	1472
1 1/4" SCH 80	224
2" SCH 40	224
2 1/2" SCH 160	148
3" SCH 40	148
5" SCH 160	30
6" SCH 40	30

STIRLING TECHNOLOGY COMPANY
2952 George Washington Way
Richland, WA 99352
509-375-4000

HYDRAULIC CIRCUIT
500 KW SOLAR ARRAY
NASA CONTRACT NO. DEN3-371

PREPARED D. LEWIS 5-29-87 | SIZE D | DRAWING NO. 11008 | REV A
SCALE NONE | SHEET 1 of 1

Appendix G

HEATER TUBE STRESS EVALUATION

Heater Tube Stress Analysis

The heater tubes are subjected to fluctuating internal pressure of 2,600 ± 397 psi. The candidate material is CG-27 with the following properties at 700°C.

σ_c = 45,000 psi 60,000 hr creep rupture strength

$E = 20.0 \times 10^6$ psi

A preliminary evaluation indicates the critical failure mode will be creep rupture. Fatigue does not appear to be critical although further evaluation should be made in the next design phase for low cycle fatigue in startup and shutdown, and high cycle fatigue in the normal operating mode. A first order stress analysis follows.

Hoop stress is given by: $\sigma_h = PD_m/2t$

where σ_h = hoop stress

P = Pressure

D_m = mean tube diameter

t = tube wall thickness

Applying this relationship and allowing 0.0015 inches per year material loss results in a hoop stress σ = 24,700 psi. This includes a 1.5 factor applied to the nominal operating pressure. The margin of safety is

M.S. = (45,000/24,700) - 1 = <u>0.82</u> (in addition to 1.5 load safety factor)

First cycle thermal stress is calculated as a measure of the severity of this type loading. While a rigorous analysis would include a consideration of ratcheting, it appears that sufficient relaxation will occur that thermal stress will not be an overriding problem, although some accounting of it should be made. The beneficial effect of relaxation is discussed further. First cycle thermal stress is calculated as follows.

The temperature gradient across the tube wall is given by

$$\Delta T = q_o \ln(D_o/D_i)/2\,\pi k$$

where q_o = radical heat flux/unit tube length

D_o and D_i = tube O.D. and I.D.

k = thermal conductivity

Stress is given by

$$\sigma_T = (1/2)\alpha \, \Delta TE$$

where α = coefficient of thermal expansion

E = modulus of elasticity

for q_0 = 504 BTU/hr-in

(263 in^2 tube area and 75 Kw heat load)

k = 1.05 BTU/hr °F in

A = 10.6 x 10^6 in/in °F

σ_T = 2,140 psi

These stress levels are considered satisfactory for initial sizing. An estimate was made of the rate of relaxation of heater tube thermal stress over the 60,000 hour life. The approach is very approximate in that relaxation of only the outer fiber was considered. The restraining effect of adjacent inner fibers, stressed less highly and thus relaxing at a slower rate, was not considered.

The initial stress differential assumed to be 7,600 psi with a corresponding .038% strain. Creep rate was from Figure A-7 of Appendix A, for CG-27. Relaxation was allowed in tenative steps of 10,000 hours. The tenative process is tabulated in Table G-1. with dimensional and stress relaxation plotted in Figure G-1.

A true assessment of the effect of thermal gradient on the tube stress levels is a very complex problem complicated further by 20,000 cycles of startup and shutdown.

During steady state operation the tube experiences a fluctuating pressure stress. The OD of the tube is at a higher temperature than the ID causing a thermal stress gradient of tension at the ID and compression at the OD. The relaxation throughout the tube wall is not uniform because of the varying stress level, which is dependent not only upon the thermally induced stress but also, and probably mainly, on the pressure stress. The restraint and interaction between successive fibers is not easily determined.

A further complication comes with startup and shutdown. On cooling, the tube OD (which had thermally induced compressive stress components at temperature) will go into tension. If stress concentration or notch sensitivity develops due to intergranular corrosion, low cycle fatigue could become a problem with startup and shutdown cycling. This question should be addressed in follow up work.

Table G-1

STRESS RELAXATION ANALYSIS

TIME INCREMENT (10,000 HRS.)	% RELAXATION	RESIDUAL STRAIN	STRESS RELIEF	RESIDUAL STRESS	HOURS
0	-	.0380	-	7,600	0
1	.013	.0250	2,600	5,000	10,000
2	.004	.0210	800	4,200	20,000
3	.0025	.0185	500	3,700	30,000
4	.002	.0165	400	3,300	40,000
5	.0017	.0148	340	2,960	50,000
6	.0015	.0133	300	2,660	60,000

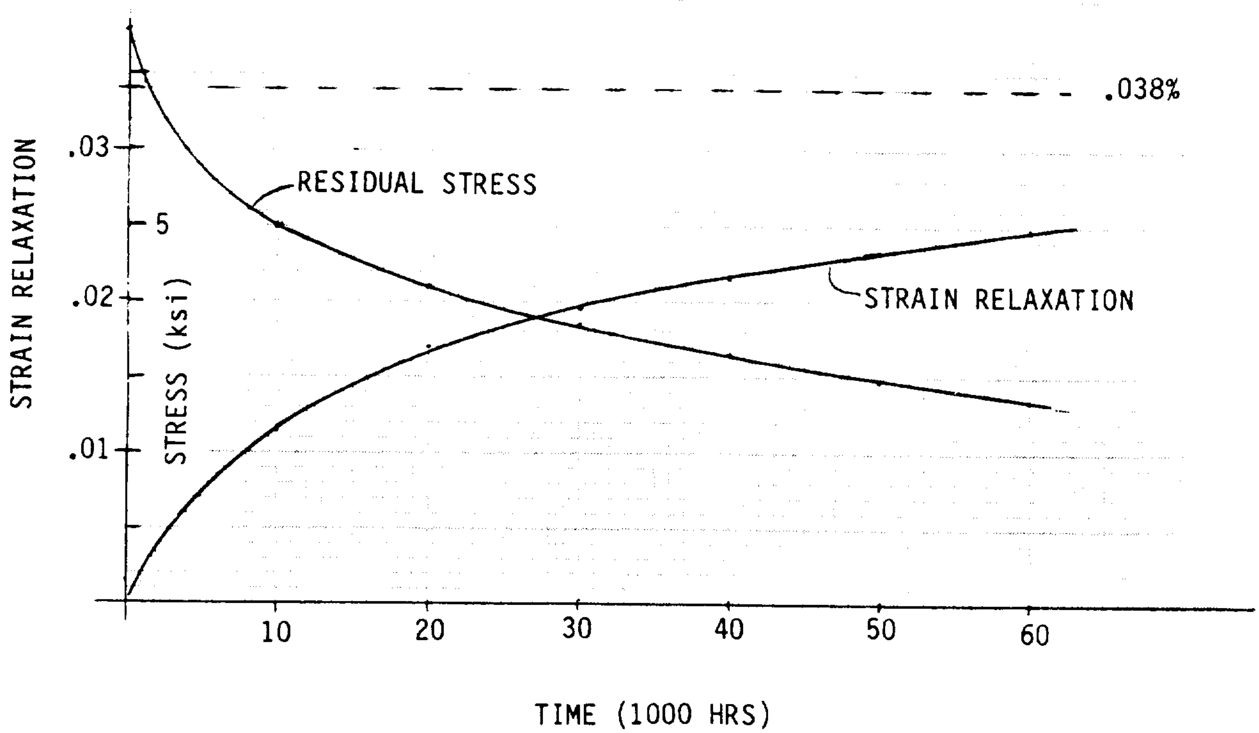


Figure G-1. Stress and Strain Relaxation

Appendix H

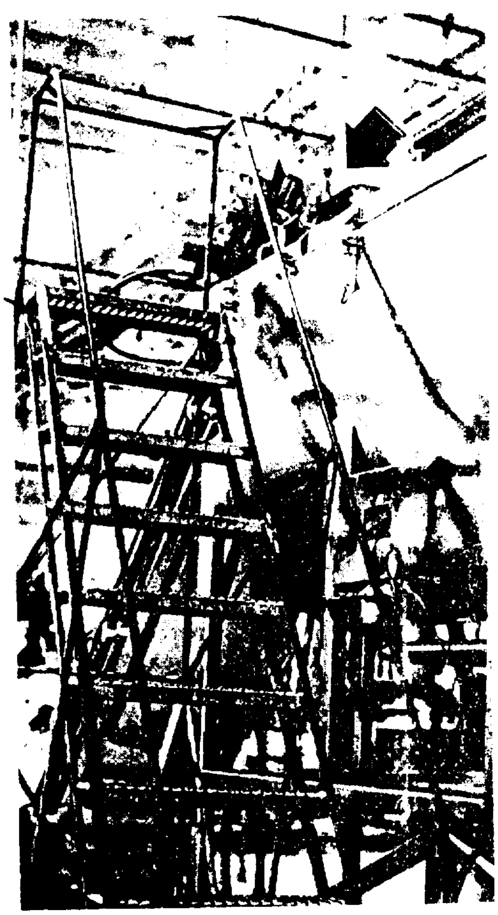
HYDRAULIC MOTOR RELIABILITY

Plant uses 53 hydraulic motors—no motor downtime in 2½ years

ROBERT LUKARIS, General Manager
Cut & Ready Foods

KARL ROBE,
Associate Editor

SANITATION & MAINTENANCE



Small hydraulic motor (arrow), one of 53 at Cut and Ready Foods, runs vertical elevator. Similar motor and drive operates blancher.

NEW SOLUTIONS OF PLANT PROBLEMS

PROBLEM: Frequent washdowns and spray cleaning of product and equipment in a potato processing plant make it a difficult environment for electrical equipment. Shorting and shock hazards require extra precautions.

Cut and ready Foods, subsidiary of Del Monte Corp., wanted to minimize shock hazards as well as downtime from motor failure when building their plant at San Lorenzo, Calif.

SOLUTION: The 53 motors driving conveyors, elevators, sorting belts, and processing equipment are hydraulic. They are driven by hydraulic fluid continuously circulated from a central pumping station in the engine room. Electrical components are thus isolated from processing areas.

Five 50 hp electric motors drive a series of variable volume, constant pressure pumps (see diagram and photo). Pumps are vane type with a "walking ring" that revolves the wear surface during use to distribute vane wear equally, and thus extend pump life between overhauls. A separate in-line filter continuously removes particles larger than 10 microns.

Three pipelines and manifolds deliver fluid to motors in processing room. There are two return lines. Each small hydraulic

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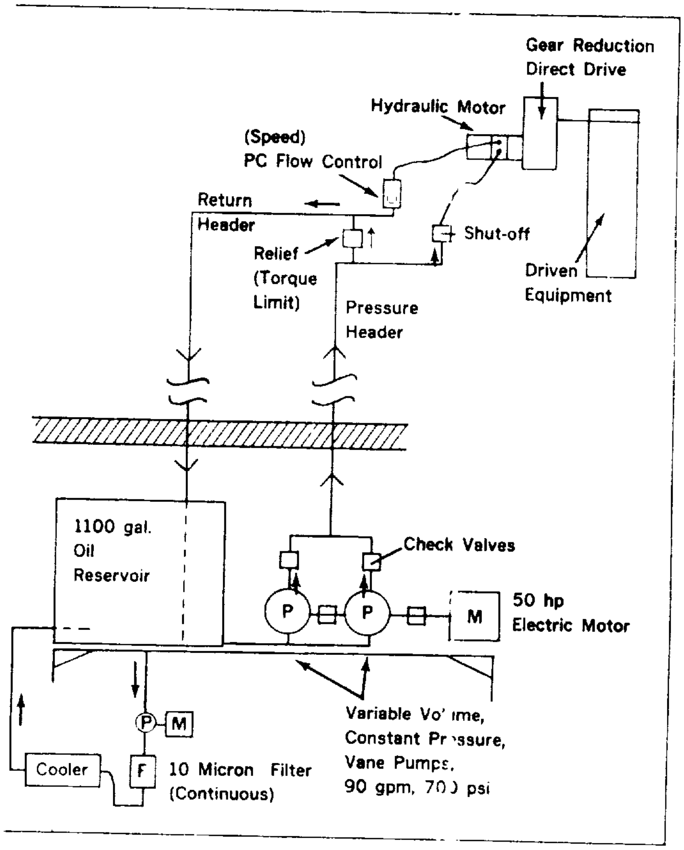

Typical hydraulic motor driven equipment layout. No chains or sprockets to adjust, lubricate, or protect

motor and rugged gear reduction drive unit mounts directly to shaft of driven equipment.

Rotary hydraulic motors are a unique departure from conventional piston-and-cylinder design. Moving fluid drives an orbiting gerotor mechanism which converts fluid power to rotary power.

Several inherent hydraulic features provide built-in control that would require separate systems in electrical or mechanical designs. For instance, variable speeds at high torque are achieved simply by adjusting a pressure compensating flow control valve in the hydraulic line to or from each individual motor. Control valves are self-contained, pressure compensated flow control units which vary inflow (or outflow) to preset rate regardless of change in line pressure. Valves require no solenoids, wiring, auxiliary air or pneumatics. Speed is adjusted simply by turning knurled knob.

Cushioned acceleration and deceleration are inherent in a properly designed hydraulic system. When fluid flow is shut off to stop equipment, back pressure dampens forward motion of gear motor, effectively smoothing deceleration.

When equipment jams, hydraulic motor stalls against a greater than normal load or countertorque. By placing a pressure relief valve in the line, flowing fluid automatically bypasses motor when countertorque gets too high. This inherent torque limiting prevents damage to motor and gears automatically in the Cut and Ready hydraulic system.

Ten pumps in central engine room have a unique, built-in, self-regulating control design that also reduces need for elaborate accessory controls.

RESULTS: There has not been a production shutdown due to motor failure in over 2½ years. Previously, several electric motors would fail each year, causing downtime and replacement expense running into thousands of dollars. (Because this was a new plant, comparison cannot be made with operation at old site.)

Automatic torque limiting has saved many hours formerly spent clearing jammed equipment. If a machine or conveyor jams now, it does no further harm because motor stops automatically. Operator then shuts off oil flow, or he can reverse it to free jam. In most cases, no other attention is needed.

When a jam progresses gradually, as when a conveyor belt starts to get out of alignment, additional friction slows motor, giving operator advance warning of failure. Motors themselves give advance warning of impending failure by slowing down. This allows motor to be removed during non-operating hours for repair, without lost production time.

Only major preventive maintenance needed is to check condition of oil filter every 6 months and to add replacement oil to reservoir. Yearly lab tests show no degradation of hydraulic oil, which has not yet required replacement.

Equipment cleaning is now safer, simpler and faster. Motors can be hosed directly without special precautions against shock or motor damage. Electrical components are housed in separate room, tended by experienced personnel.

Cost of basic hydraulic motor system plus installation is roughly same as one using comparable, totally enclosed, fan-cooled drive, fixed speed electric motors. Cost will be slightly higher than system using standard open, drip-proof, fixed speed electric motor; and up to 20-30% less than electric systems using more elaborate variable drive, gear-head motors with dynamic braking and torque limiting options, according to survey made by Cut and Ready about three years ago.

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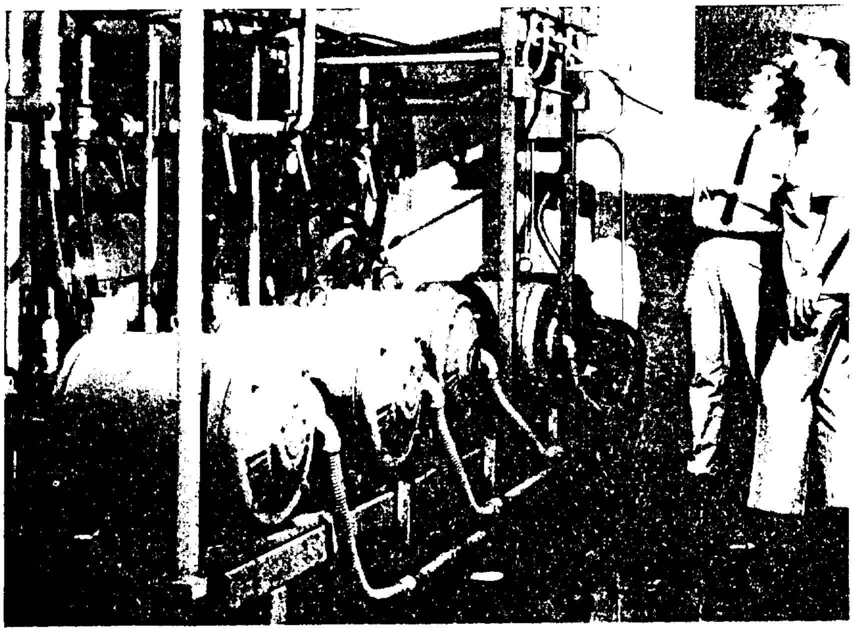


Variable volume pumps, electric motors, and oil reservoir are in separate room

Char-Lynn Orbit® high torque rotary hydraulic motors are manufactured by Eaton Corp., Fluid Power Div., 15151 Highway 5, Eden Prairie, Minn. 55343 and are described in Cat 11-811.

Circle 277 opposite last page.

Hydraulic drive system and gear reduction units were designed and installed by PTE Corp., 1345 N. 10th St., San Jose, Calif. 95112.

Circle 278 opposite last page.

Variable volume, pressure compensating vane pumps with "walking ring" vane housing are products of Continental Hydraulics, Savage, Minn. 55378. Bul HD-264 describes pumps.

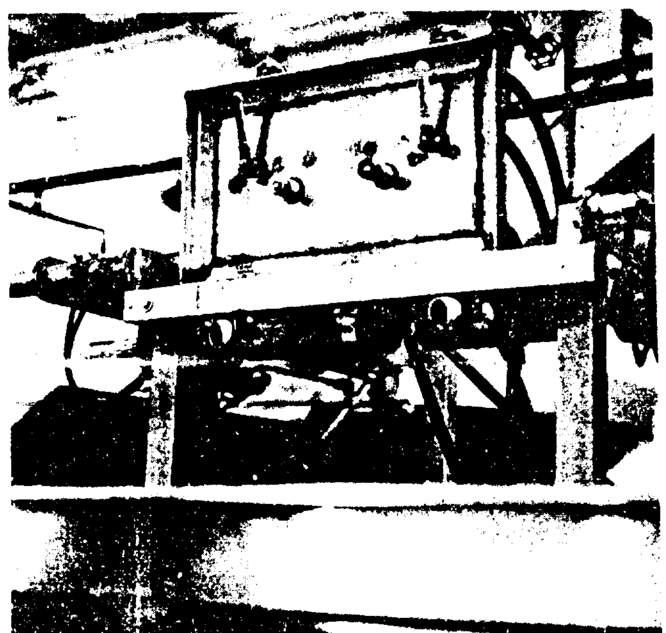
Circle 279 opposite last page.

Notes on hydraulic motor systems

When high surge loads are encountered, with peaks 20 per cent above normal load, a stall condition results. Torque limiting feature has advantages mentioned in article. When stall occurs and conveyor or equipment stops, incoming product continues to pile up until infeed conveyors and equipment are stopped. If this is regular problem, overload sensors should be installed. Stalls have not been much of a problem at Cut and Ready.

A central hydraulic pumping system is not economical if used to replace an existing electrical system and motors. In existing plants, if hydraulics are used to replace outmoded electrical systems, or in critical areas, individual pumping systems should be installed close to hydraulic motors.

Range of equipment on which supplier of hydraulic system at Cut and Ready has applied hydraulic motor drives is from one rev/hr to 1800 rpm, with majority up to 1800 rpm. Above 1800 rpm, larger diam oil lines or higher oil pressure (or both) are needed to get adequate amount of oil through motor to maintain high torques



Two motors on silver remover machines withstand regular washdowns with high pressure water

Appendix I

COST ANALYSIS FOR STIRLING TECHNOLOGY 25 KW SOLAR DRIVE ELECTRICAL GENERATOR AS FORECAST UTILIZING PARETO'S LAW

FINAL REPORT

OCTOBER 1987

PREPARED BY

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R. Osen
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NASA LEWIS DISCLAIMER

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REPORT DOCUMENTATION PAGE	1. REPORT NO.	2.	3. Recipient's Accession No.
4. Title and Subtitle COST ANALYSIS FOR STIRLING TECHNOLOGY 25 KW SOLAR DRIVE ELECTRICAL GENERATOR AS FORECAST UTILIZING PARETO'S LAW			5. Report Date SEPTEMBER 1987
			6.
7. Author(s) M. STEWART, W. JACKSON, R. OSEN, R. HEITSCH			8. Performing Organization Rept. No.
9. Performing Organization Name and Address PIONEER ENGINEERING & MANUFACTURING COMPANY RESEARCH & DEVELOPMENT DIVISION 32384 EDWARD MADISON HEIGHTS, MICHIGAN			10. Project/Task/Work Unit No.
			11. Contract(C) or Grant(G) No. (C) (G)
12. Sponsoring Organization Name and Address NASA LEWIS RESEARCH CENTER 21000 BROOKPARK ROAD CLEVELAND, OHIO 44135			13. Type of Report & Period Covered FINAL REPORT
			14.

15. Supplementary Notes

16. Abstract (Limit: 200 words)

NASA Lewis is engaged in extensive solar stirling research. This cost exercise is one element in a competitive proposed design effort. The costing contractor and the design contractor interacted at the concept level in an effort to assure that manufacturing designs were a product of the exercise.

17. Document Analysis a. Descriptors

"Manufacturing Cost" — The sum of material, labor and burden cost in the manufacture of both a specific item or total assembly.

"Purchase Cost" — The purchase cost including inbound freight and handling for specific components at the O.E.M. level in quantities deemed prudent for a given annual manufacturing volume.

18. Availability Statement:	19. Security Class (This Report)	21. No. of Pages
	20. Security Class (This Page)	22. Price

(See ANSI-Z39.18) See Instructions on Reverse OPTIONAL FORM 272 (4-77) (Formerly NTIS-35) Department of Commerce

RESEARCH SUMMARY

Title	Cost Analysis For Stirling Technology 25 KW Solar Drive Electrical Generator As Forecast Utilizing Pareto's Law
Contractor	Pioneer Engineering & Manufacturing Company, Research & Development Division
Principle Investigators	M. Stewart, W. Jackson, R. Osen, R. Heitsch
Objective	To analyze cost by functional groupings for competitive comparison. This costing technique to utilize Pareto's Law, where deemed applicable. 10,000 units per annum was the given volume.

AREAS OF COMPARISON

M.T.I.		S.T.C.
	RECEIVER	
Receiver Shell Arteries Wicking		Receiver Shell Reflux Boiler
	CONVERSION SYSTEM	
Stirling Engine With Vibration Assembly		Stirling Engine
	POWER GENERATION	
Linear Alternator		Hydraulic Output and Generator
	POWER CONDITIONING AND CONTROLS	
Temperature Sensors Accelerometers Auto Transformer Tuning Capacitors		
	AUXILIARIES	
Radiator Fan and Driver Water Pump and Driver		Filter 2/10 Micron Isolation Valve Fan and Driver Pump and Driver Radiator

Results: This device was processed and costed from dimensioned layouts and detail drawings. Tolerances were, in most cases, given by the design contractor. Where tolerances were not given, discussions with the design contractor were held and tolerances were assigned. Where exotic materials were encountered, availability was deemed to be market driven and the assumption that there would be adequate capacity available. All manufacturing processes considered are current state-of-the-art and do not reflect any forecast outreach. Final costs, as estimated by Pioneer, are based upon Pareto's Law which basically states that 20% of the major items constitute 80% of the whole. Identical approaches were taken on the competitive designs.

Technical Approach

Components were analyzed for complexity and 20% of the total detail were selected to be cost representative, utilizing Pareto's Law. These selected components were detail processed and costed utilizing Pioneer's computerized asset center costing method. Verbal and written dialogues were maintained with design contractors. Total costs were generated utilizing Pareto's Law and these cost reflect Michigan labor and material cost as we know for the year 1986. Extrapolations were made to reflect 1984 cost as well.

TABLE OF CONTENTS

RECAP OF MANUFACTURING COST FOR ONE (1) UNIT OF **ARRAY** ARRANGEMENT

STC 25 KW SOLAR DRIVEN STIRLING ENGINE ELECTRICAL GENERATOR

1986 COST

PAGE KEY			MATERIAL COST	DIRECT LABOR COST	BURDEN COST	SCRAP ALLOWANCE	MANUFACTURING COST	LABOR MINS.
8	1.	Receiver Shell Reflux Boiler	$ 117.19	$ 6.42	$ 15.88	$ 1.40	$ 140.89	29.01
20	2.	Stirling Engine	2,817.82	30.88	93.57	27.22	2,969.49	139.11
7	3.*	Hydraulic Output & Generator	3,134.00				3,134.00	
	4.	Synchronous Machine	N/A				N/A	
7	5.*	Filter 2/10 Micron Isolation Valve Fan & Driver Pump & Driver Radiator	1,530.75				1,530.75	
			$ 7,599.76	$ 37.30	$109.45	$ 28.62	$ 7,775.13	168.12
		Praeto's Extension @ 125% On Cost	$ 9,499.70	$ 46.63	$136.81	$ 35.78	$ 9,718.92	168.12

*Prorated components extracted from Purchasing Study.

Estimated Tooling Cost: $205,000

Estimated Capital Equipment: $5,264,400

RECAP OF MANUFACTURING COST FOR ONE (1) UNIT OF **ARRAY** ARRANGEMENT

STC 25 KW SOLAR DRIVEN STIRLING ENGINE ELECTRICAL GENERATOR

1984 COST

PAGE	KEY		MATERIAL COST	DIRECT LABOR COST	BURDEN COST	SCRAP ALLOWANCE	MANUFACTURING COST	LABOR MINS.
8	1.	Receiver Shell Reflux Boiler	$ 103.13	$ 5.84	$ 13.97	$ 1.25	$ 124.19	29.01
20	2.	Stirling Engine	2,479.68	28.10	83.28	24.23	2,615.29	139.11
7	3.*	Hydraulic Output & Generator	2,757.92				2,757.92	
	4.	Synchronous Machine	N/A				N/A	
7	5.*	Filter 2/10 Micron Isolation Valve Fan & Driver Pump & Driver Radiator	1,347.06				1,347.06	
			$ 6,687.79	$ 33.94	$ 97.25	$ 25.48	$ 6,844.46	168.12
		Praeto's Extension @ 125% On Cost	$ 8,359.74	$ 42.43	$121.56	$ 31.85	$ 8,555.58	168.12

*Prorated components extracted from Purchasing Study.

Estimated Tooling Cost: $205,000

Estimated Capital Equipment: $5,264,400

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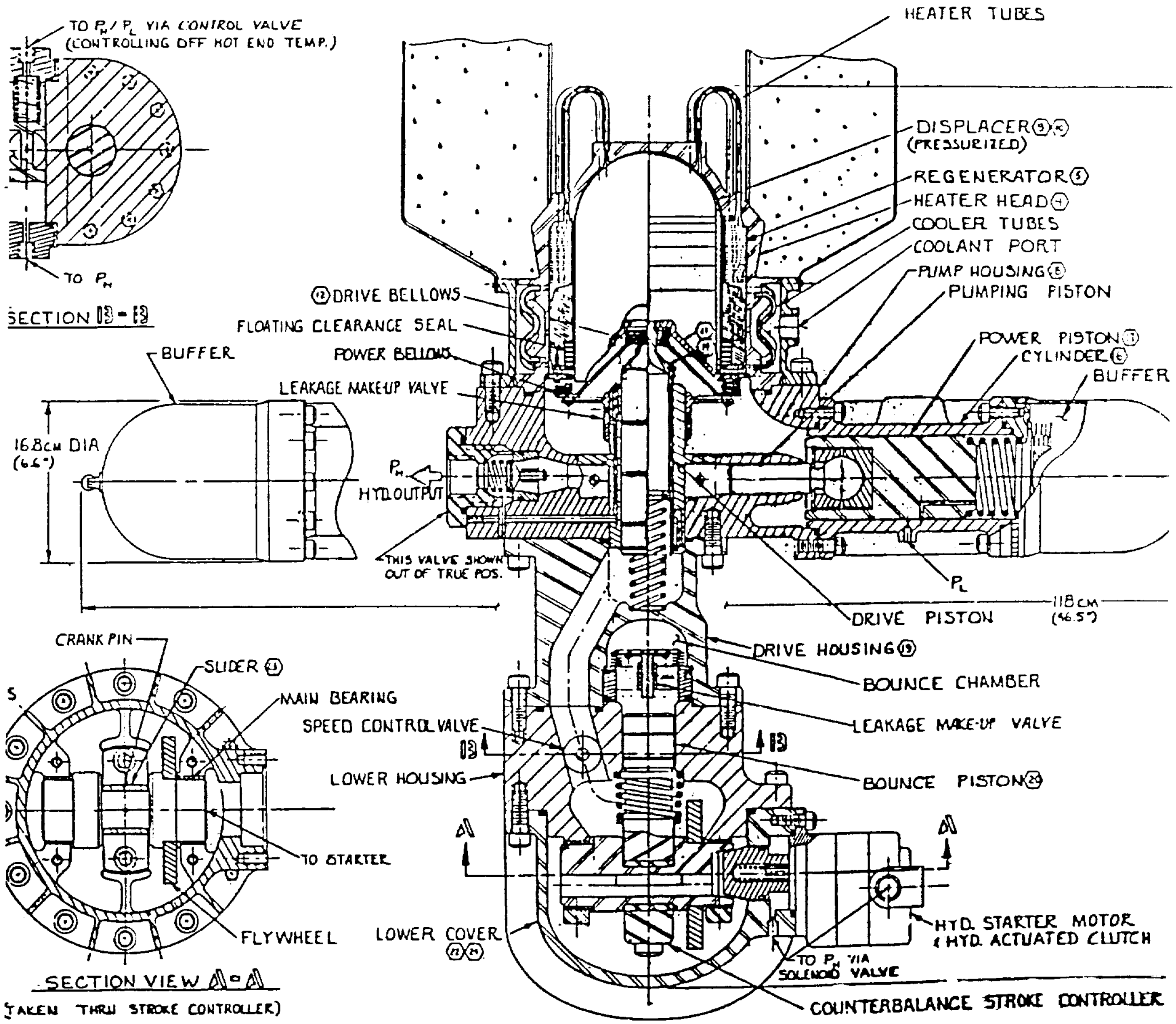
25 KW(E) SOLAR STIRLING HYDRAULIC ENGINE CONCEPTUAL DESIGN LAYOUT



STIRLING TECHNOLOGY COMPANY
STAND ALONE COST ANALYSIS

As an adjunct to the base study, Pioneer was requested to summarize a cost of the STC unit as a stand alone unit for direct comparison to the MTI unit. The following summary reflects our costing. A quotation schedule is included to show the development of our cost. There was not time for factory negotiated prices, so an average of 55% off list was used as a basic assumption. The component selection was done by STC. We have included dealer quoted prices and the STC estimates in our quotation schedule for comparison.

	LIST	-55%
Area #3	$6,529.96	$2,938.48
Area #5	$2,048.99	$ 922.05

All summaries include cost developed from the quotation study in Areas 3 and 5, where applicable.

RECAP OF MANUFACTURING COST FOR **STAND ALONE**

STC 25 KW SOLAR DRIVEN STIRLING ENGINE ELECTRICAL GENERATOR

1986 COST

PAGE KEY			MATERIAL COST	DIRECT LABOR COST	BURDEN COST	SCRAP ALLOWANCE	MANUFACTURING COST	LABOR MINS.
8	1.	Receiver Shell Reflux Boiler	$ 117.19	$ 6.42	$ 15.88	$ 1.40	$ 140.89	29.01
20	2.	Stirling Engine	2,817.82	30.88	93.57	27.22	2,969.49	139.11
7	3.**	Hydraulic Output & Generator	2,938.48				2,938.48	
	4.*	Synchronous Machine	NA	NA	NA	NA	NA	NA
7	5.**	Filter 2/10 Micron Isolation Valve Fan & Driver Pump & Driver Radiator	922.05				922.05	
			$ 6,795.54	$ 37.30	$109.45	$ 28.62	$ 6,970.91	168.12
		Praeto's Extension @ 125% On Cost	$ 8,494.43	$ 46.63	$136.81	$ 35.78	$ 8,713.65	168.12

*MTI extractions.

**STC listed components.

Estimated Tooling Cost: $205,000

Estimated Capital Equipment: $5,264,400

RECAP OF MANUFACTURING COST FOR **STAND ALONE**

STC 25 KW SOLAR DRIVEN STIRLING ENGINE ELECTRICAL GENERATOR

1984 COST

PAGE KEY			MATERIAL COST	DIRECT LABOR COST	BURDEN COST	SCRAP ALLOWANCE	MANUFACTURING COST	LABOR MINS.
8	1.	Receiver Shell Reflux Boiler	$ 103.13	$ 5.84	$ 14.13	$ 1.25	$ 124.35	29.01
20	2.	Stirling Engine	2,479.68	27.74	83.28	24.23	2,614.93	139.11
7	3.**	Hydraulic Output & Generator	2,585.86				2,585.86	
	4.*	Synchronous Machine	NA	NA	NA	NA	NA	NA
7	5.**	Filter 2/10 Micron Isolation Valve Fan & Driver Pump & Driver Radiator	811.40				811.40	
			$ 5,980.07	$ 33.58	$ 97.41	$ 25.48	$ 6,136.54	168.12
		Praeto's Extension @ 125% On Cost	$ 7,475.09	$ 41.98	$121.76	$ 31.85	$ 7,670.68	168.12

*MTI extractions.

**STC listed components.

Estimated Tooling Cost: $205,000

Estimated Capital Equipment: $5,264,400

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STAND ALONE HYDRAULIC AND COOLANT CIRCUIT

PARTS LIST

QTY	ITEM	NAME	DESCRIPTION	STC EST/KW	SUMMARY AREA #	OEM LIST $	AS QUOTED 25 PCS	AS QUOTED 500 PCS
1	1	Engine	STC 25 KW Solar Stirling	N/A	N/A	N/A	N/A	N/A
1	2	Coolant Radiator	41.6 BTU/min^{o}F Young #AV600S	14.88	#5	$1,995.00	$1,356.60	$1,276.80
1	3	Filter	10 micron, 23 gpm, 30 psi Fairey GA120SAE20-TX3D-10-V/E-22	4.84	#3	368.00	220.80	Factory
2	4	Pulsation Suppressor	23 gpm, 50 Hz NACOL #S210-02	6.00	#3	232.00	174.24	Factory
1	5	Isolation Valve	Manual, 30 psi, 23 gpm, BALL 1 1/4" pipe, Worchester 1 1/2" 446YBSW	6.04	#3	135.96	102.23	90.21
1	6	Pressure Switch	Close @ < 2750 psi, open @ 2950 psi ASCO #SB11A/TN10A21	2.36	#3	110.90		57.96*
1	7	Coolant Pump	6.2 gpm, .15 hp, 5 ft. head, Grundfos UPS-15-42F	1.84	#5	52.99	37.85	34.40
1	8	Isolation and Control Valve	1" Pipe, 3000 psi, 23 gpm, Worchester 1" H446YBSW, 3039S, and MK030	21.00	#3	496.26	373.13	333.64
1	9	Reservoir	2 gal, 30 psi	1.00	#3	28.60	Factory	Factory
1	10	Oil Cooler	6 KW, 23 gpm, 30 psi, 3.9 BTU/min^{o}F Young OCH55	15.20	#3	630.00	428.40	403.20
1	11	Auxiliary Pump and Motor	3000 psi, .1 gpm, piston type, 1/4 hp	4.00	#3	184.00	138.00	74.24
1	12	Filter	5 micron, 3000 psi, 23 gpm Fairey Arlon 174A-2N35-SL205	8.40	#3	295.00	177.00	Factory
1	13	Hydraulic Motor	Var. Displacement, 3000 psi, 28.4 KW Volvo V11-60-MS-SN-D-AC01-200/015-58.2/12.6-001-225/225	40.64	#3	2,416.00	1,256.00	Factory
1	14	Induction Generator	3 phase, 27 KW, 1800 rpm	20.00	#3	1,633.24	1,228.00	851.00
1	15	Piping		1.00	#5	1.00	1.00	1.00
				147.20		$8,578.95		

10,000 UNITS
ASSUME LIST - 559
AVERAGE.
THUS 8578.95 X 45
= $3860.53
3860.53 ÷ 25KW
= 154.42.

*1,500 Pieces.

RTG021 PROJECT - 1J

PIONEER ENGINEERING
BILL OF MATERIAL WITH COST 44.43

PAGE 1
87/10/16

VOLUME - 10,000 PART - 40 DESC - ASSY AREA # 1 RECEIVER VENDOR

COMPONENT	DESC - TRUNCATED TOOLING	WEIGHT	QTY	MATERIAL	LAB MIN	LABOR $	BURDEN	SCRAP	MARK-UP	TOT COST
2	ASSY RECEIVER		1	85.97	17.19	3.81V	.00	.00		
VENDOR	14.0	60.4968				M	7.51	.97	.00	98.26 *
1	APERTURE PLATE		1	9.69	5.25	1.15V	.00	.00		
VENDOR	2.0	7.4451				M	4.74	.16	.00	15.74 *
3	SUPPORT TUBE RECEVER		1	21.53	6.57	1.46V	.00	.00		
VENDOR	7.0	15.0000				M	3.63	.27	.00	26.89 *
COMPONENT TOTAL COST		82.9419		117.19	29.01	6.42V	.00	.00		117.19
	23.0					M	15.88	1.40	.00	140.89
ASSEMBLY COST				.00	.00	.00 V	.00	.00		.00
	.0					M	.00	.00	.00	.00
TOTAL COST		82.9419		117.19	29.01	6.42 V	.00	.00		117.19
						M	15.88	1.40	.00	140.89

TOOLING 23.0

EQUIPMENT 4,519.700

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RT6021	PROJECT - 1J	PIONEER ENGINEERING BILL OF MATERIAL WITH COST	44.43	PAGE 1 87/10/16		

VOLUME - 10,000 PART - 2 DESC - ASSY RECEIVER VENDOR

COMPONENT	DESC - TRUNCATED TOOLING	QTY WEIGHT	MATERIAL	LAB MIN	LABOR $	BURDEN	SCRAP	MARK-UP	TOT COST
2-A VENDOR	ABSORBER RECEVER 6.0	1 22.0968	31.37	2.02	.46V M	.00 1.58	.00 .33	.00	33.74 *
2-B VENDOR	AFT DOME RECEVER 5.0	1 38.4000	54.60	2.07	.48V M	.00 1.59	.00 .57	.00	57.24 *
	COMPONENT TOTAL COST 11.0	60.4969	85.97	4.09	.94V M	.00 3.17	.00 .90	.00	85.97 90.98
	ASSEMBLY COST 3.0		.00	13.10	2.87 V M	.00 4.34	.00 .07	.00	.00 7.28
	TOTAL COST	60.4969	85.97	17.19	3.81 V M	.00 7.51	.00 .97	.00	85.97 98.26

TOOLING 14.0

OPERATION DESCRIPTION

[illegible]	[illegible]			LAB RT -	[illegible]
MAT CODE -	ASSEMBLY				[illegible]
[illegible] -	[illegible]	WT TYPE -	[illegible]	[illegible]	[illegible]
[illegible] -	[illegible]	[illegible]	[illegible]	[illegible]	4.5[illegible]
[illegible] -	[illegible]	[illegible] -	[illegible]	SCRAP -	[illegible]
FINAL WT -	[illegible]	OTHER -	[illegible]	MATERIAL-	.0000

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RTE014	PROJECT - 1J	PIONEER ENGINEERING MANUFACTURING COST ANALYSIS	32.15	87/10/16

VOLUME- 10,000 P/A- 1
PART #- 2-A DESC- ABSORBER RECEVER UPG-

OPER	OPERATION DESCRIPTION EQUIP	M P	STD MIN	LAB COST LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING
005	BL2	.0	.020	.0000	.0003	V .00	.0000	.0000	.0
				.2357		M 29.04	.0087	.0087	
010	BC2	1.0	.020	.0047	.0003	V .00	.0000	.0000	5.0
				.2357		M 82.17	.0247	.0294	
020	7E5B	1.0	2.000	.4596	.0333	V .00	.0000	.0000	1.0
				.2298		M 46.53	1.5494	2.0090	

ANNUAL REQ-	10,000			LAB MIN -	2.0200		
MAT CODE -	ST/STL	ECON YR-LOC		LABOR $ -	.4643		
COST/LB -	1.400	PT TYPE	- VENDOR	BURDEN V-	.0000	TOOL $000	6.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	1.5828		
ROUGH WT -	22.4047	MARK-UP -	.0000	SCRAP -	.3341	TOTAL VAR	33.7478
FINAL WT -	22.0968	OTHER -	.000	MATERIAL-	31.3666	TOTAL MFG	33.7478

VOLUME- 10,000 P/A- 1
PART #- 2-B DESC- AFT DOME RECEVER UPG-

OPER	OPERATION DESCRIPTION EQUIP	M P	STD MIN	LAB COST LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING
005									
	BL2	.0	.020	.0000	.0003	V .00	.0000	.0000	.0
				.2357		M 29.04	.0087	.0087	
010									
	BC2	1.0	.020	.0047	.0003	V .00	.0000	.0000	5.0
				.2357		M 82.17	.0247	.0294	
020									
	7E5B	1.0	2.000	.4596	.0333	V .00	.0000	.0000	.0
				.2298		M 46.53	1.5494	2.0090	
030									
	19B	1.0	.050	.0107	.0008	V .00	.0000	.0000	.0
				.2146		M 14.20	.0114	.0221	

ANNUAL REQ-	10,000			LAB MIN -	2.0700		
MAT CODE -	ST/STL	ECON YR-LOC		LABOR $ -	.4750		
COST/LB -	1.400	PT TYPE	- VENDOR	BURDEN V-	.0000	TOOL $000	5.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	1.5942		
ROUGH WT -	39.0020	MARK-UP -	.0000	SCRAP -	.5667	TOTAL VAR	57.2387
FINAL WT -	38.4000	OTHER -	.000	MATERIAL-	54.6028	TOTAL MFG	57.2387

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VOLUME- 10,000 P/A- 1
PART #- 3 DESC- SUPPORT TUBE RECEVER UPG-

OPER	OPERATION DESCRIPTION EQUIP	M P	STD MIN	LAB COST LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING
005									
	BL2	.0	.020	.0000	.0003 V	.00	.0000	.0000	.0
				.2357		M 29.04	.0087	.0087	
010									
	8C2	1.0	.020	.0047	.0003 V	.00	.0000	.0000	5.0
				.2357		M 82.17	.0247	.0294	
020									
	7E5B	1.0	2.000	.4596	.0333 V	.00	.0000	.0000	.0
				.2298		M 46.53	1.5494	2.0090	
030									
	7S9	1.0	2.500	.5439	.0417 V	.00	.0000	.0000	1.0
				.2175		M 33.05	1.3782	1.9220	
040									
	7U1	1.0	2.000	.4370	.0333 V	.00	.0000	.0000	1.0
				.2185		M 19.81	.6597	1.0967	
050									
	19B	1.0	.050	.0107	.0008 V	.00	.0000	.0000	.0
				.2146		M 14.20	.0114	.0221	

ANNUAL REQ-	10,000			LAB MIN -	6.5700		
MAT CODE -	ST/STL	ECON YR-LOC		LABOR $ -	1.4558		
COST/LB -	1.400	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $000	7.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	3.6321		
ROUGH WT -	15.3770	MARK-UP -	.0000	SCRAP -	.2662	TOTAL VAR	26.8819
FINAL WT -	15.0000	OTHER -	.000	MATERIAL-	21.5278	TOTAL MFG	26.8819

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ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	2	ASSY - RECEIVER	1

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	060					1 %	___ %	V ✓ / C

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	WELD SUPPORT TUBE TO DOME	2A1	1	1.00	1.00
20	WELD ABSORBER TO DOME	2A1	1	2.00	2.00
30	CLEAN	14B	1	.050	—
40	GRIND	759	1	5.00	—
50	POLISH	7U1	1	5.00	—
60	INSPECT	19B	1	.050	—

— MATERIAL SPECS & BACK-UP DATA —

— SKETCH —

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NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		[illegible]			[illegible]	9-8-87		2

PEM 6-83 DF

ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.		PART NAME				PCS. REQ.	
10,000	2-A		ABSORBER - RECEIVER				1	
PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	041	1.40		22.4047	22.0968	1 %	___ %	V ✓

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
5	UNCOIL-STRAIGHTEN-FEED	8L2	—	.020	—
10	BLANK	8C2	1	.020	5.00
20	FLOTURN BLANK TO FORM DOME	7ESB	1	2.00	1.00

MATERIAL SPECS & BACK-UP DATA

316 STAINLESS STEEL
36"X36"X1/16

SKETCH

NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE ___ OF ___	PART NO.
		21447			MGS	9-8-87		2-A

PEM 6-83 DF

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ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	2 B	AFT DOME RECEIVER	1

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	041	1.40		39.002	38.400	1 %	___ %	V ✓ / C

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)	MATERIAL SPECS & BACK-UP DATA
5	UNCOIL-STRAIGHTEN-FEED	8L2	–	.020	–	316 STAINLESS STEEL 32"X32"X1/8"
10	BLANK	8C2	1	.020	5.00	
20	FLOTURN BLANK TO FORM DOME	YESB	1	2.00	–	
30	INSPECT	19B	1	.050	–	

— SKETCH —

NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		21447			MGS	9-8-87		2-B

PEM 6-83 DF

ESTIMATING DEPARTMENT — OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	1	APERTURE PLATE	1

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP RATE V	EQUIP RATE C
	003	1.21		8.0055	7.4451	1 %	___ %	✓	

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	TREPAN CENTER HOLE	7E5A	1	1.00	—
20	TURN O.D.	7E5A	1	2.00	—
30	DRILL BOLT HOLES	7G6	1	.250	2.00
40	DEBURR	7W3	1	2.0	—

MATERIAL SPECS & BACK-UP DATA

2024-T6 ALUM
38"X38"X.060

SKETCH

S T C

NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		21447			MGS	9-8-87		1

PEM 6-83 DF

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ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	3	SUPPORT TUBE - RECEIVER	1

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	041	1.40		15.377	15.00	1 %	___ %	V ✓ / C

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)	MATERIAL SPECS & BACK-UP DATA
5	UNCOIL-STRAIGHTEN-FEED	822	✓	.020	—	316 STAINLESS STEEL
10	BLANK	8C2	1	.020	5.00	20" X 20" X 1/8
20	FLOTURN BLANK TO FORM	7ESB	1	2.00	—	
30	GRIND I.D.	7S8	1	2.50	1.00	
40	POLISH I.D.	7U1	1	2.00	1.00	
50	INSPECT	19B	1	.050	—	

— SKETCH —

NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		21447			MGS	9-8-[illegible]		3

PEM 6-83 DF

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RTS021 PROJECT - 1J PIONEER ENGINEERING
BILL OF MATERIAL WITH COST 12.19 PAGE 1
87/11/12

VOLUME - 10,000 PART - 60 DESC - ASSY AREA #2 CONVERSION VENDOR

COMPONENT	DESC - TRUNCATED / TOOLING	QTY / WEIGHT	MATERIAL	LAB MIN	LABOR $	BURDEN	SCRAP	MARK-UP	TOT COST
4	HEATER HEAD	1	1884.393	3.85	.86V	.00	.00		
VENDOR	3.0	53.2485			M	2.87	18.88	.00	1907.00 *
5	REGENERATOR	10	433.580	2.40	.60V	.00	.00		
VENDOR	27.0	2.1500			M	3.30	4.40	.00	441.88 *
9	DOME-DISPLACER	1	79.785	3.10	.68V	.00	.00		
VENDOR	10.0	7.7000			M	2.59	.83	.00	83.89 *
10	BAFFLE	6	.216	.24	.06V	.00	.00		
VENDOR	5.0	.1068			M	.36	.00	.00	.63 *
11	COLD PLATE-DISPLACER	1	6.758	1.55	.34V	.00	.00		
VENDOR	.0	2.0948			M	1.12	.05	.00	8.33 *
14	DRIAE PISTON	1	2.156	3.01	.68V	.00	.00		
VENDOR	.0	3.3099			M	3.29	.06	.00	6.19 *
15	ASSY BUFFER	2	24.230	14.60	3.29V	.00	.00		
VENDOR	11.5	61.8190			M	10.80	.38	.00	38.69 *
16	POWER CYLINDER	2	30.498	9.25	2.02V	.00	.00		
VENDOR	3.0	37.2240			M	8.20	.40	.00	41.11 *
17	POWER PISTON	2	26.626	6.00	1.32V	.00	.00		
VENDOR	2.0	32.9740			M	5.48	.34	.00	33.76 *
18	PUMP HOUSING	1	27.444	14.10	3.17V	.00	.00		
VENDOR	.0	31.7776			M	11.89	.43	.00	42.93 *
19	BUFFER HOUSING	1	11.903	1.83	.41V	.00	.00		
VENDOR	2.0	15.2000			M	1.25	.14	.00	13.70 *
20	BOUNCE PISTON	1	2.176	4.35	.95V	.00	.00		
VENDOR	1.5	5.0989			M	4.03	.07	.00	7.22 *
21	CRANKSHAFT	1	5.001	8.01	1.75V	.00	.00		
VENDOR	.0	13.0164			M	5.12	.12	.00	11.99 *
23	SLIDER BLOCK	1	2.719	3.74	.82V	.00	.00		
VENDOR	1.5	1.7653			M	1.53	.05	.00	5.11 *
24	COVER	1	31.464	7.40	1.61V	.00	.00		
VENDOR	3.0	40.4000			M	8.05	.41	.00	41.53 *
25	CAP-SCOTCH YOKE	1	1.553	1.47	.32V	.00	.00		
VENDOR	1.0	3.6390			M	.85	.03	.00	2.75 *
33	HEATER TUBES	60	21.960	27.00	6.00V	.00	.00		
VENDOR	5.0	2.3940			M	11.40	.60	.00	39.96 *

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RT6021 PROJECT - 1J

PIONEER ENGINEERING
BILL OF MATERIAL WITH COST 12.19 87/11/12

VOLUME - 10,000 PART - 60 DESC - ASSY AREA #2 CONVERSION VENDOR

COMPONENT	DESC - TRUNCATED TOOLING	QTY WEIGHT	MATERIAL	LAB MIN	LABOR $	BURDEN	SCRAP	MARK-UP	TOT COST
34	COOLER TUBES	60	3.360	27.00	6.00V	.00	.00		
VENDOR	5.0	1.4100			M	11.40	.00	.00	20.76 *
12	BELLOWS - DRIVER	1	23.000	.00	.00V	.00	.00		
PURCHASED	.0	.0000			M	.00	.00	.00	23.00 *
38	BELLOWS - POWER	1	70.000	.00	.00V	.00	.00		
PURCHASED	.0	.0000			M	.00	.00	.00	70.00 *
39	BELLOWS - BUFFER	2	104.000	.00	.00V	.00	.00		
PURCHASED	.0	.0000			M	.00	.00	.00	104.00 *
41	BELLOWS - BOUNCE	1	25.000	.00	.00V	.00	.00		
PURCHASED	.0	.0000			M	.00	.00	.00	25.00 *
COMPONENT TOTAL COST		315.3282	2817.822	139.11	30.88V	.00	.00		2817.75
	80.5				M	93.57	27.22	.00	2969.42
ASSEMBLY COST			.00	.00	.00 V	.00	.00		.00
	.0				M	.00	.00	.00	.00
TOTAL COST		315.3282	2817.822	139.11	30.88 V	.00	.00		2817.75
					M	93.57	27.22	.00	2969.42

TOOLING 80.5

EQUIPMENT 4,379,700

VOLUME- 10,000 P/A- 1
PART #- 4 DESC- HEATER HEAD UPG-

OPER	OPERATION DESCRIPTION	EQUIP	M P	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
010		7E3C	1.0	2.500	.5745	.0417	V .00	.0000	.0000	.0
					.2298		M 51.46	2.1459	2.7204	
020		766	1.0	1.000	.2118	.0167	V .00	.0000	.0000	3.0
					.2118		M 34.74	.5802	.7920	
030		14B	1.0	.050	.0107	.0008	V .00	.0000	.0000	.0
					.2146		M 35.79	.0286	.0393	
040		14F	1.0	.250	.0531	.0042	V .00	.0000	.0000	.0
					.2124		M 25.55	.1073	.1604	
050		195	1.0	.050	.0107	.0008	V .00	.0000	.0000	.0
					.2146		M 14.20	.0114	.0221	

ANNUAL REQ-	10,000			LAB MIN -	3.8500		
MAT CODE -	ST/STL	ECON YR-LOC		LABOR $ -	.8608		
COST/LB -	29.400	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $000	3.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	2.9734		
ROUGH WT -	64.0950	MARK-UP -	.0000	SCRAP -	18.8813	TOTAL VAR	1,907.0085
FINAL WT -	53.2485	OTHER -	.000	MATERIAL-	1,884.3930	TOTAL MFG	1,907.0085

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PIONEER ENGINEERING
MANUFACTURING COST ANALYSIS

RT6014	PROJECT - 1J		3.44.08	PAGE 1 87/10/31

VOLUME- 10,000 P/A- 10
PART #- 5 DESC- REGENERATOR UPG-

OPER	OPERATION DESCRIPTION / EQUIP	M / P	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
010	BC2	1.0	.080	.0189 .2357	.0013	V .00 M 82.17	.0000 .1068	.0000 .1257	25.0
020	BC2	1.0	.160	.0377 .2357	.0027	V .00 M 82.17	.0000 .2219	.0000 .2596	2.0

ANNUAL REQ-	100,000			LAB MIN -	.2400		
MAT CODE -	FELT	ECON YR-LOC		LABOR $ -	.0566		
COST/LB -	75.000	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $000	27.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	.3287		
ROUGH WT -	.5781	MARK-UP -	.0000	SCRAP -	.4374	TOTAL VAR	44.1802
FINAL WT -	.2150	OTHER -	.000	MATERIAL-	43.3575	TOTAL MFG	44.1802

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RT6014 PROJECT - 1J PIONEER ENGINEERING MANUFACTURING COST ANALYSIS 1.56.02 PAGE 1 87/11/09

VOLUME- 10,000 P/A- 1
PART #- 9 DESC- DOME-DISPLACER UPG-

OPER	OPERATION DESCRIPTION	EQUIP	M P	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
005										
		BL3	.0	.050	.0000	.0008 V	.00	.0000	.0000	.0
					.2357		M 31.74	.0254	.0254	
010										
		BD3	1.0	.050	.0118	.0008 V	.00	.0000	.0000	10.0
					.2357		M145.47	.1164	.1282	
020										
		7E3B	1.0	3.000	.6702	.0500 V	.00	.0000	.0000	.0
					.2234		M 48.75	2.4375	3.1077	
030										
		19B	1.0	.050	.0107	.0008 V	.00	.0000	.0000	.0
					.2146		M 14.20	.0114	.0221	

ANNUAL REQ-	10,000			LAB MIN -	3.1000		
MAT CODE -	ST/STL	ECON YR-LOC		LABOR $ -	.6927		
COST/LB -	9.850	PT TYPE	- VENDOR	BURDEN V-	.0000	TOOL $000	10.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	2.5907		
ROUGH WT -	8.1000	MARK-UP -	.0000	SCRAP -	.8307	TOTAL VAR	83.8991
FINAL WT -	7.7000	OTHER -	.000	MATERIAL-	79.7850	TOTAL MFG	83.8991

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PIONEER ENGINEERING
MANUFACTURING COST ANALYSIS

RI6024 PROJECT - 10 2.22.43 PAGE 1

VOLUME- 10,000 P/A- [illegible]
PART #- 10 DESC- BAFFLE UFG-

OPER	OPERATION DESCRIPTION	EQUIP	M F	STD MIN	LAB COST LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING
005		BL2	.0	.030	.0000 .2357	.0005	V .00 M 29.04	.0000 .0145	.0000 .0145	.0
010		BC2	1.0	.030	.0071 .2357	.0005	V .00 M 82.17	.0000 .0411	.0000 .0482	5.0
020		14A2	1.0	.006	.0013 .2146	.0001	V .00 M 23.42	.0000 .0023	.0000 .0036	.0

ANNUAL REQ-	60,000			LAB MIN -	.0360		
MAT CODE -	ST/STL	ECON YR-LOC		LABOR $ -	.0084		
COST/LB -	1.310	PT TYPE - SENIOR		BURDEN V-	.0000	TOOL $ -	5.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	.0579		
ROUGH WT -	.0276	MARK-UP -	.0000	SCRAP -	.0010	TOTAL UF	.1025
FINAL WT -	.0178	OTHER -	.000	MATERIAL-	.0362	TOTAL MFG	.1025

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PIONEER ENGINEERING

RT3024 PROJECT - 10 MANUFACTURING COST ANALYSIS 2.22.43 PAGE 1 57/10/14

VOLUME- 10,000 P/A- 1

PART #- 1[illegible] DESC- COLD PLATE-DISPLACER UPG-

OPER	OPERATION DESCRIPTION	EQUIP	M / P	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
010		7510	1.0	.500	.1088	.0083 V	.00	.0000	.0000	.0
					.2175		M 41.17	.3417	.4515	
020		7515	1.0	1.000	.2230	.0167 V	.00	.0000	[illegible]	.0
					.2230		M 47.25	.7891	1.0[illegible]	
030		145	1.0	.050	.0107	.0008 V	.00	.0000	[illegible]	.0
					.2146		M 35.79	.0286	.0393	

ANNUAL REQ -	10,000			LAB MIN -	1.55[illegible]		
MAT CODE -	STL FORG	ECON [illegible]-LOD		LABOR $ -	.5428		
COST LB -	3.000	PT TYPE	- VENDOR	BURDEN V-	.0000	TOOL $[illegible]	.0
SCRAP FAC -	1.0[illegible]	MARK-UP FAC-	0.0[illegible]	BURDEN M-	1.[illegible]		
ROUGH WT -	2.2525	MARK-UP -	.000[illegible]	SCRAP -	.[illegible]	TOTAL [illegible]	8.[illegible]
FINAL WT -	2.0948	OTHER -	.000	MATERIAL-	6.7575	TOTAL MFG	8.[illegible]

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		PIONEER ENGINEERING	PAGE 1
RT6024	PROJECT - 1J	MANUFACTURING COST ANALYSIS	2.22.4[illegible] [illegible]

VOLUME- 10,000 P/A- 1
PART #- 14 DESC- DRIVE PISTON UPG-

OPER OPERATION DESCRIPTION

OPER	EQUIP	M F	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
010	7B3	1.0	2.500	.5635	.0417	V .00	.0000	.0000	.0
				.2254		M 65.80	2.7479	3.[illegible]	
020	13A2	1.0	.300	.0664	.0050	V .00	.0000	[illegible]	[illegible]
				.2212		M 84.65	.4233	.4[illegible]	
030	7S2	1.0	.160	.0354	.0027	V .00	.0000	.0000	[illegible]
				.2212		M 41.17	.1112	.1466	
040	[illegible]	[illegible]	[illegible]	.0107	.000[illegible]	V .0[illegible]	[illegible]	[illegible]	[illegible]
				.2146		M 14.2[illegible]	.0[illegible]	.0221	

ANNUAL REQ-	10,000			LAB MIN -	3.01[illegible]		
MAT CODE -	STEEL	ECON YR-LOD		LABOR $ -	.6760		
COST/LB -	.285	PT TYPE -	[illegible]	BURDEN V-	.0000	TOOL [illegible]	[illegible]
SCRAP FAC -	[illegible]	MARK-UP FAC-	[illegible]	BURDEN M-	3.2[illegible]		
ROUGH WT -	7.5652	MARK-UP -	.00[illegible]	SCRAP -	[illegible]	TOTAL VAR	[illegible]
FINAL WT -	3.3099	OTHER -	.000	MATERIAL-	[illegible]	TOTAL MFG	[illegible]

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PIONEER ENGINEERING
MANUFACTURING COST ANALYSIS

RTB024 PROJECT - 13 2.22.43 [illegible]

VOLUME- 10,000 P/A- 2
PART #- 15 DESC- ASSY BUFFER UPG-

OPER	OPERATION DESCRIPTION	EQUIP	M P	STD MIN	LAB COST LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING
010		2A1	1.0	1.000	.2234	.0167 V	.00	.0000	.0000	.5
					.2234		F 22.45	.3749	.5983	
020		7B1B	1.0	2.000	.4460	.0333 V	.00	.0000	[illegible]	[illegible]
					.2230		M 47.25	1.5734	2.0194	
030		7B6	1.0	.300	.0635	.0050 V	.00	.0000	.0000	1.0
					.2116		M 34.74	.1737	.2372	

ANNUAL REQ-	20,000			LAB MIN -	3.3000		
MAT CODE -	ASSEMBLY	ECON YR-LOC		LABOR $ -	.7029		
COST/LB -	.000	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $	[illegible]
SCRAP FAC -	1.0%	MARK-UP FAC-	[illegible]	BURDEN M-	2.1220		
ROUGH WT -	.0000	MARK-UP -	.0000	SCRAP -	.0285	TOTAL VAR	2.8534
FINAL WT -	.0000	OTHER -	.000	MATERIAL-	[illegible]	TOTAL MFG	2.8534

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RTS024	PROJECT - 13	PIONEER ENGINEERING MANUFACTURING COST ANALYSIS	2.22.43	PAGE 1 87/10/14

VOLUME- 10,000 P/A- 2
PART #- 15-A DESC- FLANGE-BUFFER UPG-

OPER	OPERATION DESCRIPTION EQUIP	M P	STD MIN	LAB COST LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING
010									
	7B1B	1.0	1.000	.2230	.0167 V	.00	.0000	.0000	.0
				.2230		M 47.25	.7891	1.0121	

ANNUAL REQ-	20,000			LAB MIN -	1.0000		
MAT CODE -	STEEL	ECON YR-LOC		LABOR $ -	.2230		
COST/LB -	.280	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $000	.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	.7891		
ROUGH WT -	11.7512	MARK-UP -	.0000	SCRAP -	.0430	TOTAL VAR	4.3454
FINAL WT -	10.0125	OTHER -	.000	MATERIAL-	3.2903	TOTAL MFG	4.3454

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RT6024 PROJECT - 13 MANUFACTURING COST ANALYSIS 2.22.43 87/10/14

VOLUME- 10,000 F/A- 2
PART #- 15-B DESC- DOME-BUFFER UFG-

OPER	OPERATION DESCRIPTION EQUIP	M P	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
005	BL3	.0	.050	.0000	.0008	V .00	.0000	.0000	.0
				.2357		M 31.74	.0254	.0254	
010	3D	1.0	.050	.0109	.0008	V .00	.0000	.0000	10.0
				.2181		M 23.50	.0188	.0297	
020	7E3B	1.0	3.000	.6702	.0500	V .00	.0000	.0000	.0
				.2234		M 48.75	2.4375	3.1077	
030	19B	1.0	.050	[illegible]	.0008	V .00	.0000	.0000	.0
				.2146		M 14.[illegible]	.0114	.0223	

ANNUAL REQ-	20,000			LAB MIN -	[illegible]		
MAT CODE -	STEEL	ECON YR-LOC		LABOR $ -	[illegible]		
COST/LB -	.260	PT TYPE - VENDOR		BURDEN V-	[illegible]	TOOL $000	10.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	2.4[illegible]		
ROUGH WT -	31.5150	MARK-UP -	.0000	SCRAP -	[illegible]	TOTAL VAR	12.13[illegible]
FINAL WT -	20.8970	OTHER -	.000	MATERIAL-	[illegible]	TOTAL MFG	12.1300

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VOLUME- 10,000 P/A- 2
PART #- 16 DESC- POWER CYLINDER UPG-

OPER	OPERATION DESCRIPTION EQUIP	M F	STD MIN	LAB COST LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING
010									
	7B16	.5	1.650	.1840	.0275	V .00	.0000	.0000	.0
				.2230		M 47.25	1.2994	1.4834	
020									
	7B16	.5	1.650	.1840	.0275	V .00	.0000	.0000	.0
				.2230		M 47.25	1.2994	1.4834	
030									
	766	1.0	.500	.1059	.0083	V .00	.0000	.0000	2.0
				.2118		M 34.74	.2883	.3942	
040									
	762	1.0	1.600	.3480	.0267	V .00	.0000	.0000	1.0
				.2175		M 24.19	.6459	.9939	
050									
	14F	1.0	.250	.0531	.0042	V .00	.0000	.0000	.0
				.2124		M 25.55	.1073	.1604	
060									
	1302	1.0	.080	.0177	.0013	V .00	.0000	.0000	.0
				.2212		M135.70	.1764	.1941	
070									
	75B	1.0	.500	.1088	.0083	V .00	.0000	.0000	.0
				.2175		M 33.05	.2743	.3831	
080									
	19B	1.0	.050	.0107	.0008	V .00	.0000	.0000	.0
				.2146		M 14.20	.0114	.0221	

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	PIONEER ENGINEERING		PAGE 2
RT6024 PROJECT - 10	MANUFACTURING COST ANALYSIS	2.22.43	87/10/14

VOLUME- 10,000 P/A- 2

PART #- 16 DESC- POWER CYLINDER UPG-

OPER	OPERATION DESCRIPTION									
		EQUIP	M	STD	LAB COST	OCC HRS	BURDEN	BURDEN	VAR COST	TOOLING
			P	MIN	LAB RATE		RATE	COST	MFG COST	

ANNUAL REQ-	20,000			LAB MIN -	4.6300		
MAT CODE -	CST IRON	ECON YR-LOC		LABOR $ -	1.0122		
COST/LB -	.760	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $000	3.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	4.1024		
ROUGH WT -	20.0643	MARK-UP -	.0000	SCRAP -	.2036	TOTAL VAR	20.5671
FINAL WT -	18.6120	OTHER -	.000	MATERIAL-	15.2489	TOTAL MFG	20.5671

RT6024 PROJECT - 13 PIONEER ENGINEERING MANUFACTURING COST ANALYSIS 2.22.47 PAGE 1 87/10/14

VOLUME- 10,000 P/A- 2
PART #- 17 DESC- POWER PISTON UPG-

OPER	OPERATION DESCRIPTION EQUIP	M P	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
010									
	781B	.5	1.500	.1673	.0250	V .00	.0000	.0000	.0
				.2230		M 47.25	1.1813	1.3486	
020									
	781B	.5	1.000	.1115	.0167	V .00	.0000	.0000	.0
				.2230		M 47.25	.7891	.9006	
030									
	763	1.0	1.000	.2175	.0167	V .00	.0000	.0000	2.0
				.2175		M 24.19	.4640	.6815	
040									
	758	1.0	.500	.1088	.0083	V .00	.0000	.0000	.0
				.2175		M 33.05	.2743	.3831	
050									
	14F	1.0	.200	.0425	.0033	V .00	.0000	.0000	.0
				.2124		M 25.55	.0843	.1268	
060									
	15E	1.0	.050	.0107	.0008	V .00	.0000	.0000	.0
				.2145		M 14.20	.0114	.0221	

ANNUAL REQ-	20,000			LAB MIN -	3.0000		
MAT CODE -	CST IRON	ECON YR-LOC		LABOR $ -	.6583		
COST/LB -	.760	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $[illegible]	2.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	2.7444		
ROUGH WT -	17.5170	MARK-UP -	.0000	SCRAP -	.1672	TOTAL VAR	15.8828
FINAL WT -	16.4870	OTHER -	.000	MATERIAL-	13.3125	TOTAL MFG	16.8828

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RTB024 PROJECT - 13 MANUFACTURING COST ANALYSIS 2.22.43 87/10/14

VOLUME- 10,000 P/A- 1
PART #- 1B DESC- PUMP HOUSING UPG-

OPER	OPERATION DESCRIPTION	EQUIP	M F	STD MIN	LAB COST / LAB RATE	OCC HRS		BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
010		72	1.0	14.000	3.1514	.2333	V	.00	.0000	.0000	.0
					.2251		M	50.78	11.8470	14.9984	
020		148	1.0	.050	.0107	.0008	V	.00	.0000	.0000	.0
					.2146		M	35.79	.0286	.0393	
030		198	1.0	.050	.0107	.0008	V	.00	.0000	.0000	.0
					.2146		M	14.20	.0114	.0221	

ANNUAL REQ-	10,000			LAB MIN -	14.1000		
MAT CODE -	CST IRON	ECON YR-LOC		LABOR $ -	3.1728		
COST/LB -	.760	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $000	.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	11.8870		
ROUGH WT -	36.1110	MARK-UP -	.00%	SCRAP -	.4250	TOTAL VAR	42.9292
FINAL WT -	31.7776	OTHER -	.00%	MATERIAL-	27.4444	TOTAL MFG	42.9292

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PAGE 1

RT6024 PROJECT - 1J MANUFACTURING COST ANALYSIS 2.22.47 07/10/14

VOLUME- 10,000 P/A- 1
PART #- 19 DESC- BUFFER HOUSING UFG-

OPER	OPERATION DESCRIPTION	EQUIP	M P	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
010										
		7E5B	1.0	1.250	.2873	.0208	V .00	.0000	.0000	.0
					.2298		M 46.53	.9678	1.2551	
020										
		7G6	1.0	.240	.0508	.0040	V .00	.0000	.0000	2.0
					.2118		M 34.74	.1390	.1898	
030										
		14F	1.0	.340	.0722	.0057	V .00	.0000	.0000	.0
					.2124		M 25.55	.1456	.2178	

ANNUAL REQ-	10,000			LAB MIN -	1.8300		
MAT CODE -	CST IRON	ECON YR-LOC		LABOR $ -	.4103		
COST/LB -	.760	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $000	2.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	1.2524		
ROUGH WT -	15.6616	MARK-UP -	.0000	SCRAP -	.1357	TOTAL VAR	[illegible]
FINAL WT -	15.2000	OTHER -	.000	MATERIAL-	11.9028	TOTAL MFG	[illegible]

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PIONEER ENGINEERING
MANUFACTURING COST ANALYSIS

RT6024 PROJECT - 1J 2.22.43 PAGE 1
87/10/14

VOLUME- 10,000 P/A- 1
PART #- 20 DESC- BOUNCE PISTON UFG-

OPER	OPERATION DESCRIPTION	EQUIP	M P	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
010		7B1B	.5	1.500	.1673 / .2230	.0250	V .00 / M 47.25	.0000 / 1.1813	.0000 / 1.3486	.0
020		7B1B	.5	1.500	.1673 / .2230	.0250	V .00 / M 47.25	.0000 / 1.1813	.0000 / 1.3486	.0
030		7B6	1.0	1.000	.2118 / .2118	.0167	V .00 / M 34.74	.0000 / .5802	.0000 / .7920	1.5
040		13D2	1.0	.050	.0111 / .2212	.0008	V .00 / M135.70	.0000 / .1086	.0000 / .1197	.0
050		7B8	1.0	1.000	.2175 / .2175	.0167	V .00 / M 33.05	.0000 / .5519	.0000 / .7654	.0
060		7B8	1.0	.750	.1631 / .2175	.0125	V .00 / M [illegible]	.0000 / .4131	[illegible] / .5762	.0
070		19B	1.0	.050	.0107 / .2146	.0008	V .00 / M 14.20	.0000 / .0114	.0000 / .0221	.0

ANNUAL REQ-	10,000			LAB MIN -	4.3500		
MAT CODE -	STEEL	ECON YR-LOC		LABOR $ -	.9488		
COST/LB -	.350	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $000	1.5
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	4.0278		
ROUGH WT -	6.2182	MARK-UP -	.0000	SCRAP -	.0715	TOTAL VAR	7.2245
FINAL WT -	5.0989	OTHER -	.000	MATERIAL-	2.1764	TOTAL MFG	7.2245

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PIONEER ENGINEERING
MANUFACTURING COST ANALYSIS

[illegible] PROJECT [illegible] [illegible] PAGE 1

VOLUME 10,000 P/A 1
PART # [illegible] DIESEL CRANKSHAFT [illegible]

OPER	OPERATION DESCRIPTION	EQUIP	M/P	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
010		7B16	.5	1.000	.1115	.0167 V	.00	.0000	.0000	.0
					.2230		M 47.25	.7891	.9006	
020		7B16	.5	1.000	.1115	.0167 V	.00	.0000	.0000	.0
					.2230		M 47.25	.7891	.9006	
030		[illegible]	1.0	[illegible]	[illegible]	[illegible]	.00	[illegible]	[illegible]	[illegible]
					.2094		M 41.9[illegible]	.7041	[illegible]	
[illegible]		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
					[illegible]		M 21.95	.7309	[illegible]	
050		[illegible]	1.0	[illegible]	[illegible]	[illegible]	.00	[illegible]	[illegible]	[illegible]
					[illegible]		M [illegible]	[illegible]	[illegible]	
[illegible]		[illegible]	1.0	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
					[illegible]		M[illegible]	[illegible]	[illegible]	
070		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	.00	[illegible]	[illegible]	[illegible]
					.2175		M 41.37	.7150	.9[illegible]	
080		14B	1.0	.050	.0107	.0008 V	.00	.0000	.0000	.0
					.2146		M 35.79	.0286	.0393	
090		14F	1.0	.330	.0701	.0055 V	.00	.0000	.0000	.0
					.2124		M 25.55	.1405	.2106	
100		19B	1.0	.050	.0107	.0008 V	.00	.0000	.0000	.0
					.2146		M 14.20	.0114	.0221	

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[illegible] ENGINEERING — PAGE [illegible]

[illegible] PROJECT - [illegible] — MANUFACTURING COST ANALYSIS — [illegible] [illegible]

VOLUME- 10,000 — P/A- [illegible]
PART #- [illegible] — DESC- [illegible] — UPG- [illegible]

OPER — OPERATION DESCRIPTION

SETUP	[illegible]	STD	LAB COST	[illegible]	BURDEN	BURDEN	MAT COST	TOOLING
	P	MIN	LAB RATE		RATE	COST	MFG COST	

ANNUAL REQ-	[illegible]			LAB MIN -	[illegible]		
MAT CODE -	[illegible]	[illegible]		[illegible] -	[illegible]		
COST/LB -	[illegible]	[illegible] -	[illegible]	[illegible] -	[illegible]	[illegible]	[illegible]
SCRAP FAC -	[illegible]	[illegible]	[illegible]	BURDEN M-	[illegible]		
ROUGH WT -	[illegible]	[illegible] -	[illegible]	[illegible] -	[illegible]	TOTAL [illegible]	[illegible]
FINAL WT -	[illegible]	OTHER -	[illegible]	MATERIAL-	[illegible]	TOTAL MFG	[illegible]

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RTG024 PROJECT - 1J MANUFACTURING COST ANALYSIS 2.22.43 87.10/14

VOLUME- 10,000 P/A- 1
PART #- 23 DESC- SLIDER BLOCK UPG-

OPER	OPERATION DESCRIPTION EQUIP	M P	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST / MFG COST	TOOLING
010									
	7P2	1.0	1.000	.2194	.0167	V .00	.0000	.000[illegible]	.5
				.2194		M 21.95	.3656	.585[illegible]	
020									
	7P2	1.0	.500	.1097	.0083	V .00	.0000	.000[illegible]	.5
				.2194		M 21.95	.1822	.291[illegible]	
030									
	7G3	1.0	.300	.0653	.0050	V .00	.0000	.000[illegible]	.0
				.2175		M 24.19	.1210	.18[illegible]	
040									
	7S6	1.0	1.000	.2175	.0167	V .00	.0000	.000[illegible]	.5
				.2175		M 32.67	.5456	.76[illegible]	
050									
	7V1	1.0	.500	.1093	.0083	V .00	.0000	.00[illegible]	.0
				.2185		M 19.81	.1644	.27[illegible]	
060									
	7C2	1.0	.250	.0544	.0042	V .00	.0000	.00[illegible]	.0
				.2175		M 20.50	.08[illegible]	.14[illegible]	
070									
	14E2	1.0	.140	.0308	.0023	V .00	.0000	.000[illegible]	.0
				.2200		M 23.87	.0549	.08[illegible]	
080									
	19B	1.0	.050	.0107	.0008	V .00	.0000	.000[illegible]	.0
				.2146		M 14.20	.0114	.022[illegible]	

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RT6024 PROJECT - 1J MANUFACTURING COST ANALYSIS 3.56 87/11/13

VOLUME- 10,000 P/A- 1
PART #- 23 DESC- SLIDER BLOCK UPG-

OPER	OPERATION DESCRIPTION EQUIP	M P	STD MIN	LAB COST LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING
010									
	7P2	1.0	1.000	.2194	.0167	V .00	.0000	.0000	.5
				.2194		M 21.95	.3666	.5860	
020									
	7P2	1.0	.500	.1097	.0083	V .00	.0000	.0000	.5
				.2194		M 21.95	.1822	.2919	
030									
	7B3	1.0	.300	.0653	.0050	V .00	.0000	.0000	.0
				.2175		M 24.19	.1210	.1863	
040									
	7S5	1.0	1.000	.2175	.0167	V .00	.0000	.0000	.5
				.2175		M 32.67	.5456	.7631	
050									
	7V1	1.0	.500	.1093	.0083	V .00	.0000	.0000	.0
				.2185		M 19.81	.1644	.2737	
060									
	7D2	1.0	.250	.0544	.0042	V .00	.0000	.0000	.0
				.2175		M 20.50	.0861	.1405	
070									
	14E2	1.0	.140	.0308	.0023	V .00	.0000	.0000	.0
				.2200		M 23.87	.0549	.0857	
080									
	19B	1.0	.050	.0107	.0008	V .00	.0000	.0000	.0
				.2146		M 14.20	.0114	.0221	

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VOLUME- 10,000 P/A- 1
PART #- 23 DESC- SLIDER BLOCK UPG-

OPER	OPERATION DESCRIPTION								
		EQUIP	M P	STD MIN	LAB COST LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING

ANNUAL REQ-	10,000			LAB MIN -	3.7400		
MAT CODE -	CRS	ECON YR-LOC		LABOR $ -	.8171		
COST/LB -	1.200	PT TYPE	- VENDOR	BURDEN V-	.0000	TOOL $000	1.5
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	1.5322		
ROUGH WT -	2.2660	MARK-UP -	.0000	SCRAP -	.0507	TOTAL VAR	5.1192
FINAL WT -	1.7653	OTHER -	.000	MATERIAL-	2.7192	TOTAL MFG	5.1192

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VOLUME- 10,000 — P/A- 1

PART #- 24 — DESC- COVER — UPG-

OPER	OPERATION DESCRIPTION EQUIP	M P	STD MIN	LAB COST LAB RATE	CCC HRS	BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING
010	7E5A	1.0	3.000	.6636	.0500	V .00	.0000	.0000	.0
				.2212		M 80.75	4.0375	4.7011	
020	7G6	1.0	2.000	.4236	.0333	V .00	.0000	.0000	3.0
				.2118		M 34.74	1.1568	1.5804	
030	7E5A	1.0	2.000	.4424	.0333	V .00	.0000	.0000	.0
				.2212		M 80.75	2.6890	3.1314	
040	14B	1.0	.050	.0107	.0008	V .00	.0000	.0000	.0
				.2146		M 35.79	.0286	.0393	
050	14F	1.0	.300	.0637	.0050	V .00	.0000	.0000	.0
				.2124		M 25.55	.1278	.1915	
060	19B	1.0	.050	.0107	.0008	V .00	.0000	.0000	.0
				.2146		M 14.20	.0114	.0221	

ANNUAL REQ-	10,000			LAB MIN -	7.4000		
MAT CODE -	STEEL	ECON YR-LOC		LABOR $ -	1.6147		
COST/LB -	.760	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $000	3.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	6.0511		
ROUGH WT -	41.4000	MARK-UP -	.0000	SCRAP -	.4113	TOTAL VAR	41.5411
FINAL WT -	40.4000	OTHER -	.000	MATERIAL-	31.4640	TOTAL MFG	41.5411

C-3

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OPERATION DESCRIPTION

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PIONEER ENGINEERING PAGE 1

RTE024 PROJECT - 1J MANUFACTURING COST ANALYSIS 3.5E 87/11/13

VOLUME- 10,000 P/A- 60

PART #- 3C DESC- HEATER TUBES UFE-

OPER	OPERATION DESCRIPTION	EQUIP	M P	STD MIN	LAB COST LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING
010		7A1	1.0	.200	.0437 .2185	.0033	V .00 M 23.42	.0000 .0773	.0000 .1210	.0
020		BE	1.0	.200	.0471 .2357	.0033	V .00 M 26.71	.0000 .0881	.0000 .1352	5.0
030		14E	1.0	.050	.0107 .2146	.0008	V .00 M 35.79	.0000 .0286	.0000 .0393	.0

ANNUAL REE-	600,000			LAB MIN -	.4500		
MAT CODE -	STL TUBE	ECON YR-LOD		LABOR $ -	.1015		
COST/LB -	8.330	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $000	5.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	.1940		
ROUGH WT -	.0438	MARK-UP -	.0000	SCRAP -	.0066	TOTAL VAR	.6678
FINAL WT -	.0399	OTHER -	.000	MATERIAL-	.3657	TOTAL MFG	.6678

RTB024	PROJECT - 12		PIONEER ENGINEERING MANUFACTURING COST ANALYSIS						3.56		PAGE 1 87/11/13
VOLUME- 10,000		P/A- 60									
PART #- 34		DESC- COOLER TUBES						UPB-			

OPER	OPERATION DESCRIPTION EQUIP	M F	STD MIN	LAB COST LAB RATE	OCC HRS		BURDEN RATE	BURDEN COST	VAR COST MFG COST	TOOLING
010										
	7A1	1.0	.200	.0437	.0033	V	.00	.0000	.0000	.0
				.2185		M	23.42	.0773	.1210	
020										
	8B	1.0	.200	.0471	.0033	V	.00	.0000	.0000	5.0
				.2357		M	26.71	.0881	.1352	
030										
	14E	1.0	.050	.0107	.0008	V	.00	.0000	.0000	.0
				.2146		M	35.79	.0286	.0393	

ANNUAL REQ-	600,000			LAB MIN -	.4500		
MAT CODE -	STL TUBE	ECON YR-LOC		LABOR $ -	.1015		
COST/LB -	2.250	PT TYPE - VENDOR		BURDEN V-	.0000	TOOL $000	5.0
SCRAP FAC -	1.0%	MARK-UP FAC-	0.0%	BURDEN M-	.1940		
ROUGH WT -	.0245	MARK-UP -	.0000	SCRAP -	.0035	TOTAL VAR	.3550
FINAL WT -	.0275	OTHER -	.000	MATERIAL-	.0560	TOTAL MFG	.3550

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[illegible]

PART #- [illegible]

OPERATION DESCRIPTION [illegible]

ANNUAL REQ-	[illegible]			[illegible]	[illegible]		
MAT CODE	STEEL	[illegible]		[illegible]	[illegible]		
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
SCRAP [illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]		
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

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PIONEER ENGINEERING [illegible]
[illegible] PROJECT - [illegible] MANUFACTURING COST ANALYSIS [illegible]

VOLUME- [illegible] [illegible]
PART #- [illegible] DESC- BELLOWS - POWER [illegible]

OPER. OPERATION DESCRIPTION

EQUIP [illegible] STD LAB COST TOT MFG [illegible] BURDEN [illegible] [illegible]
[illegible] MIN LAB RATE RATE COST MFG COST

ANNUAL REQ	[illegible]			LAB MIN -	[illegible]		
MAT CODE -	STEEL	[illegible]		LABOR $ -	[illegible]		
COST/LB -	[illegible]	PT TYPE	- PURCHASED	BURDEN [illegible]	[illegible]	TOOL [illegible]	[illegible]
SCRAP [illegible] -	[illegible]	PART-OF PKG-	0.04	BURDEN [illegible]	[illegible]		
[illegible] WT -	[illegible]	[illegible]	[illegible]	[illegible] -	[illegible]	TOTAL [illegible]	[illegible]
FINAL WT -	[illegible]	OTHER -	[illegible]	MATERIAL -	[illegible]	TOTAL [illegible]	[illegible]

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PART #- 39

OPER OPERATION DESCRIPTION

ANNUAL REQ- [illegible]

MAT CODE - STEEL

COST/LB - [illegible]

SCRAP FAC - [illegible]

ROUGH WT - [illegible]

FINAL WT - [illegible]

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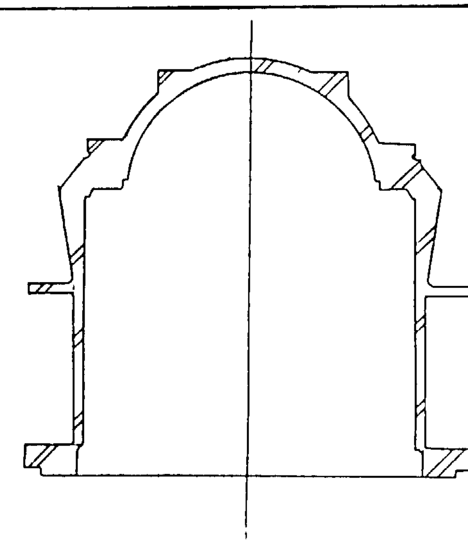
ESTIMATING DEPARTMENT — OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	[illegible]	[illegible]	

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V	EQUIP. RATE C
	[illegible]	29.40 [illegible]		[illegible]	[illegible]	[illegible] %	___ %	[illegible]	

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)	MATERIAL SPECS & BACK-UP DATA
10	FACE AND BORE I.D.	[illegible]	1	2.5	—	[illegible]
20	DRILL HEATER TUBES HOLES	[illegible]	1	1.0	3.00	
30	WASH	[illegible]	1	.050	—	
40	THERMAL DEBURR	[illegible]	1	.250	—	
50	INSPECT	[illegible]	1	.050	—	

SKETCH



HEATER HEAD

NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE ___ OF ___	PART NO.
		[illegible]			[illegible]	[illegible]		

PEM 6-83 DF

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ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS REQ
10,000	5	[illegible]	10

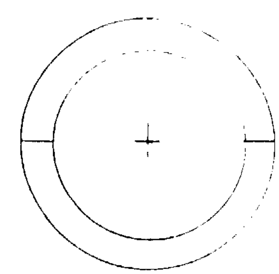
PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP RATE V / C
	017	75.00		.5781	.2150	1 %	___ %	V

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	BLANK CUT 1/2 CIRCLES	[illegible]	1	.090	25.0
20	TRIM SECTIONS	[illegible]	1	.160	2.00

— MATERIAL SPECS & BACK-UP DATA —

BRUNSWICK FELT
[illegible] SQ. FT. @ [illegible]
SQ. FT.

— SKETCH —



FINISHED PART

NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		21447			[illegible]	8-[illegible]		5

PEM 6-83 DF

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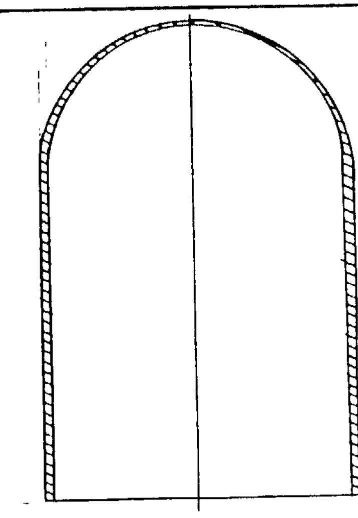
ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	5	[illegible]	[illegible]

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	[illegible]	[illegible]		8.1 RH	7.7 RH	1 %	___ %	V ✓

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)	MATERIAL SPECS & BACK-UP DATA
5	UNCOIL - STRAIGHTEN-FEED	[illegible]	—	.050	—	[illegible]
10	BLANK - PIECE	[illegible]	1	.050	10.0	
20	FLOTURN BLANK TO FORM DOME	[illegible]	1	[illegible]	—	
30	INSPECT	[illegible]	1	.050	—	

— SKETCH —



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NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE ___ OF ___	PART NO.
		[illegible]			[illegible]	[illegible]		5

PEM 6-83 DF

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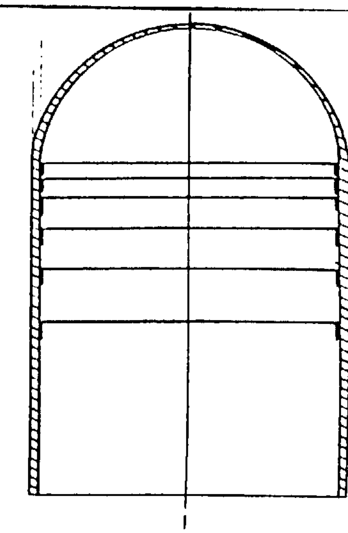
ESTIMATING DEPARTMENT — OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	10	BAFFLE	6

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	041	1.31		.0276	.0178	1 %	___ %	V: ✓ / C:

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)	MATERIAL SPECS & BACK-UP DATA
5	UNCOIL-STRAIGHTEN-FEED	8L2	—	.030	—	STAINLESS STEEL
10	BLANK-FORM	BC2	1	.030	5.00	
20	WASH	MA2	1	.006	—	

— SKETCH —



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NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		21447			MGS	8-21-87		10

PEM 6-83 DF

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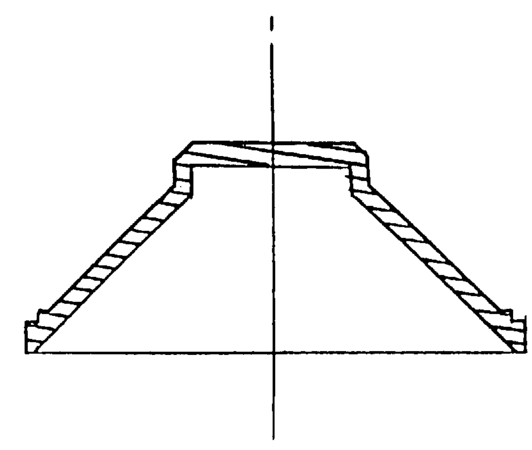
ESTIMATING DEPARTMENT — OPERATION SHEET

VOLUME	PART NO.		PART NAME					PCS. REQ.
10,000	11		COLD PLATE-DISPLACER					1
PPG/UPG NO.	**MATL. CODE**	**COST/LB.**	**OTHER COST**	**RGH. WT.**	**FIN. WT.**	**SCRAP**	**MARK-UP**	**EQUIP. RATE V / C**
	043	3.00		2.2525	2.0998	1 %	___ %	V ☑ C ☐

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)	MATERIAL SPECS & BACK-UP DATA
10	BLANCHARD GRIND FLANGE	7510	1	.500	—	15-5 STAINLESS STEEL
20	FACE AND BORE TOP	7B1B	1	1.000	—	
30	WASH	14B	1	.050	—	

— SKETCH —



NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		21447			MGS	5-21-87		11

PEM 6-83 DF

ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	14	DRIVE PISTON	1

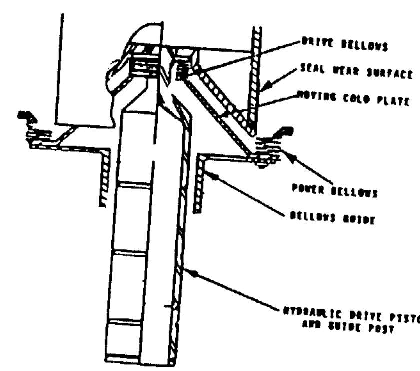
PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V	EQUIP. RATE C
	042	.285		7.5658	3.3099	1 %	___ %	✓	

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	DRILL, BORE, TURN, GROOVE, THREAD AND CUT-OFF	Y[illegible]3	1	2.50	—
20	HEAT TREAT	13PE	1	.300	—
30	GRIND O.D.	YS2	1	.160	—
40	INSPECT	19R	1	.050	—

— MATERIAL SPECS & BACK-UP DATA —

STEEL SAE 1020
2.00 DIA. 8.5 LONG

— SKETCH —



NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE ___ OF ___	PART NO.
		2144F			MGS	1-21-85		14

PEM 6-83 DF

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ESTIMATING DEPARTMENT — OPERATION SHEET

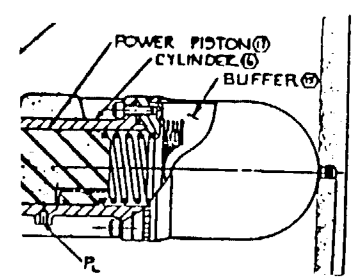
VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	15	ASS'Y- BUFFER	2

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	060					1 %	___ %	✓ /

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	WELD FLANGE TO DOME	2A1	1	1.00	.500
20	FACE, BORE, DRILL AND TAP DOME FOR FITTING	7B1B	1	2.00	—
30	DRILL AND TAP FLANGE HOLES	7G6	1	.300	1.00

— MATERIAL SPECS & BACK-UP DATA —

— SKETCH —



NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE	PART NO.
		2144Y			MGS	4-22-97	PAGE __ OF __	15

PEM 6-83 DF

ESTIMATING DEPARTMENT — OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	15-A	FLANGE - BUFFER	2

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V	EQUIP. RATE C
	042	.280		11.7512	10.0125	1 %	___ %	✓	

OPER	— OPERATION DESCRIPTION —	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)	— MATERIAL SPECS & BACK-UP DATA —
10	DRILL, TURN AND FACE ONE END	1B18	1	1.00	—	STEEL SAE 4130

— SKETCH —

NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE ___ OF ___	PART NO.
		21147			MGS	1-22-87		15-A

PEM 6-83 DF

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ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	15-B	DOME-BUFFER	2

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V	EQUIP. RATE C
	042	.280		31.518	20.897	1 %	___ %		

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)	MATERIAL SPECS & BACK-UP DATA
5	UNCOIL-STRAIGHTEN-FEED	8L3	-	.050	—	STEEL SAE 4130
10	BLANK-PIERCE	3D3	1	.050	10.0	
20	FLOTURN BLANK TO FORM DOME	7E3B	1	3.00	—	
30	INSPECT	19B	1	.050	—	

— SKETCH —

NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE ___ OF ___	PART NO.
		21997			MGS	10-13-87		15-B

PEM 6-83 DF

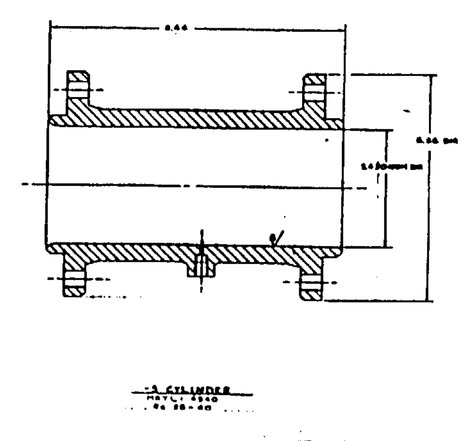
ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	16	POWER CYLINDER	2

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	042	6.00		21.345	19.800	1 %	___ %	V: ✓ C:

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)	MATERIAL SPECS & BACK-UP DATA
10	TURN, FACE FLANGE AND BORE ONE END	731B	.5	2.00	—	CAST STEEL 4340
20	TURN FACE FLANGE AND BORE ONE END	731B	.5	2.00	—	
30	DRILL FLANGE HOLES BOTH ENDS	7G6	1	.500	2.00	
40	DRILL, TAP AND SPOTFACE VENT HOLE	7G3	1	2.00	1.00	
50	THERMAL DEBURR	14F1	1	.250	—	
60	HEAT TREAT	13C2	1	.080	—	
70	GRIND PISTON BORE	7SB	1	.500	—	
80	INSPECT	19B	1	.050	—	

— SKETCH —



NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		21447			MGS	4-13-81		16

PEM 6-03 DF

ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.		PART NAME	PCS. REQ.
10,000	17		POWER PISTON	2

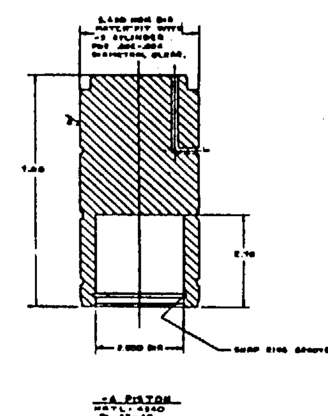
PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	008	.76		17.517	16.487	1 %	%	V ✓ / C

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	TURN, DRILL, BORE, GROOVE FACE ONE END	7B1B	.5	1.500	—
20	TURN, GROOVE AND FACE ONE END	7B1B	.5	1.00	
30	DRILL VENT HOLES	7G3	1	1.00	2.00
40	GRIND O.D.	758	1	.500	—
50	THERMAL DEBURR	1KF1	1	.200	—
60	INSPECT	192	1	.050	—

— MATERIAL SPECS & BACK-UP DATA —

CAST IRON

— SKETCH —



NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		21477			MGS	10-13-87		17

PEM 6-83 DF

ESTIMATING DEPARTMENT — OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	18	PUMP HOUSING	1

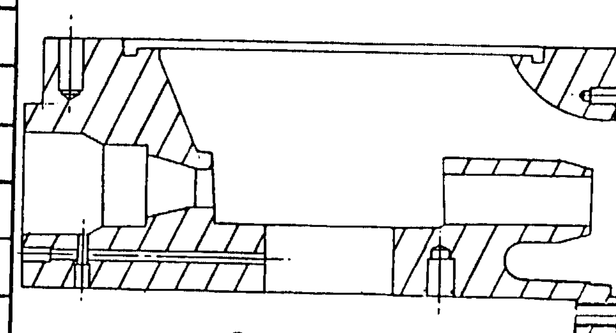
PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V	EQUIP. RATE C
	042	6.00		38.416	33.806	1 %	%	✓	

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	MACHINE TOP FACE, DRILL AND TAP MOUNTING HOLES AND BORE DRIVE PISTON BORE. MACHINE FACE, DRILL AND TAP MOUNTING HOLES AND BORE PUMPING PISTON BORE, BOTH SIDES. MACHINE BOTTOM FACE AND CHECK VALVE PORT	72	1	17.5	—
20	WASH	14B	1	.050	—
30	INSPECT	19B	1	.050	—

— MATERIAL SPECS & BACK-UP DATA —

CAST STEEL

— SKETCH —



PUMP HOUSING

NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		[illegible]			MGS	1-22-85		18

PEM 6-83 DF

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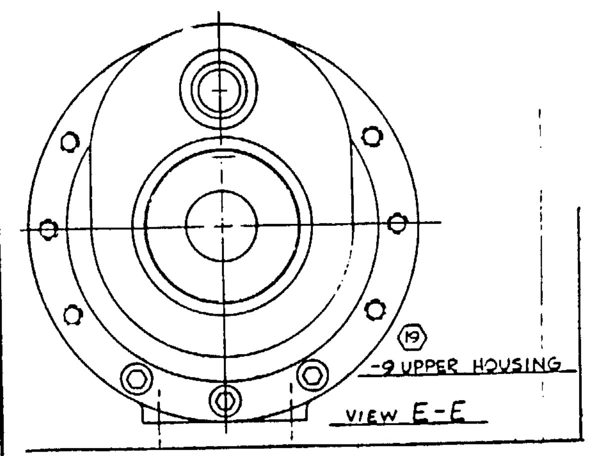
ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	19	BUFFER HOUSING	1

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	012	6.00		19.577	19.007	1 %	___ %	V ✓ / C

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)	MATERIAL SPECS & BACK-UP DATA
10	MACHINE BOTH ENDS AND BORE	7E5B	1	1.500	—	CAST STEEL
20	DRILL FLANGE HOLES	7G6	1	.300	2.00	
30	THERMAL DEBURR	14F	1	.340	—	

SKETCH



NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		2114Y			MGS	4-22-84		19

PEM 6-83 DF

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ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	20	BOUNCE PISTON	1

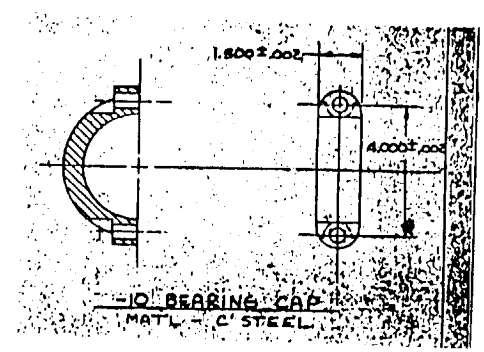
PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	042	.35		6.2182	5.0989	1 %	___ %	V: ✓ C:

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	TURN SHAFT AND GROOVE	7B1B	.5	1.500	—
20	TURN AND FACE FLANGE	7B1B	.5	1.500	—
30	DRILL, C'BORE AND TAP (2) HOLES	7G6	1	1.00	1.500
40	HEAT TREAT	13C2	1	.050	—
50	GRIND SHAFT	7S8	1	1.00	—
60	GRIND FLANGE O.D.	7S8	1	.750	—
70	INSPECT	19B	1	.050	—

MATERIAL SPECS & BACK-UP DATA

FORGING SAE 4130

SKETCH



NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE ___ OF ___	PART NO.
		[illegible]			MGS	5-6-87		20

PEM 6-83 DF

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ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	21	CRANKSHAFT	1

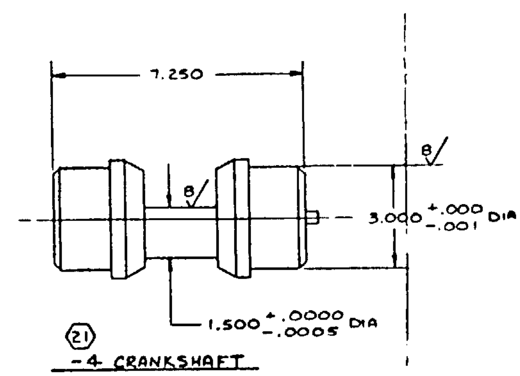
PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	018	.35		14.288	13.0164	1 %	%	V: ✓ C:

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	TURN AND FACE ONE END	781B	.5	1.00	—
20	TURN AND FACE ONE END	781B	.5	1.00	—
30	TURN JOURNAL	7F6	1	1.00	—
40	MILL TANG	7P2	1	2.00	—
50	DRILL AND COUNTERBORE BALANCE HOLE	7G3	1	2.50	—
60	HEAT TREAT	13C2	1	.080	—
70	GRIND JOURNAL	7S11	1	1.00	—
80	WASH	14B	1	.050	—
90	THERMAL DEBURR	14F1	1	.330	—
100	INSPECT	19B	1	.050	—

MATERIAL SPECS & BACK-UP DATA

FORGING SAE 4130

SKETCH



NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		21444			MGS	4-13-84		21

PEM 6-83 DF

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ESTIMATING DEPARTMENT   OPERATION SHEET

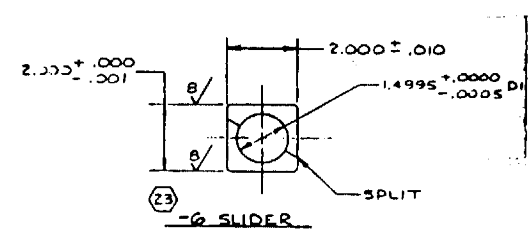
VOLUME	PART NO.	PART NAME		PCS. REQ.
10,000	23	SLIDER BLOCK		1

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	013	1.20		2.266	1.7653	1 %	___ %	V: ✓ C:

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	MILL (4) SIDES	7P2	1	1.00	.500
20	MILL (2) SIDES	7P2	1	.500	.500
30	DRILL & REAM SHAFT HOLE	7G3	1	.300	—
40	GRIND (2) SIDES	7S6	1	1.00	.500
50	HONE SHAFT HOLE	7V1	1	.500	—
60	SAW	7G2	1	.250	—
70	TUMBLE DEBURR	14E2	1	.140	—
80	INSPECT	19B	1	.050	—

— MATERIAL SPECS & BACK-UP DATA —

— SKETCH —



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NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE ___ OF ___	PART NO.
		21447			MGS	4-21-87		23

PEM 6-83 DF

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ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	24	COVER	1

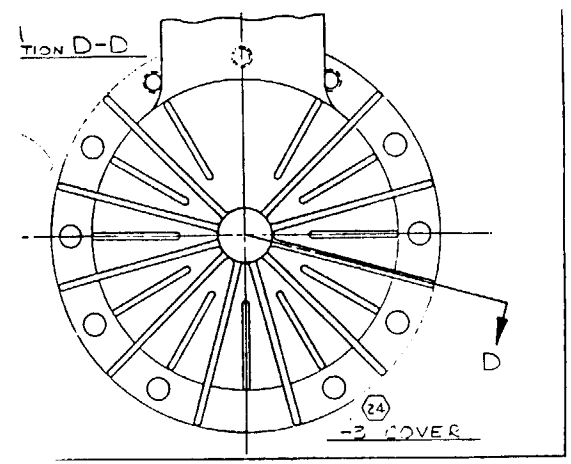
PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V	EQUIP. RATE C
	[illegible]	.76 [illegible]		41.4 [illegible]	40.4 [illegible]	1 %	___ %	[illegible]	

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	TURN FLANGE	[illegible]	1	3.00	—
20	DRILL & TAP FLANGE	[illegible]	1	[illegible]	5.00
30	FACE BORE [illegible] & TAP PORT	[illegible]	1	2.00	—
40	WASH	[illegible]	1	[illegible]	—
50	[illegible] DEBURR	[illegible]	1	.500	—
100	INSPECT	[illegible]	1	.050	—

— MATERIAL SPECS & BACK-UP DATA —

[illegible] NODULAR

— SKETCH —



NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE ___ OF ___	PART NO.
		[illegible]			[illegible]	[illegible]		[illegible]

PEM 6-83 DF

ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	25	CAP-SCOTCH YOKE	1

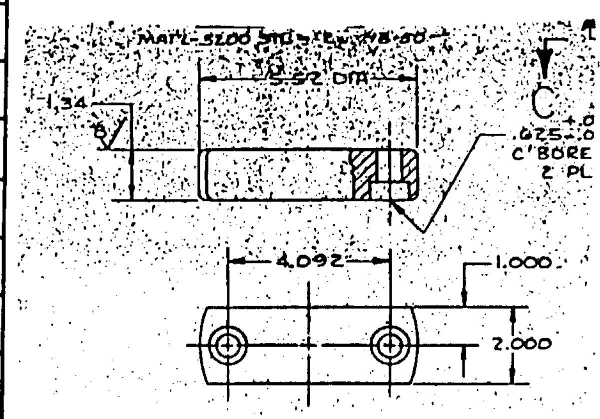
PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	18	.35		4.438	3.639	1 %	%	V ✓ / C

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	GRIND ONE SIDE	7510	1	.600	—
20	DRILL & C'BORE (2) PLACES	766	1	.500	1.000
30	TUMBLE DEBURR	14E2	1	.370	—

MATERIAL SPECS & BACK-UP DATA

SAE 4340 FORGING

SKETCH



NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		21447			MGS	5-6-87		25

PEM 6-83 DF

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ESTIMATING DEPARTMENT — OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	33	HEATER TUBES	60

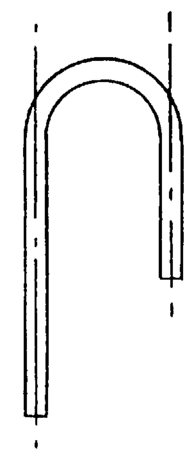
PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP. RATE V / C
	063	8.35		.0433	.0399	1 %	___ %	V ✓

OPER	OPERATION DESCRIPTION	EQUIP. CODE	M/P	LABOR MINS.	TOOLING $(000)
10	CUT TO LENGTH	7A1	1	.200	—
20	BEND 180° RETURN	8B	1	.200	5.0
30	WASH	74B	1	.050	—

MATERIAL SPECS & BACK-UP DATA

C-627 TUBING
O.D. .205 I.D. .165 WALL .020
9.5 LONG

SKETCH



HEATER TUBES

NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE ___ OF ___	PART NO.
		21447			MGS	8-23-82		33

PEM 6-83 DF

ESTIMATING DEPARTMENT OPERATION SHEET

VOLUME	PART NO.	PART NAME	PCS. REQ.
10,000	34	COOLER TUBES	[illegible]

PPG/UPG NO.	MATL. CODE	COST/LB.	OTHER COST	RGH. WT.	FIN. WT.	SCRAP	MARK-UP	EQUIP RATE V / C
	063	2.25 ea		.0249	.0235	1 %	%	V

OPER	OPERATION DESCRIPTION	EQUIP CODE	M/P	LABOR MINS.	TOOLING $(000)
10	CUT TO LENGTH	[illegible]	1	.200	—
20	BEND	8E	1	.200	5.00
30	WASH	ME	1	.050	—

MATERIAL SPECS & BACK-UP DATA

M.S. STEEL TUBING
.200 O.D. X .030 WALL
5" LONG

SKETCH

COOLER TUBES

NEXT ASM.	DWG. DATE	JOB NO.			ENGINEER	DATE	PAGE __ OF __	PART NO.
		21447			MGS	[illegible]		34

PEM 6-83 DF

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PIONEER ENGINEERING 87/10/19

RTB010 PROJECT - 3J EQUIPMENT COST ANALYSIS BY VOLUME 3.32.56 PAGE 1

VOLUME	EQUIPMENT		DESCRIPTION	TOTAL HOURS	EQUIPMENT COST	TOTAL EQUIP COST
10,000	2A1	V	AUTOMATIC	834.0	8,000	8,000
	3D	V	TINNING - BATH DIP	16.0	20,000	20,000
	5T3	V	12.5" FEED, 2.50" WD, .0937" STK	60.0	140,000	140,000
	7A1	V	1/8" TO 2" O.D. STOCK	1,980.0	20,000	20,000
	7B1E	V	2AC	4,167.0	80,000	80,000
	7B3	V	LARGE - 8-SPINDLE	417.0	280,000	280,000
	7E3B	V	MEDIUM	1,500.0	120,000	120,000
	7E3C	V	LARGE	417.0	200,000	200,000
	7E5A	V	6-SPINDLE DOUBLE INDEX BULLARD	1.333.0	750,000	750,000
	7E5B	V	2-SPDL NC CHUCKER MOTCH 235D	1,249.0	460,000	460,000
	7E6	V	CRANKSHAFT TURNING - SINGLE PIN	167.0	60,000	60,000
	7G3	V	MULTIPLE SPINDLE	1,467.0	2,400	2,400
	7G6	V	NATCO MDL G316-15 HP	1,108.0	75,000	75,000
	7P2	V	MEDIUM	583.0	26,000	26,000
	7Q2	V	RADIAL SAW - LIGHT DUTY 9" SHAPES	42.0	3,000	3,000
	7S2	V	CENTERLESS	27.0	75,000	75,000
	7S5	V	DBL. WHEEL GRINDER	167.0	67,000	67,000
	7S8	V	O.D. & I.D. GRINDER	1,041.0	40,000	40,000
	7S9	V	HAND GRINDER - PORTABLE	833.0	1,000	1,000
	7S10	V	BLANCHARD 48" TABLE	183.0	80,000	80,000
	7S11	V	CRANK GRINDING - SINGLE WHEEL	167.0	70,000	70,000
	7U1	V	SGL. SPINDLE	1,166.0	13,000	13,000
	7V1	V	SGL. STROKE UP TO 4" DIA.	83.0	15,000	15,000
	7W3	V	MANUAL DEBURR	333.0	300	300
	7Z	V	MACHINING CENTER	2,917.0	270,000	270,000
	8S	V	SMALL - UP TO 36" X 48" BED AREA	1,980.0	25,000	25,000

71

VOLUME	EQUIPMENT		DESCRIPTION	TOTAL HOURS	EQUIPMENT COST	TOTAL EQUIP COST
10,000	8C2	V	36" X 48" THRU 60" X 120" (W/AIR)	439.0	70,000	70,000
	8D3	V	72" X 120" THRU 100" X 200"	8.0	800,000	800,000
	8L2	V	2 1/2 TON	39.0	18,000	18,000
	8L3	V	5 TON	24.0	30,000	30,000
	13A2	V	MEDIUM PARTS	50.0	200,000	200,000
	13C2	V	MEDIUM PARTS	47.0	400,000	400,000
	14A2	V	MEDIUM	6.0	18,000	18,000
	14B	V	WASH	528.0	41,000	41,000
	14E2	V	LARGE MACHINE	85.0	24,000	24,000
	14F	V	ELECTRICAL DEBURRING	354.0	17,000	17,000
	19B	V	100% INSPECTION - VISUAL	136.0	1,000	1,000

TOTAL HOURS 25,953.0

TOTAL EQUIPMENT COST 4,519,700

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ASSET CENTER

MANUFACTURING COSTING METHODOLOGY

The methodology used in the development of manufacturing costs by Pioneer follows typical estimating procedures used in the auto industry. The specific approach is the application of burden and labor rates for each piece of equipment, or type of operation. This is labeled an "Asset Center Costing Methodology".

Some costing methodologies use department wide or plant wide burden rates. The later rates are average rates and do not reflect the costs of a specific operation. A particular operation may be the most expensive, or least expensive, in a department, or plant. A design may require a series of operations that are all above the "average" and therefore, an analysis conducted in this manner can be significantly above that of the "real" costs of producing a part.

The following paragraphs discuss the methodology in detail. Pioneer has developed computer programs utilizing micro-computers for the process and cost analyses. However, for clarification, the initial paragraphs describe an operation sheet used for manual process analysis. Later paragraphs discuss the current computerized version.

INITIAL EVALUATIONS

Manufacturing engineers analyze the part or assembly and list each of the manufacturing processes, or operations required to complete the fabrication cycle from the raw material to the finished product.

DETAILED PROCESSING AND COST ESTIMATING

Process engineers and cost estimators, under the direction of manufacturing engineers, conduct a detailed process and cost analysis for each part and assembly. All information developed during this analysis is recorded on the form shown in Figure 1. A Process/Cost Sheet is made out for each part and subassembly. The results are summarized to obtain the total assembly cost.

Two costs can be developed in this process, variable cost and manufacturing cost. The variable cost contains only those costs associated with the manufacture of the part or assembly. Manufacturing cost consists of the variable cost plus fixed burden costs.

An example of the process and cost estimating process shown in Figure 1 is discussed in the following paragraphs. This is a process sheet for forming a bumper face bar.

ESTIMATING DEPARTMENT — OPERATION SHEET

DRAWING DATE ____

OPER	OPERATION DESCRIPTION VOL. 300,000 P/A REQ. 300,000	TYPE OF EQUIPMENT	M/P	PCS/HR. / MINS.	LABOR COST / LABOR RATE	OCC HOURS		BURDEN RATE	BURDEN COST	VAR COST / MFG COST	DIE MODEL $(000)	TOOLING $(000)
	Flex Roll & Apply Drawing	Flex		480	.1628		V	32.217	.0670	.2298	-0-	-0-
10	Compound	Roll	4	.500	.3255	.00208	M	93.389	.1942	.3570		
	Rough Blank	300 Ton		400	.0977		V	"	.0805	.1782	-0-	9
20		Press	2	.300	.3255	.00250	M	"	.2335	.3312		
	Draw (Use Automatic Unloader)	Toggle		400	.0488		V	"	.0805	.1293	7.5	72
30		Press	1	.150	.3255	.00250	M	"	.2335	.2823		
	Cam From Top & Bottom, Pierce Parking Lite	84x120"		400	.0977		V	"	.0805	.1782	-0-	62
40	Openings & (9) Holes on Front Surface	SS Press	2	.300	.3255	.00250	M	"	.2335	.3312		
	Cam Trim Both Ends, Flange (2) P/L	"		400	.0488		V	"	.0805	.1293	-0-	50
50	Openings		1	.150	.3255	.00250	M	"	.2335	.2823		
	Fin. Flg. (2) Openings, Pierce (2) Pkg. Slots			400	.0488		V	"	.0805	.1293	4.0	48
60	& Cam Pierce, Bottom Attaching Holes		1	.150	.3255	.00250	M	"	.2335	.2823		
	Finish Cam Cup Engs			400	.0977		V	"	.0805	.1782	4.5	43
70			2	.300	.3255	.00250	M	"	.2335	.3312		
	NEXT OPER.											
	NEXT ASSY.											

SHT	COIL	GAGE	QUAL	MAT'L		PCS	ROUGH WT
X		121	DQ	HR	33" x 91" =	2	48.0724
				.950	Fine Grain		
					$.1571 X 48.0724 lbs.		7.5522
					(11.57# of Scrap	-	.1736
					@ .015 = .1736		7.3786
					Scrap Credit)		

COST PER LB .1571 MAT'L COST 7.3786

PART NAME
FACE BAR - FRONT BUMPER (1) REQUIRED

SKETCH REMARKS VOLUME	TOOL COST	
TRY-OUT		
	284,160	00
	MACH. & FACILITIES	
EQUIP	2,050,000	00
INSTALLATION	164,000	00
FREIGHT	61,500	00
TOTAL	$2,275,500	00

TOTAL VAR LABOR & BURD	1.1523			
TOTAL MFG LABOR & BURD	2.1975	$26.5	$284	
MATERIAL	7.3786	KEY ☐		
SCRAP 1%	.0738	NON KEY ☐		
SET UP		VOL ____		
OTHER		UPC ____		
MARK UP 10%				
TOTAL VAR. COST	8.7047	PART NO 22010		
TOTAL TRANS. COST	9.6499			

T & FC / PP JOB NO. 2744 PEM-3-80-A PROCESS ENG V.H. DATE 3-10-81 DET NO. 1

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The process sheet entries include all operations, from straightening the sheet steel to the final forming of the bumper.

The column headings and other items of interest on the process sheet are:

●OPER (Upper left corner)	Each operation is coded in this column. For this part seven distinct operations are required and are coded 10 through 70.
●VOL	The production volume at which the items are being costed.
●P/A	The number of pieces per assembly of the particular part being costed.
●REQ	The number of pieces per year required of the piece being costed. It is a product of VOL (Volume Per Year) and P/A (Pieces Per Assembly).
●OPERATION DESCRIPTION	Each distinct operation is described.
●TYPE OF EQUIPMENT	Capital Equipment employed in each operation.
●M/P	Number of men required for each operation.
●PCS/HR / MINS	PCS/HR is the pieces produced per hour per operation. MINS is the minutes per piece to process one piece through each operation.
●LABOR COST / RATE	LABOR COST is the <u>direct labor dollars</u> per piece. LABOR RATE is the <u>direct labor dollars</u> per minute (including fringes).
●OCC. HOURS	The time, in hours, that it takes to process the part through the operation. For example, if the production rate is 400 pieces per hour, the occupancy hours is one hour divided by 400 pieces per hour or .0025 hours per piece.
●BURDEN RATE	There are two burden rate entries, "V" for Variable Burden Rate and "M" for Manufacturing Burden Rate.

"V" (Variable Burden Rate) includes Set-Up Costs, In-Bound Freight, Perishable Production Tools, and other Miscellaneous Costs that vary with volume changes. "M" (Manufacturing Burden Rate) includes Variable and Fixed Burden. Fixed Burden covers Taxes, Insurance, Depreciation on Capital Equipment and Building, Maintenance Costs that do not vary with volume. See Figure 5 for a more definitive list of burden factors for both variable and fixed.

●BURDEN COST — Per piece burden cost is calculated by multiplying each burden rate by the occupancy hours.

●VAR COST / MFG COST — VAR COST is the variable burden plus direct labor cost. MFG COST is the cost of each operation including direct labor, variable burden, and fixed burden.

●DIE MODELS — Unique die models required for each operation.

●TOOLING — Dies, fixtures and other special tooling required for each operation. Tooling and equipment costs are summarized in the lower middle section.

●MATERIAL — Material is noted and cost calculated in the special box located on the lower left corner of the sheet. Column headings in this area are self explanatory. The type of material is determined in several ways; i.e., by specification on drawing, by chemical analysis, by contacting appropriate technical personnel responsible for material selection. Once the correct material specification is obtained appropriate sources are contacted to obtain the cost per pound of the material in the form and quantity required to produce the part.

●TOOLING COST SUMMARY — The total tooling cost for a given part is summarized in the lower middle section of the Process/Cost Sheet. The tooling cost is reported as a lump sum, leaving specific amortization up to the client. Tooling is an expense item and may be amortized in the year of use. Competitive economics, however, may preclude this move, so that a more extended amortization period may be used. Since this is a variable subject to the client's marketing strategy, tooling amortization is not a standard entry on these sheets. As a general rule the automotive firms amortize major tools and dies over a three year period. Pioneer has reported consumer costs which include the amortized tooling cost, usually in summary documents, if requested by the client.

●EQUIPMENT COSTS

The lower middle section summarizes cost of equipment, equipment installation and freight, and the cost of all pieces of equipment required to meet the production schedule. For instance, if the annual requirement is 300,000 units, and the shops works two shifts (4000 hours, or 250 days times 16 hours per day), the planning rate of production per operation is 93 units per hour ($\frac{300,000}{4,000}$ divided by .8, inherent delay factor), and if the equipment selected for the particular process can only produce 50 pieces per hour it is assumed that two such processes, or pieces of equipment, will be installed to meet the schedule.

●PART OR ASSEMBLY COST SUMMARY

Costs for producing the part are totaled in the lower right side of the form. The entries are:

TOTAL VARIABLE LABOR AND BURDEN; direct labor plus variable burden.

TOTAL MANUFACTURING LABOR AND BURDEN; direct labor, variable burden and fixed burden.

MATERIAL; total material cost.

SCRAP; an allowance for scrap based on experience. (% of Var. Cost)

MARKUP; since this is a part involving inter-divisional transfer, a markup is included.

TOTAL VARIABLE COST; the sum of items (a), (c) and (d).

TOTAL TRANSFER COST; the sum of (b), (c) (d) and (e).. This part is obviously a very high material sensitive part since approximately 70% of the transfer cost is reflected in the cost of steel.

All sub-assembly and final assembly cost will also be developed on these process sheets.

A work flow chart illustrating the methodology used to build up assembly cost is presented in Figure 2.

Figure 3 presents a flow diagram of the cost build up from basic cost items through consumer costs.

COST METHODOLOGY VIA COMPUTER PROGRAM

To permit more expeditious data processing, Pioneer uses a computer program to make all of the calculations discussed above.

Using the computer requires that the manufacturing engineer process the part being costed, select the equipment required, and define the operation cycle time. Figure 4 illustrates the Process/Cost Sheet prepared by the manufacturing engineer for the computer method. Note the equipment code specified for each operation. From this information the computer selects the appropriate labor and burden rates, as well as equipment costs. Using the specific cycle time, indicated manpower level and the equipment code, the computer calculates the labor cost, occupancy hours, variable burden, and manufacturing burden. It is also programmed to determine the multiples of a given machine required for an operation to produce the required number of pieces per hour. This is particularly important where costs are determined for a series of different production rates, where a process may not change from one rate to another, but only one machine may satisfy the requirement instead of two at a greater requirement. The scrap material costs are computed and the total cost is calculated.

Use of the computer permits error free accumulation of the total cost of a product, eliminating manual build up of sub-assembly to final assembly costs. Other cost data manipulations and extractions are possible using the computer which are cost prohobitive if attempted manually.

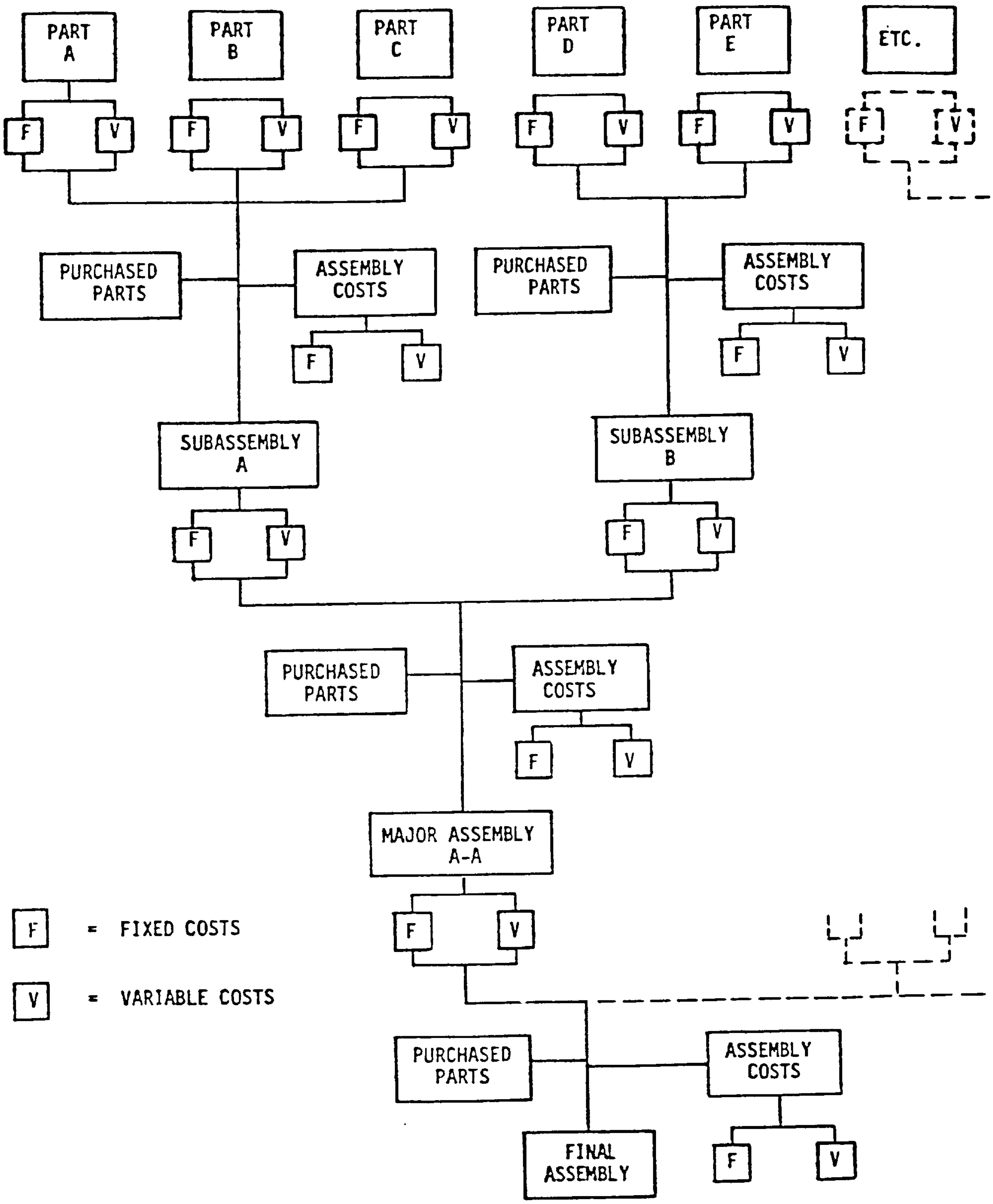
PART A
PART B
PART C
PART D
PART E
ETC.
F
V
PURCHASED PARTS
ASSEMBLY COSTS
SUBASSEMBLY A
SUBASSEMBLY B
MAJOR ASSEMBLY A-A
F = FIXED COSTS
V = VARIABLE COSTS
FINAL ASSEMBLY

Figure 2

DETERMINATION OF MANUFACTURING AND CONSUMER COSTS

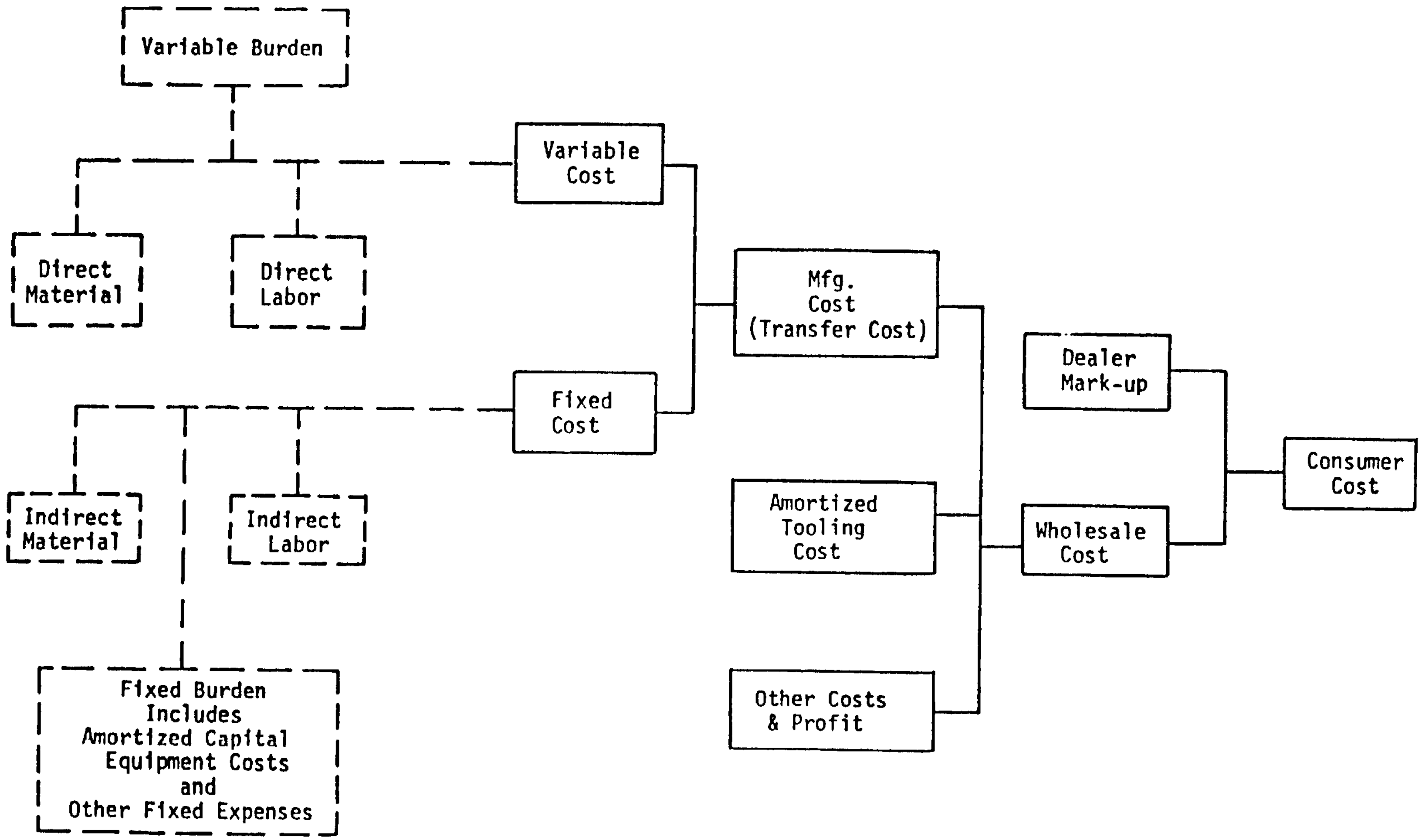

Figure 3

RT0005 PROJECT 1C — PIONEER ENGINEERING — MANUFACTURING COST ANALYSIS — 17.00.05 — PAGE 4/1(

VOLUME- 400,000 P/A- 1
PART #- 1 DESC- DOOR - OUTER PANEL - IN/WHITE UPG-

OPER	OPERATION DESCRIPTION / EQUIP	M P	STD MIN	LAB COST / LAB RATE	OCC HRS	BURDEN RATE	BURDEN COST	VAR MFG COST	DIE MODEL	TC $C
010	ROUGH BLANK FROM SHEET AUTO LOAD & EJECT / 8D1	1.00	.12	.0381 / .3174	.0020	V 66.12	.1322	.1703	0	
020	DRAW COMPL. AUTO LOAD, EJECT & TURNOVER / 8D1	1.00	.16	.0508 / .3174	.0027	V 66.12	.1785	.2293	0	
030	TRIM BINDER STK. & PIERCE LK. & HDL. HLS / 8D1	.00	.16	.0000 / .3174	.0027	V 66.12	.1785	.1785	0	
040	FORM HEM FLANGE 3 SIDES & BELT FLANGE / 8D1	.00	.16	.0000 / .3174	.0027	V 66.12	.1785	.1785	0	
050	FIN. FORM BELT FLANGE COMPLETE / 8D1	1.00	.16	.0508 / .3174	.0027	V 66.12	.1785	.2293	0	

ANNUAL REQ-	400,000			LAB MIN -	.4400		0
MAT CODE -				LABOR $ -	.1397		
COST/LB -	.294			BURDEN V-	.8462		
SCRAP FAC -	.0%						
ROUGH WT -	16.69			SCRAP -	.0000	TOTAL VAR	5.89
FINAL WT -	10.40	OTHER-	.00	MATERIAL-	4.9069		

Figure 4

BURDEN RATE DERIVATION

Pioneer does its cost estimating using the "Asset Center" burden approach, as opposed to the more common, less demanding technique of deriving manufacturing cost by applying departmental or plant wide burden as a percentage of direct labor cost. The "Asset Center" approach is not normally used by most companies because it requires a more refined and sophisticated data collection system, the complexity of which is shunned by comptrollers. It is however, more accurate and for this and many other reasons is the approach used by Pioneer. The following paragraphs review some of the philosophical rationale for using "Asset Center" burden rates.

Classically burden rates are historically determined — the burden rates for this year's projected costs are based on what was accumulated last year. The resultant burden rates are closely guarded secrets by most companies. The question could easily be asked, then, how does Pioneer — a consultant house with manufacturing operations —come to possess burden rates, especially in an "Asset Center" format?

Pioneer has been applying the "Asset Center" costing methodology for well over a decade. The costing personnel is, and has been, composed of individuals who have had significant, in depth, experience in costing and manufacturing, especially in the automotive industry. This depth of exposure has been harnessed to quantify the factors contributing to the operation of a nominal manufacturing facility. This process is tedious and time consuming, requiring a number of iterations to verify the choice of coefficients. The results are variable and manufacturing burden rates that are representative of a reasonably well managed production facility. These rates are for obvious reasons considered proprietary.

The evidence of the sufficiency of the burden rates has been two-fold. First, Pioneer has had the opportunity to compare its costs for various items directly with those produced for various companies by their personnel. These comparisons have been made on the level of labor, material, and burden costs, not merely an end item summary.

Second, Pioneer routinely does purchase analysis, that is, checking the cost being paid for purchased items. Where a Pioneer cost estimate is below the purchase cost, Pioneer has gone out to qualified vendors for new quotations. Literally millions of dollars have been saved by Pioneer clients where Pioneer costs have indicated that the purchase price should be lower than that being paid.

As a result Pioneer has gained confidence in the reliability of its "Asset Center" burden rates.

PIONEER ENGINEERING & MANUFACTURING

BURDEN FACTORS

FIXED

Salaries & Fringes
Maint. Repair
  (Grounds & External Bldg.)
Welding Equipment
Material Handling
Non Capitalized Project Expense
Preproduction Expense set up as
  a fixed cost

Dies (Maintenance)
Operating Supplies
Office Supplies

Janitor Supplies
Misc. Supplies
Heating
Transportation
Electric Power & Light
  (Based on min. rate x usage
   set by Utility)

Water
Communications
  (Wats)
Plant Protection

Non Productive Freight
Company Car & Travel Expense
Executive Fringes & Services
State & Local Taxes
Insurance
Depreciation
Pensions & Leaseholds

VARIABLE

Salaries & Fringes
Maint. Repair
  (Internal Bldg. & Production Equipt.)
Welding Equipment
Material Handling
Power Tools
Expense Tools

Set-up
Dies
Operating Supplies
Office Supplies
Welding Supplies
Janitor Supplies
Other Misc. Supplies

Transportation
Electric Power & Light

Fuel
Water

Other Purchased Services
  (i.e. Kelly Girls)
Non Productive Freight

Figure 5

Figure 5 lists the factors that have been considered in the determination of the Pioneer burden rates. The ratio of application of these costs between fixed and variable burden are not shown inasmuch as this is considered proprietary.

COST METHODOLOGY VARIANCE

Estimating as the name implies, is not an exact science, rigidly controlled by natural laws. There are variables. The variables are:

1. The method manufacture of the part.
2. The skill of the estimator.
3. The applicable labor and burden rates used by the estimator.
4. The estimating methodology.

Each of these variables is capable of producing differences in cost estimates of the same part.

Much of estimating is based on judgement. The first variable, method of manufacture, is judgement dominated. How a part is to be made is conditioned by the estimator's background and work experiences. For example, because one estimator's background is stamping-intensive, chances are his judgements (opinions), reflecting a higher degree of skill, will produce a highly reliable estimate of a sheet metal part. The same man, estimating a machined part, will not produce as reliable an estimate.

In many cases, there is no single, best way to make a part. When the production volume is large enough to justify a double tool-up, for example, some manufacturers will deliberately tool the same part differently in order to gain operating experience in their search of optimum methods. For example: Today, door panels — both inner and outer — are produced singly by one automotive company, and doubly (two-at-a-time) by a competitor. In each case, production volumes are similar. What factors prompted these dissimilar tool-ups? Presumably, both methods were considered by each process engineer before the final choice. Each had to consider the "economics" of both methods. Is one "more right" than the other? What this illustrates, is the flexibility inherent in the estimating process.

Some men, cautious by nature, will play it safe and "throw in two or three more operations".(1) This generosity is, in turn, compounded by the multiplier effect —three to five times — when the burden cost is applied.

(1)Operations = Steps in the manufacturing sequence.

From these examples, it is easy to see how estimating variances can occur in the first two variables.

The third variable, labor and burden rates, is the most abused element in cost estimating. The reason is that most estimators are excellent mechanics and engineers, know manufacturing techniques, but are poor financiers — most have only a rudimentary comprehension of how burden rates and burden costs are developed and applied. Their principal interest is in developing the manufacturing sequence, and specifying the equipment and tooling. Of secondary importance (interest) is the selection of the proper labor and burden rates. This step, performed almost casually by most estimators, is perhaps the most important in the estimating process because of the multiplier effect (most estimators calculate the burden cost of an operation by multiplying the direct labor cost by a burden percentage factor, usually two to eight times the labor cost).

Most manufacturing operations involve a single machine, such as a punch press, run by a single operator. To illustrate how the typical estimator develops a cost estimate, assume such a machine, run by a single operator, performing a forming operation, a sheet metal part, 300 parts per hour are produced in this operation. The direct labor, therefore, is .2 minutes per part ($\frac{60}{300}$). Assuming a direct labor cost of \$10.00 per hour the labor cost for this operation comes to:

$$\frac{.2 \text{ X } 10.00}{60} = \$.033$$

The next step is the calculation of the burden or factory overhead. Estimating departments have a schedule of burden rates, a specific rate for a specific machine, developed by the plant comptroller.

One of the methods used in calculating burden is to multiply the direct labor cost for a given operation by a percentage factor: e.g., 300%, 400%, etc. These percentage factors are developed from historical data accumulated over a number of accounting periods. These factors usually are based on data covering a whole department (sometimes on data which is not broken down below that of a whole plant). Consequently, the factors can be influenced by departmental conditions not specifically related to the operation itself. Burden rates based on historical data can very easily include inefficiencies that get lost in the overall departmental or plant operation.

Burden costs developed as a percentage of labor are still related to the type of equipment. It should be noted that labor can vary relative to a piece of equipment depending upon the complexity of the part and specific operation performed but the burden remains the same. As an illustration of this and expanding on the example discussed above:

Labor Cost (.033) X Burden Factor (300%) = \$.099.

The combined labor and burden cost for this operation, then, is .033 + .099 = \$.132.

Assume in our example that a second man, a helper, is required to man the stamping press. The labor cost now becomes \$.066 per operation per part. The unwary estimator will often assume that the burden cost should then be 300% X .066, or \$.198. This is obviously false, since the overhead doesn't double simply because another man has been added. Only the incremental costs, in this situation, associated with the additon of the second man should be added to the base cost calculated earlier. The estimator should "up the cost" of the operation by only the direct labor cost of the second man (\$.033). The burden cost would remain as it was when one man operated the press. The new cost for the press operation, now manned by an operator and a helper, is .033 + .033 + .099 = \$.165.

Another problem which occurs frequently in estimating, is the application of burden to an unmanned manufacturing operation. For example, assume a sequence of six press operations required to make a stamping. The first, or blanking operation, required two operators to remove the blank, dope it with lubricant and insert it in the draw die of the following operation, making sure that two blanks have not stuck together (a double blank could wreck the draw die). The next three operations are loaded and unloaded mechanically, the part is even inverted between operation 3 and 4, all without operator intervention. The final operation, a cam-piercing operation, requires one operator who removes the part, applies a dab of paint for indentification, and hangs the part onto a conveyor.

What cost does the estimator assign to each operation? If he is using the burden percentage method, there is no problem with the first and final operations, since these have operators. The estimator simply calculates the direct labor cost for each of these, then multiplies these by the burden percentage rates to obtain the burden cost, making sure, of course, that he has not doubled the burden cost in the first operation which has two operators.

The problem arises when the estimator tries to apply his formula to those operations which are unmanned. There is no direct labor cost, nothing he can multiply by his burden percentage rate. The unwary estimator will frequently assume that, since there is no labor cost, there can be no burden cost.

We know this to be false, since all of the burden elements — with the exception of fringe benefits — are still there whether an operator is present or not.

Another method of burden cost calculation used by Pioneer, is the "Burden Center" concept.

Whereas the "Burden Percentage" method covers a full department, sometimes an entire plant, the "Burden Center" approach considers a much smaller entity: a single machine plus only those expenses directly associated with the operation of the machine. These expenses are both variable (expenses which vary with product volume changes) and fixed (expenses which are unaffected by volume changes).

Typical variable expenses considered in burden would be (this is not a complete list):

- Indirect Labor
- Maintenance
- Perishable Tools
- Fringe Benefits
- Fuel
- Utilities

Typical fixed and non-variable expenses would be:

- Taxes
- Insurance
- Amortization
- Some Supervision
- Some Clerks & Janitors
- Some Utility Bills

A pro rata share of each of these elements is assigned to each burden center. The result is a carefully-developed, localized cost for a specific machine or other asset, reflecting only those expenses unique to that machine. These costs are stated in "dollars per machine-hour" giving rise to the expression: Machine-hour rate.

"Burden Center" rates can be generated historical data, or they can be developed from equipment specifications and requirements for power, lubrication, light, heat, indirect labor, average maintenance, material handling, and other costs required to keep the equipment operating. The latter method of burden development is beneficial when developing costs for a new plant or facility where historical data has not been developed. Another advantage in the latter approach is that nominal burden costs can be developed around nominal equipment production rates.

Costs developed around nominal production rates for a piece of equipment are an important consideration when assessing production costs. For example, a piece of equipment has a theoretical production rate for which it is designed. This theoretical rate may not be achieved because of inherent equipment and human operational conditions. However, "nominal" rates have been established through experience of an acceptable "efficient" plant. Well managed plants can achieve these nominal rates. All cost analyses should be developed around burden rates based on "nominal" production standards. Costs developed with burden rates established with other than nominal standards should not be used for comparison because they include variances in production inefficiencies and do not have a common base. Pioneer costs are established around nominal production rates.

There are other cost methodologies. One such method uses the cost-per-pound approach. Under this method, the parts of a car, for example, are grouped by classes of material: steel stampings, castings, forgings, molded plastics, etc. The cost of each part is divided by its finished weight, and a cost-per-pound obtained: a "meat-market" approach. Pioneer does not endorse this method because of its dependence on a straight-line relationship between weight and cost. For example, if a seven-pound brake drum cost \$3.50, will a nine-pound drum cost \$4.50? (\$.50 per pound.) Unlikely. The labor and burden will remain essentially the same for each size of drum, but the material cost, obviously, will be different. In spite of its imprecision, the method has some utility: as a "rough-and-dirty" indicator of approximate cost, as a crude verification that the estimate is "in the ball park".

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16. Abstract

This report documents the conceptual design and analysis of a solar thermal free-piston Stirling hydraulic engine system designed to deliver 25 kWe when coupled to the 11-meter Test Bed Concentrator at Sandia National Laboratories. A manufacturing cost assessment for 10,000 units per year was made by Pioneer Engineering and Manufacturing. The design meets all program objectives including a 60,000-hr design life, dynamic balancing, fully automated control, >33.3% overall system efficiency, properly conditioned power, maximum utilization of annualized insolation, and projected production costs of $300/kW. The system incorporates a simple, rugged, reliable pool boiler reflux heat pipe to transfer heat from the solar receiver to the Stirling engine. The free-piston engine produces high-pressure hydraulic flow which powers a commercial hydraulic motor that, in turn, drives a commercial rotary induction generator. The Stirling hydraulic engine uses hermetic bellows seals to separate helium working gas from hydraulic fluid which provides hydrodynamic lubrication to all moving parts. Maximum utilization of highly refined, field proven commercial components for electric power generation minimizes development cost and risk. The engine design is based on a highly refined Stirling hydraulic engine developed over 20 years as a fully implantable artificial heart power source.

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