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November 29, 1977

Mr. Robert E. Adams, Sr.
Department of Energy Technical Information Center
P.O. Box 62
Oak Ridge, Tennessee 37830

Subject: Phase I Report Issued Under Contract Number EY-76-C-02-2764

Dear Mr. Adams:

In response to your letter of November 1, 1977, the report dated March 30, 1977, and identified as No. 77TR40, should carry the number C00-2764-001. If you have any further questions, please let me know.

Sincerely,

Steven J. Piller
Project Manager

SJP/mlg

DEMONSTRATION OF A FREE PISTON STIRLING
ENGINE DRIVEN LINEAR ALTERNATOR
PHASE I REPORT

March 30, 1977

NO. 77TR40

DATE: March 30, 1977

TECHNICAL REPORT

DEMONSTRATION OF A FREE PISTON STIRLING ENGINE DRIVEN LINEAR ALTERNATOR PHASE I REPORT

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Prepared for

Division Of Nuclear Research & Applications
Energy Research & Development Administration

Prepared under

Contract No. EY-76-C-2764

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Prepared Under:
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Energy Research & Development Administration

MECHANICAL TECHNOLOGY INCORPORATED
968 Albany-Shaker Road
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SUMMARY

This report describes the results of the work performed under Phase I of the Free Piston Stirling Engine Demonstrator Program. The objective of this program is to develop a 2 KW Free Piston Stirling Engine/Linear Alternator energy conversion system, for an isotopic heat source, with a greater than thirty percent overall efficiency. Phase I was a 15-month effort to demonstrate the feasibility of the system through analysis and experimental testing of the individual components. Phase II will demonstrate the performance and life capability of the 2 KW energy conversion system.

The goals achieved during Phase I are as follows:

1. The preliminary design of an advanced free piston Stirling engine which was matched to a linear alternator was completed. An overall engine thermal efficiency of 45% was projected as a reasonable level of performance for this engine at the required operating conditions. A preliminary conceptual mechanical arrangement of the engine/alternator system was also developed.
2. The design and initial performance tests of a linear alternator were successfully completed during Phase I. Although the measured performance was somewhat lower than desired due to the decision to fabricate the alternator with conventional methods and materials, the predictive design tools were verified and areas of improvement and correction identified.
3. A model free piston Stirling engine was procured from Sunpower, Inc., installed, and successfully tested at MTI during Phase I. The engine testing was intended to provide operating experience, establish instrumentation techniques and test procedures, and obtain data to validate the computer simulation tools also being developed under this program. The level of performance of the engine was of interest in assessing the overall

potential efficiency and the results confirmed projections. Projection of the test data to the expected operating temperature of the demonstrator engine indicated that with the model engine in its present state, a thermal efficiency of 30% was achievable.

4. Procurement and modification of a free piston Stirling engine computer simulation developed by T. Finkelstein, installation of that code, and successful operation on MTI's computer facility was also accomplished. Validation of the code with the actual model engine data is being continued in Phase II of the demonstrator program. The present status of this effort can be summarized as follows:
 - a. The computer code as modified after delivery has operated successfully, although the model has not been run to steady-state conditions. It seems unlikely that a steady-state solution can be achieved without allowing the program to proceed through many cycles.
 - b. With the 13 node arrangement, as delivered, the accuracy of the solution is questionable. At present, the code is being extended to 24 nodes.

An introduction to Stirling engines and the details of the above tasks are presented in this report in five major sections.

- Section 1.0 - Introduction to Stirling Engine
- Section 2.0 - Preliminary Design of an Advanced Free Piston Stirling Demonstrator Engine
- Section 3.0 - Design and Test of a 1 KWE Output Linear Alternator
- Section 4.0 - Test of a Model Free Piston Stirling Engine
- Section 5.0 - Development of a Free Piston Stirling Engine Computer Simulation Code

SECTION 1.0

INTRODUCTION TO STIRLING ENGINES

The Stirling engine can be described as a closed cycle piston engine with cyclic recirculation of the working fluid. Power is produced in the conventional manner; that is, compression of the working gas at low temperature, heating the gas, expanding it at high temperature and as a result, cooling it again. A major difference between the Stirling engine and conventional piston machines is the method of adding the heat. In the internal combustion engine, output work is achieved when air is compressed at low temperature, fuel is added to the chamber, the mixture is rapidly heated by combustion, and the gases are allowed to expand at the subsequent high temperature. In the Stirling engine, the required heat is added to the working gas inside the engine through a heat exchanger wall. Since this wall has a high heat capacity, it is not possible to rapidly heat and cool the wall. Therefore, in one form of the engine, a second piston called the displacer is added to the machine to move the gas between two stationary chamber volumes called the hot space and the cold space (see Figure 1.1).

Since the alternating heating and cooling would inherently waste large quantities of heat, a matrix type regenerator is placed between the hot and cold sources. Heat is then stored in the regenerator as the gas moves toward the cold space (compression space) and then released as the working fluid returns from the cold space to the hot space (expansion space). The power piston moves in a phased relationship to the displacer piston, alternately compressing and expanding the gas, thereby providing output power.

1.1 IDEAL STIRLING CYCLE DESCRIPTION

The ideal Stirling engine cycle is shown in the pressure-volume (P-V) and temperature-entropy (T-S) diagrams of Figure 1.2.

- a) Process A-B is an isothermal compression where heat is transferred from the working fluid at the minimum cycle temperature to the external sink. (Heat rejection from the engine)

Fig. 1.1 Typical Displacer Stirling Engine

$$\text{Work Out} = (\text{Area Under C-D Curve}) - (\text{Area Under B-A Curve})$$

Figure 1.2 Ideal Stirling Cycle P-V and T-S Diagram

- b) Process B-C is at constant volume where heat is transferred to the working fluid from the regenerator matrix. (Heat released)
- c) Process C-D is an isothermal expansion where heat is transferred to the working fluid at the maximum cycle temperature from the external source. (Heat added to the engine)
- d) Process D-A is at constant volume where heat is transferred from the working fluid to the regenerative matrix. (Heat absorbed)

Figure 1.3 shows the operation of a Stirling engine cycle. To start the cycle, assume the power piston is at its bottom dead center point and the displacer piston is at its top dead center point. All of the working fluid is, therefore, in the compression (cold) space. This starting position corresponds to point A in the P-V and T-S diagrams where the pressure and temperature are at the minimum level and the volume is maximum.

With the displacer piston stationary, the power piston moves toward the top dead center point compressing the working fluid isothermally. Since heat is being taken from the compression space, the temperature remains constant. This corresponds to the process A-B shown in the diagram and to the piston position as shown in Figure 1.3b.

With the power piston remaining stationary in its top dead center position, the displacer piston advances toward the power piston, moving the working fluid from the compression space to the expansion space. Heat is thereby transferred from the regenerative matrix to the fluid and the temperature is increased. The volume remains constant at the minimum cycle level and, therefore, the increase in temperature causes an increase in pressure. The process B-C depicts this motion with the piston positions shown in Figure 1.3c.

The displacer continues to move away from its top dead center point as the expansion process proceeds (shown in Figure 1.3d). Pressure is decreasing as the volume is increased, however, temperature is constant since external heat is being added. This isothermal process is that process shown from C-D.

- (a) Piston at bottom dead center. Displacer at top dead center. All gas in cold space.

- (b) Displacer remaining at top dead center. Piston has compressed gas at lower temperature.

- (c) Piston remaining at top dead center. Displacer has shifted gas through cooler regenerator and heater into hot space.

- (d) Hot gas expanded. Displacer and piston have reached bottom dead center together. With piston stationary, displacer now forces gas through heater, regenerator and cooler into cold space, thus re-attaining situation (a) above.

Fig. 1.3 Typical Stirling Engine Operation

(Taken from Meijer, R.J., "The Philips Stirling Engine", DeIngenieur, Jrg. 81, NR 18, 2 Mei 1969, pp. W69-W79, W81-W93.)

The final portion of the cycle is completed as both pistons are moving to transfer the working fluid through the regenerator matrix to the compression space, thus releasing heat to the matrix. The working fluid temperature and pressure is decreased as the volume is maintained constant. Again, this is shown in the P-V and T-S as process D-A.

If the heat that is transferred to the working fluid from the regenerator matrix is the same as that transferred from the fluid to the matrix, then only the external heat transfer processes remain and the efficiency is consistent with the Carnot cycle efficiency. The advantage of the Stirling engine cycle is that the two isentropic processes of the Carnot cycle are replaced by two constant volume processes, thereby increasing the area under the P-V diagram reflecting in higher specific work output levels without resorting to very high pressures and high swept volumes.

1.2 DEPARTURES FROM THE IDEAL CYCLE

The ideal cycle described assumes that the processes are thermodynamically reversible. That is, the expansion and compression processes are isothermal and that infinite heat rates exist in addition to infinite heat capacities. In the ideal analysis, the effects of regenerator matrix voids, clearance spaces and cylinder pockets were neglected. In addition, the pistons were assumed to move in a discontinuous manner whereas in reality the motion is a smooth continuous path. Therefore, in reality, the theoretical P-V and T-S diagrams are rounded off. Aerodynamic and mechanical losses were also neglected. Inclusion of these losses, of course, result in a lower net cycle output power and lower efficiency.

The addition of the heater and cooler components change the real heat transfer to a more adiabatic situation rather than the assumed isothermal processes. Penalties in additional aerodynamic flow losses and increased dead volume result. The use of practical equipment imposes one additional reality; that the fluid not only is heated as it flows to the expansion space, but also as it flows in the reverse direction from the expansion space to the cooler. The cooling process is also penalized in this manner as well.

1.3 FREE PISTON STIRLING ENGINE

1.3.1 Introduction

The Free Piston Stirling Engine (FPSE) invented by William Beale is a thermally driven mechanical oscillator operating on a Stirling engine cycle, deriving power from heat flow between a source and a sink. The desired displacer motion in the FPSE is produced by gas forces rather than kinematic linkages as in the case of most commonly known Stirling engines. The Beale-FPSE is uniquely suited to driving direct coupled reciprocating loads, such as linear generators of the type proposed for the Power Conversion System, in a hermetically sealed configuration and without the need for high pressure shaft seals or contaminating lubricants.

In one possible system arrangement, two identical engines are mounted such that the reciprocating members are driven in phase opposition. This design is under consideration for the 2 KWE power conversion system for ERDA. The purpose of this configuration is to essentially eliminate the large vibratory forces that might be transmitted to a vehicle or spacecraft due to reciprocating motion of the Stirling engine's pistons.

1.3.2 Operational Description

The basic engine consists of a heavy piston (power piston) and a light-weight piston (displacer) in a sealed cylinder. The displacer rod passes through the power piston and is in communication with the bounce space (that gas volume that exists below the power piston). The bounce space is essentially a constant pressure chamber. Figure 1.4 gives the time dependent and displacement variations of the cycle during an engine start.

To start the cycle, it is assumed that the system is at rest, the pressure is equal throughout and the temperature is ambient. Heat is then applied to the expansion space, causing an increase in pressure in the working spaces,

Nomenclature:

A - Displacer
 B - Piston
 C - Displacer Rod
 D - Expansion Space
 E - Compression Space

F - Heater
 G - Regenerator
 H - Cooler
 I - Bounce Space

Fig. 1.4 FPSE Displacer and Piston Relationship
 (Taken from Walker, G., Stirling-Cycle Machines,
 Oxford University Press, Ely House, London, 1973)

that acts on both the power piston and displacer, moving them down the cylinder. The forces and acceleration on the pistons are functions of the exposed piston, displacer and rod areas and their respective masses. If the mass ratio of the piston to the displacer is large, then the displacer will accelerate rapidly, moving the cold fluid from the cold compression space to the hot expansion space. The displacer and the piston will eventually contact each other and move together and as a result there is no further flow of working fluid.

The pressure now starts to decrease in the system as the expansion continues. This expansion continues until the working fluid pressure drops to the level of the bounce space pressure. Power piston inertia is, however, sufficient to continue the expansion so that the working pressure falls below the bounce space pressure at which time the piston and displacer are subjected to decelerating forces. Due to its light mass, the displacer responds first and separates from the piston. The working fluid starts to flow from the expansion space through the regenerator, giving up its heat while the working pressure falls sharply causing the displacer to move up rapidly toward the cylinder top.

Piston movement downward is now stopped and the pressure differential causes the piston to move upward compressing the working fluid until the pressure is again above the bounce space pressure. The displacer then starts to move downward again to contact the piston, starting the cycle over.

SECTION 2.0

PRELIMINARY DEMONSTRATOR ENGINE DESIGN

2.1 PRELIMINARY ENGINE SIZING STUDY

The preliminary design of a demonstrator free piston Stirling engine involves optimization of parameters with conflicting trends. Therefore, an initial sizing study was performed, prior to a final preliminary design of the engine, to establish the sensitivity of the engine to these critical parameters.

2.1.1 Method of Approach

Since the frequency of the power conversion system has a very strong effect on system weight and size, this parameter was studied extensively in the initial work. Preliminary analyses showed that if no auxiliary spring were used, the highest frequency attainable was 50 Hz. A system with this frequency was then studied in greater depth, and it was found that a considerable improvement in cycle efficiency could be made if the requirement for working gas spring effect was reduced.

The engine then was analyzed at 50 Hz without the constraint of spring effect on the assumption that the spring could later be added externally without excessive penalty. When this was done, good cycle efficiency resulted. It then remained to compute gas spring losses, seal leakage, windage, and heat leaks.

In order to assess the above-mentioned losses, a mechanical arrangement had to be decided upon. A hemispherical outer heater surface was considered since initial heater arrangements indicated that this shape may be optimum. With that shape requirement and an estimated surface thermal resistance of $1700 \text{ watts/m}^2 \text{ } ^\circ\text{K}$ ($300 \text{ BTU/ft.}^2 \text{ hr}^\circ\text{F}$) and an assumed contact ΔT of 30°C , the heater dome was sized.

Given the heater dome, it was apparent that the required regenerator, cooler, and displacer could be arranged as shown in Figure 2.1, with the following advantages:

Fig. 2.1 Demonstrator Engine Preliminary Design

- a) The regenerator walls, always a significant heat leak path because they are normally short and thick due to pressure loading, may be made much thinner, since they are only required to sustain the cycle ΔP , rather than the full pressure. Heat leak was, therefore, substantially reduced.
- b) The pressure vessel, which must be thick enough to withstand charge pressure, may be made to have a long heat leak path, with a resultant low heat flow.
- c) The large regenerator and cooler volume required is easily accommodated without increase in envelope.
- d) A large size, low pressure ratio gas spring, may also be accommodated neatly.

The 50 Hz design point performance was then computed from the results of a cycle simulation, and steady state heat transfer estimates based on the geometry chosen.

The decision to use an external gas spring released the engine cycle from frequency constraints. It was then possible to investigate engine design to match the alternator at a higher frequency of 60 Hz, and 20mm stroke. This was done with the same basic configuration that was used with the 50 Hz design.

The system then was analyzed at 60 Hz and better performance was obtained, indicating that the 50 Hz design was sub-optimum. At this point, because of program constraints, it was decided to concentrate on the analysis of the 60 Hz system, and to use scaling factors for 50 Hz and 45 Hz to get an estimate of the trade offs. The system performance calculated in this manner is shown in Table 2.1 for 60 Hz, 50 Hz, and 45 Hz.

Table 2.1 - Sizing Summary
(Power Values in Watts)

Frequency, Hz	<u>60</u>	<u>50</u>	<u>45</u>
Alternator Efficiency	0.925	0.915	0.905
Power to Alternator	1081	1093	1105
Gas Spring Power	60	38	29
Bearing, Windage & Leakage	20	15	15
Total Gas Power	1161	1146	1149
Engine Gross Efficiency (gas power/heat to gas)	0.57	0.58	0.59
Heat to Gas	2037	1975	1947
Conduction Heat Leaks	277	292	315
Total Heat from Source to Engine	2314	2267	2262
Heat to Coolant	1314	1267	1262
Alternator-Engine Mass (kg)	33	37	43

2.1.2 Gas Springs

The gas spring was made as large in diameter and as soft (low spring constant) as allowed by the required engine dynamics and envelope available. This was done in order to reduce hysteresis losses which were estimated from existing gas spring data.

2.1.3 Bearing, Windage, and Seal Losses

Bearing, windage and seal losses are strongly influenced by design detail, and by requirements such as G loading. Since this information was not available, these losses were roughly estimated from measured data on other engines. Fortunately, the losses are small for the configuration investigated and errors do not significantly effect the overall results.

2.1.4 Discussion of Results

As mentioned previously, Table 2.1 does not represent a fully optimized engine, but only a sub-optimization at 60 Hz and 25 bar, and scaled values for the lower frequencies, in which the following assumptions were made:

- a) heat exchanger pumping loss varies as the frequency cubed;
- b) alternator efficiency is a function of frequency and stroke;
- c) gas spring loss varies as a known function of spring constant K where $K \propto m \omega^2$, where m is total reciprocating mass and ω is frequency;
- d) heat leak losses are evaluated as conduction along all paths from hot to cold regions and are proportional to diameter, thickness, and inversely proportional to length.

As can readily be seen from Table 2.1, there is little system efficiency to be gained by going to the lower frequency. This is because the flow losses for the 60 Hz configuration already are quite low and the alternator efficiency

is higher. Minimization of system mass, therefore, became the criteria which led to selection of a 60 Hz design point.

Several factors contribute to the rather high efficiency reflected in Table 2.1 as compared to existing data for similar sized engines:

- a) heat exchanger sizes are large, and pumping losses low;
- b) very low engine pressure ratios result in low cycle hysteresis loss in dead spaces;
- c) gas spring diameter is large, pressure ratio low, and temperature compression swings low;
- d) there are low friction and windage losses in the FPSE relative to crank engines.

2.2 DEMONSTRATOR ENGINE PRELIMINARY OPTIMIZATION

2.2.1 Method of Approach

The major task of the preliminary design effort involved the basic thermodynamic analysis of the internal components. Once this was done and a well integrated set of specifications had been established by computer simulation, it was possible to design the mechanical structure and compute the component heat leaks. Since the thermodynamic analysis is so complex and time consuming, there was little opportunity within the program constraints to go through successive iterations of thermodynamic and mechanical design. The strategy chosen was to allow the thermodynamics to dominate, and to require the mechanical design to accommodate it.

Appropriate adjustments will be made in the final design developed in Phase II to more closely approach an optimum trade between mechanical and thermodynamic considerations.

2.2.2 Description of Engine Configuration

The engine-alternator layout shown in Figure 2.2 is an evolution of the design described in the preliminary sizing study (Section 2.1). The rationale behind the arrangement is to accommodate the required dome shaped heater, the springs, generous internal heat transfer surfaces and low pressure ratio, high efficiency Stirling cycle in a pressurized envelope of minimum thermal leakage.

A particular advantage of the configuration shown is the reduction of the stress requirement of the regenerator walls. This is a very important advantage since the regenerator must be short and of large area in order to minimize pressure drop losses. This leads to a large temperature gradient and, if the walls were thick, a large heat leak.

The engine has a regenerator with a 0.03mm thick skin, on a lightly sintered wire matrix, which is sufficient to resist only the low internal pressure swings. Even so, the regenerator conduction loss is a major one.

In order to oscillate at 60 Hz, the piston-alternator mass required a very stiff spring. Calculation of the loss in the piston gas springs showed a reasonable value for the design. The displacer gas spring was also found to have acceptable loss and was incorporated in the design as shown in Figure 2.2. Engine power and efficiency are tabulated in Table 2.2.

During design no attempt was made to specify methods of joining or other practical construction details which would be necessary in either an experimental machine or in a flight configuration. However, the design does account for what would be reasonable wall thicknesses, taking into consideration the pressures, temperatures, and required life of the engine.

A practical configuration for the demonstrator with appropriate flanges, bolts, instrumentation and other additions will compromise both thermodynamic and mechanical performance of the engine.

PRELIMINARY DESIGN FOR A FREE PISTON STIRLING ENGINE/ALTERNATOR

Figure 2.2

TABLE 2.2

SUMMARY OF ENGINE PERFORMANCE

Hot Wall Temperature	-	770C (1043 K)	}	Carnot Efficiency 71%
Cold Wall Temperature	-	30C (303 K)		
Charge Pressure	-	25 bar		
Operating Frequency	-	60 hz		
Power to Alternator (w)	-	1086 watts		
Parasitic Power Losses	-	142 watts		
Total Piston Power Cycle	-	0.58		
Gross Efficiency				
(Piston Power/Heat To Gas)				
Required Thermal Power	-	2117 watts (t)		
To Gas				
Heat Leaks	-	305 watts (t)		
Total Thermal Power	-	2422 watts (t)		
Input				
Overall Efficiency	-	0.45		
(Power to Alternator/ Total Thermal Power)				

2.2.3 Thermodynamic Component Design

The thermodynamic design of the FPSE was optimized via an algorithm which made heavy use of high speed digital computation. A program, the accuracy of which is to be verified by test data, was employed to predict engine efficiency from various combinations of design parameters. The problem involved finding the best combination of heater, regenerator and cooler, together with the optimal piston-displacer phase relationship, and swept volume ratio, to achieve a highly efficiency one (1) kilowatt output engine.

A numerical approach was necessary due to the multiplicity of conflicting requirements and the complexity of the analysis. For example, in order to achieve good efficiency one must balance such things as: good heat exchanger effectiveness with low flow friction losses, high specific power (to offset parasitic losses) with low gas hysteresis losses, and a finely divided regenerator matrix having large surface area with minimal flow resistance, axial conduction and void volume.

Mathematically, the problem was one of maximizing a function (efficiency) of several variables (heat exchanger dimensions, phase angle, and swept volumes), without access to partial derivatives. Fortunately, the efficiency function was sufficiently well behaved to yield to a simple algorithm involving successive line searches in the coordinate directions until reasonable convergence to a maximum was obtained.

In order to simplify the problem as much as possible, the piston and displacer motions were constrained to be sinusoidal during the optimization. After the optimal amplitudes and phasings were chosen, it was then possible to design a displacer drive scheme to accommodate the desired motion. The displacer drive scheme indicated in the engine drawing will probably require modification because the power output, phasing, and efficiency are very sensitive to changes in drive piston area. More stable schemes have already been devised, but have not as yet been evaluated in the simulation. Component

specifications resulting from the computer simulation are recorded in Table 2.3.

Figure 2.3 shows the effect of heater size on the engine gross efficiency (gas power/heat to gas). The tubes of Figure 2.3 are 120mm long and 2mm in diameter. A engine heater design with 120 tubes was selected for the 60 Hz engine.

Figures 2.4 and 2.5 are computer plots for the 60 Hz engine indicating the shape of the pressure and displacer-piston phase diagrams. It is noteworthy that there is little gas spring component in the working cycle, and that the displacer phase advance over the piston is less than in current designs. These are characteristics of highly efficient, but rather low power engines which are suited for this application. Figures 2.6 to 2.8 display the cyclic variations of displacer and piston position, working gas pressure and various temperatures.

2.2.4 Structural Design

The preliminary material selection was made using the hoop stresses of the various components. Stress-to-rupture data for 100,000 hours at 800°C was utilized when available. Since stress-to-rupture values for Hastelloy were lacking, they were extrapolated using the Larson-Miller equation. Allowances were made for stress concentration and for a degree of safety. Using these values, rupture-fatigue plots were drawn using a linear approximation.

Wall thicknesses required were nearly equal for Inconel X-750, Hastelloy X, or Hastelloy C; while the use of Inconel 713C casting alloy would allow the wall thickness to be much less. Walls were tapered to maintain the 100,000 hour criterion as temperature changed along their length. A summary of the wall thicknesses of selected components using various materials is shown in Table 2.4.

It should be noted that the selection of these materials is very preliminary

TABLE 2.3
ENGINE COMPONENT SPECIFICATIONS

A. Heater

Loss in bends and entry, exit	- 2 q	Tube equivalent
Length	- 12 cm	127 tubes, 2.07 mm dia.
Wetted perimeter	- 83 cm	12 cm long
Area	- 4.3 cm ²	
Pressure drop loss	- 25 watts	

B. Cooler

Loss in bends and entry, exit	- 2 q	Tube equivalent
Length	- 4 cm	537 tubes, 0.8 mm dia.
Wetter perimeter	- 1.35 meters	4 cm long
Area	- 2.7 cm ²	
Pressure drop loss	- 11 watts	

C. Regenerator

Type	- random fibers (stainless steel)
Flow length	- 2 cm
Mass fiber	- 200 grams
Total specific heat	- 92 joules/ ^o K
Fiber diameter	- .022 mm
Total volume (30% fiber volume, 70% void volume)	- 86 cm ³
Pressure drop loss	- 134 watts
Enthalpy flux loss	- 130 watts

D. Piston and Displacer

	Piston	Displacer	Displacer Rod
Mass	3.7 kg (including alternator)	0.2 kg	--
Diameter	5.35 cm	5.35 cm	0.6 cm

Table 2.3

(Continued)

E. Piston-Alternator Gas Spring

Volume	- 2340 cm ³
Piston area	- 154 cm ²
Total surface area	- 1500 cm ²
Stiffness (k)	- 4200 N/cm
Cyclic loss	- 76 watts

F. Bounce Volume Gas Spring

Volume	- 10,000 cc
Piston area	- 177 cm ²
Total surface area	- 8,000 cm ² (estimate)
Stiffness	- 1,300 N/cm
Cyclic loss	- 47 watts

G. Displacer Gas Spring

Volume	- 417 cm ³
Piston area	- 13.1 cm ²
Total surface area	- 273 cm ²
Stiffness	- 182 N/cm
Cyclic loss	- 7 watts

Fig. 2.3 Effect of Number of Heater Tubes on Engine Efficiency

Figure 2.4 Pressure vs Piston Position

12
[0.0 CM]

3.50E-02
[3.50 CM]

3.00E-02
[3.00 CM]

2.50E-02
[2.50 CM]

[METERS]

2.00E-02
[2.00 CM]

1.50E-02
[1.50 CM]

1.00E-02
[1.00 CM]

5.00E-03
[.5 CM]

DISPLACER POSITION VS PISTON POSITION

PRESSURE - 25 BAR
FREQUENCY - 60 HZ
POWER OUTPUT - 1150 WATTS
T_{HOT} - 1643°K
T_{COLD} - 303°K

← PISTON MID-STROKE

-80cm -60cm -40cm -20cm 0.0 20cm 40cm 60cm 80cm

-1.00E-02 -7.89E-03 -5.71E-03 -3.56E-03 -1.41E-03 .33E-03 2.89E-03 5.03E-03 7.17E-03 9.32E-03 1.15E-02

PISTON POS

Figure 2.5 Displacer Position vs Piston Position

Figure 2.6 Position vs Time

Figure 2.7 Pressure Vs Time

1.05E 03
9.00E 02
7.50E 02
6.00E 02
4.50E 02
3.00E 02
1.50E 02
0.0

0.000 0.005 0.010 0.015 0.020 0.025 0.030 0.035 0.040

1.51E 00 1.51E 00 1.52E 00 1.52E 00 1.53E 00 1.53E 00 1.54E 00 1.54E 00 1.55E 00 1.55E 00

Figure 2.8 Temperature vs Time

TABLE 2.4

SPECIFICATION OF PRESSURE WALLS

(100,000 Hour Desired Life)

<u>Component</u>	<u>Temp.</u>	<u>Material</u>	<u>Allowable Mean Stress Value</u>	<u>Thickness</u>	<u>Comments</u>
Hot-End Vessel Head	800°C	Inconel "713C"	93.0 M Pa	1.27mm	
		Inconel X750	25.2 M Pa	4.10mm	
		Hastalloy (X or C)	27.8 M Pa	4.00mm	
Hot-End Vessel Wall	800°C	Inconel "713C"	93.0 M Pa	2.54mm	
		Inconel X750	25.2 M Pa	8.20mm	
		Hastalloy (X or C)	27.8 M Pa	8.00mm	
Cold-End of Smaller Diameter Pressure Vessel Section		Inconel "713C"		1.30mm	This thickness was selected as a minimum value for this section. Each of these materials could withstand the stress at a temperature of ~500°C.
		Inconel X750		1.30mm	
		Hastalloy (X or C)		1.30mm	
Displacer & Displacer Cylinder	700°C	Inconel "713C"	345M Pa/used as the oscillating allowable stress (No mean stress exists)	0.24mm	
		Inconel X750		0.24mm	
		Hastalloy (X or C) 316 S.S.		9.24mm	
Large Diameter Section of Pressure Vessel, Including Torispherical Head		Inconel "713C"		1.90mm	This thickness was the assumed minimum for this section; again each of these materials could withstand the stress at a temperature of ~500°C.
		Inconel X750		1.90mm	
		Hastalloy (X or C) 316 S.S.		1.90mm	

and should only be viewed as such. During the Phase II effort, significant work to coordinate the material requirements with the potential application requirements will be conducted. Studies in conjunction with the appropriate government organizations is planned to insure proper material selection.

2.2.5 Gas Bearings

Power requirements for the gas bearings are shown in Table 2.5 and were computed by the methods of J. W. Powell ("Design of Aerostatic Bearings"-Machinery Publishing Company Ltd., 1970). The assumed loading was 20 g's for both engine piston-alternator and for the displacer.

2.2.6 Gas Springs

The stiffness required of the gas springs was derived from the dynamics of the engine in the computer program and diameters and volumes were chosen to fit the geometry. Spring losses were then computed with an analysis which considered pressure ratio, surface to volume ratio, spring constant, diameter, and stroke as parameters.

2.2.7 Heat Leaks

The thermodynamic design and structural analysis provided the required geometry from which the heat leaks through conduction paths from hot to cold regions of the engine could be estimated.

The displacer, displacer cylinder and regenerator directly couple the hot and cold regions via conduction paths. Further, the displacer shuttles heat from hot to cold portions of its cylinder as a result of its axial motion.

The regenerator was modeled as an annular ring of helium partially filled with noncontacting metal fibers, each of which was normal to a gas flow and of a uniform temperature. The gas was considered to be the heat conduction path, with the metal fibers having the effect of shortening this conduction

TABLE 2.5

SPECIFICATIONS OF GAS BEARINGS

<u>Location</u>	<u>Assumed Load (20g)</u>	<u>Diameter</u>	<u>Length/Dia. Ratio</u>	<u>Supply Pressure</u>	<u>Clearance (Diametral)</u>	<u>Required Flow</u>	<u>Power Requirement</u>
Displacer	39.2 nt	5.37cm	0.28	55 Bar	7.6×10^{-3} mm	9.6×10^{-3} gm/sec	7.5 watts
Piston	725.2 nt	5.37cm	1.00	55 Bar	7.6×10^{-3} mm	2.7×10^{-3} gm/sec	2.1 watts

path length. This model was found to give results consistent with published data for thermal conductivity of porous metal foams and layers of metal screens.

The shuttle heat transfer was estimated by the methods given by Rios. (ASME Paper #70-WA/Ener-3).

The heater head insulation heat leak was estimated by the assumption of an equivalent length and area of stagnant helium connecting the heater inner wall and the coolant jacket.

The pressure vessel heat leak was analyzed as simple conduction, with additional smaller terms added to handle the loss through the pressure wall. It was assumed that the pressure wall would be cooled only at the alternator, and that the heat leak path could be the entire length from the regenerator to the alternator. This would require insulating the pressure wall over that length.

The results of the heat leak analysis are shown in Table 2.6

2.2.8 Conclusions

Table 2.7 is a summary of the parasitic losses while overall engine performance is presented in Table 2.2. From this data, it can be seen that the preliminary analysis shows that a 60 Hz, engine incorporating external gas springs and the component characteristics described, is expected to have a high thermal efficiency.

TABLE 2.6

CONDUCTION HEAT LEAKS

<u>Component</u>	<u>Material</u>	<u>Thickness (cm)</u>	<u>Length (cm)</u>	<u>Thermal Leak (watts)</u>
Pressure Vessel	Inco X713C	Hot - .25 Cold - .13	Variable 30	80
Displacer Cylinder	Inco X713C	.0235	8	8
Displacer (Includes radiation)	316 Stainless	.024	9	8
Regenerator Wall	316 Stainless	.003	2	20
Regenerator Matrix	316 Stainless + Helium	.0015	2	84
Displacer Shuttle	-	.05cm gap	-	79
Internal Insulation Heater Head	-	Variable	-	26

TABLE 2.7

SUMMARY OF PARASITIC LOSSES

A. Heat Leaks

<u>Component</u>	<u>Leak (watts)</u>
Regenerator	104
Pressure Vessel	80
Displacer	
Shuttle	79
Conduction	8
Head Insulation	26
Displacer Cylinder	<u>8</u>
Total Heat Leak	305 Watts Thermal

B. Power Losses

<u>Component</u>	<u>Loss (watts)</u>
Gas Springs	76
Piston-Alternator	
Bounce Volume	47
Displacer	7.6
Gas Bearings	
Piston-alternator	4
Displacer	<u>7</u>
Total Power Loss	142

SECTION 3.0

DESIGN AND TEST OF A LINEAR ALTERNATOR

3.1 INTRODUCTION

An electrodynamic linear alternator (EDLA) which would be driven by the Free Piston Stirling Engine was designed, fabricated, and tested under this program. The alternator was designed to be operated in an environment that simulated actual operating conditions. Therefore, the alternator design included a complete laboratory test system which permits operation under controlled conditions with instrumentation for monitoring and evaluation. All components of the system were designed with major emphasis upon simplicity and durability to ensure trouble-free testing and an accurate appraisal of the alternator electrical performance.

3.2 ELECTRODYNAMIC LINEAR ALTERNATOR

The alternator demonstration unit, while operating on the same principle as other electrodynamic linear machines designed by MTL, is unique in its design concept and construction. Figure 3.1 is an assembly drawing of this alternator in a test fixture. It utilizes a two-piece armature consisting of a hollow reciprocating member and an inner stationary core, rather than the conventional solid armature. Accordingly, the planned testing includes a separate evaluation of the conventional design, as well as the testing of the demonstration unit and its complete system. This additional effort will provide the opportunity to compare the performance capability of the demonstration unit to that of the conventional design.

The balance of this section is devoted to a discussion of the design aspects related to the demonstration unit only.

3.2.1 Principle of Operation

The linear alternator represents a unique application of the basic principles

Fig. 3.1 Linear Alternator Test Assembly

of electromagnetic induction in which the induced voltage (e) is a function of the uniform flux field (B) the conductor length normal to the direction of the flux field (l), and the conductor velocity normal to the flux field (V):

$$e = B l V \times 10^{-8}$$

where:

$$\begin{aligned} e &= \text{volts} \\ B &= \text{lines/in}^2 \\ V &= \text{in/sec} \\ l &= \text{in} \end{aligned}$$

Figure 3.2 depicts the stator and armature assembly, as well as the impressed flux field in the extreme positions of armature motion. The center d-c coil when energized creates a constant flux field in the stator armature magnetic circuit. With motion of the armature, an a-c voltage is induced in the two a-c coils by virtue of the alternate linking of these coils by the moving d-c flux field. An additional feature of the design is the incorporation of copper compensating coils in each pole face of the reciprocating armature. These coils act as shorted single-turn conductors which suppress the associated a-c coil reactance for improved operating power factor and efficiency.

3.2.2 Initial Design Analysis

An extensive study was conducted to identify the interaction of various design parameters for a 1.0 KW alternator. Frequency, stroke, and electromagnetic and electrical characteristics were varied, and their effect on efficiency, armature weight and total weight was evaluated. The results are shown in Figures 3.3 to 3.8. This initial parametric comparison was made with the MTI alternator design computer program which determines performance capability as well as dimensional geometry.

Based on the studies described in Section 2.0, a design point of 60 hertz

Fig. 3.2 MTI Linear Alternator Configuration

Fig. 3.3 DC to AC Ampere Turns Ratio Trends

Fig. 3.4 Air Gap Trends

Fig. 3.5 Current Density Trends

Fig. 3.6 Packing Factor Trends

Fig. 3.7 Compensating Coil Area Ratio Trends

Fig. 3.8 Alternator Design Point Trends

frequency and .8 inches stroke was selected for the alternator. This preliminary design point study resulted in an approximate efficiency of 92.4 percent with an armature weight of 7.2 pounds. As shown in the figures, the corresponding design parameters were as follows:

DC/AC Ampere Turns Ratio	- .75
Stator Bore Flux Gap	- 35 mils
AC Coil Current Density	- 900 amps/sq. inch
AC Coil Packing Factor	- 6.5
Compensating Coil/AC Coil Area Ratio	- .25

3.2.3 Demonstration Unit Performance

The initial alternator configuration underwent several design iterations before being finalized for the demonstration unit. These were prompted by the following:

1. Improvements in alternator computer program sophistication to reflect the actual operation of linear motor hardware being tested in other parallel and complementary programs.
2. Improvements in the hydrostatic gas bearing and supply compressor designs to ensure reliable operation of these components.

The final alternator configuration for the demonstration unit has the following design parameters:

DC/AC Ampere Turns Ratio	- .65
Stator Bore Flux Gap	- 40 mils
AC Coil Current Density	- 885 amps/sq. inch
AC Coil Packing Factor	- .50
Compensating Coil/AC Area Ratio	- .24

The operating efficiency is slightly lower than that of the initial design, and the reciprocating armature weight is increased. The complete computer program

output data is depicted in Appendix I with a definition of terms. The overall performance capability is as follows:

Output Power	- 1 KW
Armature Stroke	- .80 in
Frequency	- 60 cps
AC Voltage	- 220 volts
AC Current	- 4.55 amp RMS
DC Voltage	- 67 volts
DC Current	- .35 amp
Capacitance Corrention (for 1.0 power factor)	- 36.5 microfarads
Operating Efficiency	- 91%
AC Coil Power Loss (each)	- 16.5 watts
DC Coil Power Loss	- 23.7 watts
Compensating Coil Power Loss (each)	- 14.1 watts
Armature Weight	- 8.2 lbs.

3.2.4 Design Concept

The EDLA configuration employed in this machine replaces the standard solid armature with a hollow reciprocating shell and an inner stationary core. Lines of flux generated by the stator DC coil flow through the stator in an identical manner as previous linear machines. The flux, however, separates within the hollow member after crossing the main flux gap at the stator bore. Some DC flux crosses a second gap between the hollow shell and solid core, while the remaining flux travels through the hollow shell (see Figure 3.9). Accordingly, the DC flux density may be maintained at the same level as with a solid armature. The main flux gap at the stator bore is reduced in the area of the pole face carrying the flux which must cross the second gap into the stationary core. In this way, the DC coil ampere-turns are held at the same level with a hollow armature and two gaps, as with a solid armature and the single gap. The hollow armature design provides a 33 percent reduction in reciprocating mass.

Figure 3.9 Two-Piece Armature Configuration

3.2.5 Side Pull Force Analysis

In all electromagnetic devices, a side pull force, normal to the axis of the machine, is created in the event that the armature becomes eccentric with the stator bore. This force acts in the direction of the eccentricity attempting to decenter the armature further. The unbalanced force is produced by an unsymmetrical flux density in the gap around the armature, between the stator bore. The magnitude of the side pull force may be evaluated using the following expression:

$$SPF = 7.056 (10)^{-8} \cdot \pi \cdot (Ni)^2 \cdot R \cdot \ell \cdot \frac{e}{C^3}$$

where:

- SPF = Side pull force (lbs)
- Ni = Nominal ampere turns in the flux gap (amp-turns)
- R = Mean gap radius (in)
- ℓ = Gap length (in)
- e = radial eccentricity (in)
- C = Nominal radial gap (in)

With the hollow armature configuration and a second flux gap, the magnitude of a possible side pull force is inherently greater than with a solid armature, and the same degree of eccentricity.

The main flux gap at the stator bore was increased from 35 mils, which was used in the initial design analysis, to 40 mils for the final demonstration unit to minimize the side pull force levels with a slight decrease in efficiency. This dimension was reduced in the area of the pole face carrying flux which would cross the second gap, such that the reduced main flux gap and stationary core gap totaled 40 mils. Side pull force calculations were made for various gap combinations to determine the minimum total force situation. A main flux gap of 22 mils and a stationary core gap of 18 mils produced this minimum situation (see Figure 3.10). All calculations were made with an assumed eccentricity of 1.3 mils for the main gap and .8 mils for the stationary core gap.

Figure 3.10 Side Pull Force Summary

The side pull force, compared to the eccentricity which produced it, may be observed as a radial magnetic stiffness. In addition, should the armature eccentricity be different at one pole than at the other, a magnetic moment would be created which could be related to an angular stiffness. The alternator decentering stiffnesses were used to initially size the gas bearing design. The resultant maximum radial magnetic stiffness of the hollow armature design is 19,000 lbs/in. The maximum magnetic angular stiffness with the hollow armature design is 67,000 in-lbs/radian.

3.2.6 Stator Construction

The stator design for this alternator is quite similar to conventional stators used in other electrodynamic linear machines. It consists of 637 laminations (18.5 mils thick) arranged to form a solid cylinder at the stator bore. The laminations fan out radially from this bore to the stator outer diameter. They are bent in such a way as to establish a uniform radial display of laminations and provide structural integrity to the complete assembly (see Figure 3.11).

Figure 3.11 Stator Lamination Assembly

The laminations are stamped from M15 fully-processed, non-oriented magnetic electrical steel. This material is a high silicon steel (3 percent vs .1 percent for low carbon steel) which accounts for its outstanding magnetic properties. It has the lowest core loss limit of the commercially available non-oriented lamination steels - 5.8 watts loss per pound of steel at 60 hertz frequency with a flux density level of 10 kilogausses.

The stator is constructed by first assembling the individual laminations into two separate, approximately 180 degree stator halves. This is accomplished with a special fixture which positions the laminations, and then permits them to be welded into a structurally sound subassembly. The two halves are then assembled around a molded DC coil. Steel rings on each end of the stator are welded to the two halves forming a complete stator body. The AC coils are then internally wound into mylar lined slots in the stator body, and the entire assembly is potted with epoxy.

The DC coil consists of 4653 turns of 23 gage electrical copper wire (.023 diameter). The wire is processed before winding with an approximately 1 mil thick coating of a laquer-type insulation. Stranded wire leads and terminal connections are fixed to the coil after winding. The coil is then placed in a mold and potted with an epoxy material to form a solid assembly, preventing any possible relative movement of the coil turns.

Each AC coil consists of 240 turns of 12 gauge electrical copper wire (.081 inches diameter). The wire insulation, leads and terminal connections, are handled the same way as described for the wire used in the DC coil.

3.2.7 Armature Construction

The two-piece armature design as discussed previously is much different than the standard solid plunger configuration. In addition, the alternator gas bearing has been incorporated into the gap between the hollow reciprocating member and the stationary core.

The hollow reciprocating member embodies the conventional low carbon steel armature pole face pieces and solid copper compensating coils. These parts are assembled with interference fits, consistent with material properties, and substantially capable of transmitting inertia loads.

The stationary core member is a welded low carbon steel cantilever beam structure, supported by a top plate to the alternator stator body. The structure is capable of transmitting the gas bearing reaction load to the stator with less than 70 microinches maximum deflection over its entire length. A radial array of tapered laminations, made from the same electrical steel as the stator laminations, is located along the beam in line with the a-c coils. These laminations minimize eddy current losses in this area, and assure a constant DC flux level in the solid steel center portion of the stationary member. The center passage in this member forms the supply volume which feeds the gas bearing discharge orifices.

3.3 TEST SYSTEM DESCRIPTION

The test system employs the MTI vibration table and a previously-designed motion amplifier apparatus to reciprocate the alternator at 60 hertz. The alternator is enclosed by a test chamber dome which permits ambient pressurization with helium at the rated level of 25 bars (362.5 psia). The motion amplifier output shaft drives the alternator armature vertically through a rod seal assembly which is required to maintain ambient pressurization. As mentioned previously, a gas bearing was incorporated into the design to guide the armature and react any electromagnetic sidepull forces. It is supplied by an integral positive displacement compressor formed by the hollow reciprocating armature shaft and the stationary armature core. Figure 3.12 depicts a layout of the test machine from the motion amplifier interface.

The helium circuit through the machine is superimposed on Figure 3.12. It was designed to conduct the heat generated by the stator coils and armature compensating coils to the external surfaces of the test chamber dome, base-plate, and seal housing. The power loss of these components is so minimal that free convection from external surfaces is sufficient to dissipate the heat.

As the armature moves upward, helium flows through the compressor discharge ports into the gas bearing chamber. The supply chamber feeds the two planes of fixed orifices. The gas discharges from the top and bottom of the bearing into the annuli formed by the stationary core laminations. Helium from the bottom discharge annulus passes across the armature pole faces, and joins with the gas leaving the top discharge annulus. The total helium flow then passes across the stator outer diameter along the test chamber wall, and is transported by four tubes to the seal housing suction annulus. As the armature moves downward, the helium flows through the suction ports into the compressor.

Some mechanical design compromises were made in the design of the alternator, gas bearing, and supply compressor to ensure that the overall objective of

Fig. 3.12 Linear Alternator Test Layout

trouble-free testing was met. These trade-offs were justified for the following reasons:

1. The major purpose of this specific unit was to verify the electrical performance predictions of the MTI computer codes. In addition, the design would be optimized prior to incorporation into the demonstrator energy conversion system.
2. The gas bearing, supply compressor, and rod seal assembly were designed specifically for this laboratory test system and do not reflect the requirements of the demonstrator Free Piston Stirling Engine installation.

This section includes a description of the test setup and a discussion of the hydrostatic gas bearing, supply compressor, drive shaft seal, and thermal analysis.

3.3.1 Hydrostatic Gas Bearing

A hydrostatic gas bearing was incorporated into the test system to react lateral vibration forces generated by the motion amplifier apparatus and any possible magnetic side pull forces resulting from an eccentricity between the alternator armature and stator. The package size and complexity was minimized by locating a single gas bearing in the gap between the hollow reciprocating armature and stationary inner core, in line with the alternator to minimize moment loading. The alternator radial and angular magnetic stiffnesses were used with an ample safety factor to initially size the bearing.

3.3.1.1 Principle of Operations

The gas bearing derives its load capacity from the integrated film pressure in the gap between the reciprocating and stationary members. Externally pressurized gas is supplied to the bearing gap through an entry fixed orifice. The bearing gap acts as a second variable restriction in series with the first fixed restriction. As the gap thickness increases, the resistance to gas flow in the gap decreases, and accordingly, gas flow increases. The increased gas flow through the first fixed orifice results in a larger pressure drop there, and thus a lower

pressure level in the gas film gap. As external load is applied reducing the gap and increasing the second variable restriction, gas flow decreases. The resulting lower pressure drop through the entry fixed orifice raises the pressure level in the gas film gap. Proper sizing of the entry orifices in conjunction with bearing dimensions, supply and discharge pressure, gas properties, etc., produces an extremely stable, positive stiffness bearing system.

3.3.1.2 Bearing Design

The hydrostatic bearing design consists of a 1.625 inch nominal diameter with a length of 2.75 inches. The operating radial clearance is .6-.9 mils. Two gas admission planes are used to provide increased bearing stiffness - particularly angular stiffness. Each plane has eight 24 mil diameter entry orifices. The bearing supply pressure is 525 psia which discharges to the test chamber ambient pressure of 362.5 psia.

The bearing provides a radial stiffness of 283,000 lbs/in., and an angular stiffness of 188,000 lbs/radian. This capability far exceeds the alternator magnetic decentering stiffness, and accordingly will ensure trouble-free operation of the test system. The margin of safety, however, represents a compromise in bearing performance due to the higher supply pressure required to maintain it. Furthermore, the bearing supply compressor must operate at a higher pressure ratio and deliver greater flow. The gas flow required by the bearing and the shear friction power loss are .11 lbs/min and .4 watts, respectively.

3.3.1.3 Bearing Fabrication

Both the inner surface of the hollow reciprocating armature and the outer surface of the stationary core are plasma spray coated with chrome oxide, and then ground and lapped to proper dimensions. The resulting surfaces are extremely hard (Rc 65 or greater), wear-resistant, corrosion and erosion resistant, non-magnetic, and offer excellent sliding capability. Laboratory tests at MTI and successful operation in the field have confirmed the superiority of this surface material combination.

The hollow armature is processed with 1-2 mils thick Nickel-Chrome Bond coat followed by 6-8 mils thick Chrome Oxide. Finish grinding dimensions are controlled to maintain a final chrome oxide coating thickness of 2-4 mils thickness per side.

The stationary core was processed differently than the hollow armature to obtain a much thicker final oxide coating. The thicker oxide coating was required to produce the same magnetic gap in the bearing area, as was physically provided in the discharge annuli on either side of the bearing. The solid core was first sprayed with 2-3 mils thick Nickel Chrome Aluminide Bond coat, followed by 14-16 mils thick chrome oxide. Finish grinding dimensions are controlled to maintain a final chrome oxide coating thickness of 10-12 mils per side.

The bearing orifices were machined in the stationary core after the coating had been applied and final grinding is performed. This was accomplished by installing plugs with a suitable press fit in the orifice supply passage such that they protruded slightly above the surface of the base material. Final grinding of the coated bearing surface removed the chrome oxide coating on the plugs leaving a small area which was easily and accurately machined.

3.3.2 Gas Bearing Supply Compressor

The gas bearing supply compressor was designed specifically for the EDLA test system with features that would ensure trouble-free operation. As mentioned previously, a performance penalty in the form of low operating efficiency was accepted for simplicity and durability. The two major sources of inefficiency were the relatively high pressure ratio, and the unusually large piston-to-cylinder radial clearance. These characteristics were identified as pertinent only to the demonstration unit test system, and not the Stirling engine installation.

3.3.2.1 Compressor Design

The compressor is formed by the stationary armature core and hollow armature drive shaft. This arrangement is different from standard compressor designs in that the piston (armature core) remains stationary, and the cylinder (hollow)

armature shell) reciprocates. Helium enters the compression chamber through the suction ports in the armature as it moves downward, and the ports are uncovered by the suction annulus in the seal housing. There are six 125 mil diameter ports which begin to open 16 mils below the mid-stroke position. The ports are symmetrically located on the 1.812 diameter of the armature.

In a similar manner, helium leaves the compression chamber as the armature moves upward and uncovers the discharge ports in the stationary core member. There are six 94 mil diameter ports which begin to open 64 mils above the mid-stroke position. These ports are symmetrically located on the 1.618 diameter of the stationary armature core.

Reed valves, which are normally used to seal the suction and discharge ports, were eliminated from the design (an unnecessary complexity and potential source of failure). In their place, the ports are sealed by the fit of the armature to the seal housing and stationary armature core. This method, however, does not provide a perfect seal since a small clearance is necessary between these parts. The radial clearance in the area of the ports was established on the basis of the operating clearance of the gas bearing. This was done to assure that a close fit in the porting areas did not interfere with the operation of the gas bearing. Accordingly, the optimum clearance for minimum leakage was increased, causing a degradation in compressor performance. The radial clearance in the area of the suction and discharge ports is as follows:

Suction Port Clearance - 1.25-1.75 mils

Discharge Port Clearance - 1.15-1.45 mils

The radial clearance between the armature core and plunger also regulates the leakage to the test chamber ambient, and significantly affects the compressor performance. Piston ring seals are normally used to restrict this leakage for optimum compressor performance. As mentioned previously, performance was sacrificed for simplicity and durability. Accordingly, no seals are incorporated in this application. In their place, the fit of the stationary core to the hollow armature restricts leakage from the compression chamber. The radial clearance is the same as for the discharge ports.

The compressor was initially sized for approximately five times the calculated helium flow requirement of the gas bearing at the calculated pressure rise. From the starting point, the compressor output can be varied to exactly satisfy the flow and pressure requirements of the gas bearing. An external connection to the gas bearing supply volume was provided to monitor the pressure and bypass excess flow during preliminary testing. The compressor output may be adjusted with a separate spacer in the compression chamber, and replaceable shims establish the engagement of the suction and discharge ports.

3.3.2.2 Compressor Performance

The compressor performance was evaluated using MTI's PROCOMP computer code, and input data was revised to simulate this type of ported compressor. The overall compressor performance predicted is as follows:

Suction pressure	- 367 psia
Discharge	- 525 psia
Output flow	- 1.636 cfm
Average gas temperature	- 144°F
Gas discharge temperature	- 166°F
Input power	- 273 watts

3.3.3 Drive Shaft Seal

During laboratory operation the electrodynamic linear alternator will be mounted within a helium-pressurized test chamber, and maintained at 362.5 PSIA to simulate the final product environment. The drive shaft reciprocating motion, which is generated externally by the motion amplifier apparatus, must penetrate the test chamber; and accordingly, some type of drive shaft seal is required.

3.3.3.1 Seal Design

The seal design incorporated into the test system consists of three separate rod seal assemblies arranged in a manner similar to labyrinth seals (see Figures 3.12 and 3.13). The configuration minimizes friction, thereby reducing element wear and precluding the need for laboratory overhauls. In addition, helium leakage from the pressurized test chamber is much less than with a similar labyrinth seal. Each seal assembly includes a split pressure-balanced inner sealing member, and an outer split tensioning member. The elements were designed and will be supplied by Koppers Company, Inc. The inner member material is Koppers K-30M Bronze Filled PTFE, which offers excellent wear resistance, low drag and breakaway friction, and exceptional thermal stability without lubrication. The ring is manufactured with a step-cut joint to facilitate the assembly procedure, and relieve the need for extreme dimensional tolerance control. The outer member material is Kopper K-Iron, which is an unalloyed grey cast iron. It serves to not only seal the inner member step-cut joint, but also position the inner member against the reciprocating shaft. The seal assemblies are radially pressure balanced by the 15-25 mil slots located around the bore of the inner member. In addition, outer member tension is strictly controlled by manufacturing techniques to only position the inner member, while applying no appreciable radial force.

3.3.3.2 Seal Performance

The seal performance may be established by evaluating the operating PV factor, friction power loss, and leakage. An analysis was made using values recommended by Koppers for the coefficient of friction (.05 - .10) and the effective operating radial clearance (.1 mils). The results are as follows:

PV factor	- 41,500 lbs-ft./in ² - sec.
friction power loss	- 15 watts
leakage	- .2 CFM

Figure 3.13 Seal Detail Drawing

3.3.4 Test Set-Up

The EDLA test system was designed to facilitate the assembly and alignment efforts in the laboratory. In addition, the choice and location of all instrumentation was carefully reviewed to assure that the alternator could be accurately evaluated for performance.

3.3.4.1 Alignment Procedure

The task of aligning the armature to the stator and stationary core is extremely critical to the operation of this machine due to the side pull forces resulting from eccentricity, and the relatively close operating clearance of the gas bearing. The following procedure was established to accomplish this task:

- 1) From Quality Control inspection data on subsequent measurements of the related components determine the required nominal thickness of the discharge port-locating shim. The discharge ports should begin to uncover as the armature moves upward 64 mils past its mid-stroke position.
- 2) In a similar manner determine the nominal thickness of the suction port-locating shims. The suction ports should begin to engage the suction supply annulus as the armature moves downward 16 mils past its mid-stroke position.
- 3) Adjust the nominal thickness of the discharge port-locating shims as required to align the stationary core parallel to the stator bore.
- 4) Install the baseplate and support structure on the vibration table with the assembled stator and stationary core.
- 5) Install the armature and determine the proper thickness for the shim between the motion amplifier drive shaft and the alternator armature. This is accomplished by exciting the stator AC coils with a low energy AC signal, and observing the effect of magnetic coupling on the DC coil with an oscilloscope. When the armature is at the exact mid-stroke position, the magnetic coupling of the top AC coil will cancel that of the bottom coil, and a null display will be observed.

- 6) Adjust the nominal thickness of the suction port-locating shims as required to align the stator and stationary core assembly parallel to the armature. This is accomplished by first inserting spacers in the gap between the armature top pole face and stator bore to concentricly position these parts. Then, the gas bearing gap is made concentric at the bottom of the armature by adjustment of the suction port shims. The gas bearing capacitance probes, located in this area, permit an accurate measurement of bearing concentricity.
- 7) With the DC coil energized the stationary core and stator may be radially positioned with respect to the armature to effect the least side pull force situation. The single direction load cells are used to measure the side pull reaction forces.
- 8) The seal housing is now concentricly positioned to the armature with its capacitance probes.

3.3.4.2 Instrumentation

The test system design includes substantial instrumentation for evaluating the performance and monitoring the operation of the demonstration alternator and other system components. All instrumentation leads penetrate the test chamber through the base plate. Commercially available Conax feed-thru packing glands are used to provide an acceptable helium-tight interface. The test chamber dome is positioned and secured, after all instrumentation is installed and verified.

The instrumentation included in the design consists of the following:

- 1) Thermocouples are located in the stator coils, the laminations, the gas bearing supply volume, the upper discharge annulus, the compressor suction annulus, and the test chamber.
- 2) Pressures are monitored in the compressor suction annulus, gas bearing supply volume, and test chamber ambient.
- 3) Capacitance probes, specifically designed by MTI for this application, are installed within the armature stationary core member. These measure bearing radial clearance for proper alignment and during actual operation.

- 4) Standard MTI capacitance probes are installed in the seal housing to facilitate the alignment of this component to the drive shaft. In addition, these probes with the gas bearing probes completely define the radial and angular motion of the reciprocating plunger.
- 5) Quartz force ring transducers are mounted between the stator upper weld ring and the armature stationary core support plate. These load cells are used during the alignment procedure to measure side pull force reactions.
- 6) Quartz force ring transducers are also mounted between the stator lower weld ring and the base plate to measure the stator total reaction forces. This signal and the vibration table velocity signal define system input power.

3.3.5 Thermal Analysis

The test system was analyzed thermally to identify the simplest effective means of cooling. The objective of this analysis was not to determine absolute temperatures throughout the machine, but rather to determine conservative maximum temperatures and their locations. The final cooling scheme was devised on the basis of this very conservative analysis, and accordingly, the actual temperatures will be well below those calculated. The thermal model of the system included only the major heat transfer paths and dissipation surfaces. This assumption neglected the relatively high thermal conductivity and heat capacity of helium. In reality, less significant transfer paths will participate in the dissipation of heat producing the expected actual reduction in calculated temperatures. The model also did not include the motion amplifier and all surfaces of the drive shaft below the test machine; as well as the cooling effect of the helium leakage through the drive shaft seal. This simplification was balanced by a proportionate reduction in compressor power loss dissipated through the test machine thermal model.

The power loss associated with the various test machine components is as follows:

Alternator AC Coil (each)	16.5 watts
Alternator DC Coil	23.7 watts
Alternator Compensating Coil (each)	14.1 watts
Gas Bearing Shear Friction	.4 watts
Gas Bearing Supply Compressor	273 watts
Drive Shaft Seal	15 watts

The above power losses were considered heat generation rates at their respective locations in the thermal model. The gas bearing shear friction was ignored and the compressor heat discharged through the model was reduced 25 percent.

The initial thermal results indicated a possibly undesirable temperature in the drive shaft seal area due to the close proximity of the relatively high compressor heat generation rate. Accordingly, the test machine design was modified to include a provision for cooling the drive shaft with shop air ejected from the seal housing. This alteration reduced the calculated temperatures to acceptable levels.

The thermal resistances to ambient from the test chamber dome, base plate, and seal housing are calculated for free convection heat transfer. Other internal resistances are determined in the usual manner for the particular geometric configuration and material thermal conductivity. The complete thermal results are documented in Appendix V. The significant temperatures of the final design are as follows:

AC Coil Temperature	210°F
DC Coil Temperature	216°F
Compensating Coil Temperature	201°F
Compressor Temperature	194°F
Drive Shaft Seal Temperature	137°F

3.4 LINEAR ALTERNATOR ASSEMBLY AND TEST

3.4.1 Introduction

Prior to testing the linear alternator design described in Sections 3.2 and 3.3, the decision was made to test a simpler mechanical design of the same

alternator. This alternator had a solid plunger and would permit verification of electrical performance with a low risk mechanical design. The alternator was assembled and tested and its performance was not as high as predicted. However, a number of potential design improvements have been identified and will be tested in Phase II of the program.

3.4.2 Alternator Assembly

A schematic of the solid plunger test assembly is shown in Figure 3.14. It utilizes the same stator as the hollow plunger design with a solid plunger supported by a cylindrical guide bearing.

The stator assembly is shown in Figure 3.15. It was assembled as described in Section 3.2 with the exception that only 236 turns of wire were able to be wound into the AC slots. The plunger was made out of low carbon steel and contained solid copper compensating coils.

3.4.3 Test Program

The alternator described above was tested to measure power output and efficiency at various operating conditions. The design characteristics of the alternator are summarized in Section 3.2.3 and the testing performed and an analysis of the results is discussed below.

3.4.4 Test Facility

When driven by a Free Piston Stirling Engine, the motion of the plunger of the linear alternator is essentially sinusoidal with respect to the stator. In order to simulate this motion, the test alternator was driven by a mechanically resonant motion amplifier attached to an electrodynamic shaker. Figure 3.16 shows a schematic representation of the motion amplifier test system while Figure 3.17 is an actual photograph of the test set-up.

SOLID PLUNGER LINEAR ALTERNATOR

Figure 3.14

Fig. 3.15 Alternator Stator

Fig. 3.16 Linear Alternator Test System Schematic

Fig. 3.17 Alternator Test Setup

The shaker was excited at the frequency of the electrical line voltage (60 Hz nominal). The motion amplifier component of the test system provided the value of stiffness (k_a) between the plunger and the shaker required to amplify the motion of mass m_p (which includes the mass of the motor plunger) relative to the motion of the shaker platen. The need for mechanical motion amplification arises from the fact that the shaker cannot produce strokes at 60 Hz equal to the required range of alternator plunger stroke.

During testing, the alternator output was dissipated in a resistive load and the current and voltage measured in order to calculate power output.

Stator force cells measured electromagnetic gap force and this force measurement along with a plunger velocity measurement was used to calculate power input. During testing the shake force (f_s in Figure 3.16) was adjusted by the test operator to maintain the desired value of stroke.

3.4.5 Instrumentation

Figure 3.18 is the instrumentation and control schematic for the linear alternator test. The field coil of the alternator was run off a DC power supply and the voltage and current to the coil was measured. A 60 Hz reference signal was provided to the shaker control from the power line. The phase angle control circuit synchronized the shaker table to the reference frequency and the shaker control varied the power input and, therefore, the stroke for a given load. The plunger velocity sensor signal was provided to the power multiplier for power input calculation and to a calibration amplifier to measure the plunger stroke. The stator force cells measured the electromagnetic gap force input to the plunger. The load cell force was summed with the stator acceleration force and provided to the power multiplier. The summed force signal was then multiplied by the velocity signal to calculate power input. Power output was measured by the voltage and current to the load. Output voltage and plunger velocity were also visually displayed on an oscilloscope.

Fig. 3.18 Linear Alternator Instrumentation and Control Schematic

3.4.6 Test Procedures & Data Reduction

The test rig was assembled and checked out at low value of plunger stroke. The instrumentation was calibrated and the performance was measured at selected values of stroke for various values of field current.

The output power was calculated from:

$$P_{out} = \frac{(I_a \times E_a) - (I_f \times E_f)}{1000}$$

where

- P_{out} = Power output - kilowatts
- I_a = RMS value of alternator current - amperes
- E_a = RMS value of alternator voltage - volts
- I_f = Alternator field current - amperes
- E_f = Alternator field voltage - volts

The input power was obtained directly from power meter readings and efficiency calculated from:

$$\eta = \frac{P_{out}}{P_{in}}$$

where

- P_{in} = Measured power input - kilowatts

Figures 3.19 and 3.20 show generator power output and efficiency, for constant values of plunger stroke, as a function of field current. The power output curves show an increase of power with stroke and with field current until saturation effects become important. For higher values of plunger stroke efficiency is essentially constant at around 78 to 80%. The change in shape of the efficiency curves for 0.6 and 0.7 inch strokes is due to the tuning effect of the compensating capacitor.

Table 3.1 provides a comparison of analytically predicted and test performance. The first column lists the design goals while the second shows the measured

Fig. 3.19 ERDA Linear Alternator Power Output

Fig. 3.20 ERDA Linear Alternator Efficiency

TABLE 3.1
 Alternator Performance Comparison At
0.8 Inch Stroke, 60 CPS

	<u>Design</u>	<u>Test</u>	<u>Revised Prediction</u>	<u>Test At Maximum Power</u>
Field Current (AMPS)	.35	.35	.35	.62
AC Voltage (VOLTS)	220.0	172.3	182.1	191.0
AC Current (AMPS)	4.55	3.44	3.65	3.80
Power (KW)	1.000	.570	.641	.658
Efficiency (%)	91	79	86	77

performance at design field current. A comparison of the two columns shows that the generator is performing well below design goals. The design goals were based on the predictions of a computer program which had not been adequately verified by test data. This computer program has since been modified based on data from other tests. The revised program was run and the performance of the actual test hardware was predicted. The revised prediction is shown in the third column. A comparison of this prediction with the actual test data shows a much better agreement, although the measured performance is still low. The test unit was run to maximum power by increasing field current until the output peaked. This occurred at a condition of 0.62 amps of field current. The power output was 658 watts and the efficiency dropped off to 77%.

Although the performance of the generator did not meet expectations, it must be remembered that this is the first linear generator designed and tested at MTI. A number of potential improvements, including design revisions and usage of better materials, have been identified and will be evaluated in Phase II of this program.

SECTION 4.0

TEST OF A MODEL FREE PISTON STIRLING ENGINE

4.1 INTRODUCTION

In order to obtain well documented test data on the performance of a current Free-Piston Stirling engine, a Model IV engine designed and fabricated by Sunpower, Inc., was tested at MTI. Figure 4.1 is a schematic drawing of the test engine. It is a single cylinder piston-displacer engine. The design pressure for this engine with helium working fluid is 40 bar. The engine is heated by resistive heating of tubes which are an integral part of the cylinder head. The regenerator, consisting of an annular section filled with a random array of stainless steel wool, is located between the heater and cooler. The engine cooler is a counter flow water/gas heat exchanger. Loading of the engine was achieved by a dynamometer consisting of a helium compressor and a loop with a flow control valve. The electrical power supplied to the heater is controlled by adjusting a variac to obtain a predetermined heater tube temperature. Force, position, velocity, pressure and temperature instrumentation was built into the engine in order to accurately monitor its operation.

4.2 ENGINE PERFORMANCE EVALUATION METHODS

As shown in Figure 4.2, a number of parameters must be measured in order to evaluate the performance of a FPSE system. In addition to power output, frequency and stroke, other system inputs/outputs must be measured to allow accurate computation of the efficiency, confirmation of the output power from an energy balance, and provide diagnostic information for system improvement.

4.2.1 Heat Input

The heat which powers the engine may be provided by a number of sources; such as an isotope heat source, gas flame, or passing an electric current

Figure 4.1 MODEL FREE-PISTON STIRLING ENGINE

Fig. 4.2 Schematic of Stirling Energy Conversion System

through the heater wall. For basic heat sources other than isotopes, the heat input may be determined by direct measurement. In the case of an electrically heated engine, this is accomplished by measuring the voltage and current into the heater, the power may then be found from:

$$Q'_I = EI$$

where

$$Q'_I = \text{power input}$$

$$E = \text{input voltage}$$

$$I = \text{input current.}$$

This results in a relatively accurate measurement of the total heat input to the system since the losses are confined to heat loss to the ambient and adjacent engine parts, in which case reasonable estimates are possible.

For the demonstrator engine, which will ultimately be fueled with an isotope heat source, an electrical simulator will be used for laboratory testing. For these tests, the method described above will be applicable if a resistance heater is used. The basic objective of this type of test would be to assure that the system is operating as expected prior to the interfacing with an actual nuclear heat source. Another possible measurement approach is one which attempts to independently determine heat input. This secondary measurement could be accomplished by observing the temperature drop across an element of known thermal conductance, the heat input to the engine would then be found from Fourier's conduction law; i.e.,

$$Q'_I = AK \frac{\Delta T}{\Delta \ell}$$

where

$$Q'_I = \text{input heat rate}$$

$$A = \text{area of thermal heat path}$$

$$K = \text{conductance of thermal heat path}$$

ΔT = temperature difference across thermal heat path

l = length of thermal heat path.

Considerable care would be required to keep the accuracy of the calculation within reasonable bounds, since the thermal heat path would likely conform to a spherical head and may, in fact, be the engine head. Also, the temperature distribution over the area of the thermal heat path is not likely to be uniform, and the thermal conductivity of the material may vary somewhat with temperature. A more detailed analysis of the heater requirements for the demonstrator will be conducted in Phase II.

For an engine fueled by a combustible fuel, the heat input may be determined from an energy balance on the fuel/air mixture flowing into and out of the heater.

4.2.2 Power Output

The power developed by the thermodynamic cycle may be measured at several locations in the system, each providing basic information regarding the engine performance. The significant quantities are the thermodynamic cycle work, shaft work, and net system work.

The total work developed by a thermodynamic cycle may be found from integrating the pressure volume diagram of the working space or

$$Q = \oint P \, dV$$

where

Q = work

P = pressure

V = volume.

Figure 4.3 shows the "work flow" in a Free Piston Stirling engine. Several of these "works" may be computed from this basic equation depending on the volume being swept. The total indicated work in the Stirling engine may be found from

$$Q_T = \oint P_H dV_H + \oint P_C dV_C$$

where

Q_T = total indicated work

P_H = pressure in hot space

P_C = cold space pressure

V_H = hot space pressure

V_C = cold space volume.

The displaced volumes can be found from the area and displacement of appropriate pistons (see Figure 4.4) as follows:

$$V_H = A_C(X_C - X_D - L_D)$$

and

$$V_C = (A_C - A_R)(X_D - X_P)$$

where

A_C = cylinder area

A_R = displacer rod area

X_C = cylinder head position

X_D = displacer position

X_P = piston position

L_D = displacer length

The work done by the gas on the power piston and displacer piston (or rod) may be computed separately. The work done on the power piston is delivered to the load (less friction losses) and may be computed from

Fig. 4.3 Work Flow in a FPSE

Fig. 4.4 Schematic of Single Cylinder Free-Piston Stirling Engine

- A_C = Cylinder Area
- A_R = Displacer Rod Area
- $A_P = A_C - A_R$ = Piston Area
- X_C = Cylinder Head Position
- X_H = Displacer Clearance
- X_D = Displacer Position
- X_P = Piston Position
- $X_{D/P}$ = Displacer Relative Position
- L_D = Displacer Length

$$Q_P = \oint_C P_C dV_P = \oint_C P_C A_P dX_P$$

where

Q_P = gas work to the power piston

$A_P = A_C - A_R$ = area of piston head

X_P = displacement of power piston

The work done by the gas on the displacer may be found from

$$Q_D = \oint_V P_V dV_R = \oint_H P_H A_R dX_D$$

where

Q_D = work done by the gas on the displacer.

The work done by the gas on the displacer goes to overcome friction, most goes to pumping the working gas through the heat exchangers and the remainder is transferred to the piston through the coupling spring or by displacer contact with the piston, as shown in Figure 4.3.

The pumping work dissipated may be computed from the pressure across the displacer and the displacement of the area less the rod: i.e.,

$$Q_{DP} = \oint (P_H - P_C) (A_C - A_R) dX_D$$

where

Q_{DP} = displacer pumping work.

The displacer pumping work will provide a check on the heat exchanger design. Since the total work done on the displacer should be slightly greater than the pumping work, the difference may be used for estimating the friction losses of the displacer.

Although the indicated work found above may be used as a close approximation of the shaft work, a direct measurement should be made. This may be determined from the cyclic integral of the output shaft force displacement diagram; i.e.,

$$Q_s = \int F dX_p,$$

where

Q_s = shaft work

F = force developed in shaft

X_p = displacement of piston.

This work should be slightly smaller than the indicated work computed above due to piston-cylinder friction.

The power corresponding to the above work integrals may be found from the product of the work and system frequency; i.e.,

$$Q' = Qf$$

where

Q' = power

f = system frequency.

However, it may be easier and more accurate to measure the power directly as can be done for the shaft power. In this case, the above work integrals may be transformed to power integrals as follows:

$$Q'_s = \frac{1}{\tau} \int_0^{\tau} F_s \dot{X}_p dt$$

where

X_p = piston velocity

F_s = shaft force

τ = period of the engine.

The model engine was loaded with a double-acting helium compressor. This device allows external adjustment of the engine load during operation. A theoretical helium compressor has an approximately rectangular force vs

displacement diagram. However, this compressor was designed for a very low pressure ratio, which changed its characteristics to more of a damper than a compressor, rounding off the corners of the theoretical diagram.

In addition, there may be significant internal friction since the bearing and piston rings were not selected to minimize friction. Since the characteristics of the load are not well known, and, in fact, are not of interest, the engine output power was measured directly rather than calculated from the output of the compressor.

4.2.3 Heat Rejected

For heat source and heat sink temperatures typically encountered, most of the heat input to the thermodynamic cycle is rejected as heat in the cooler.

Determination of the heat rejected is of particular importance in the design of the auxiliaries and it is a major component of the engine energy balance. The heat may be rejected to a heat transfer fluid or a low temperature sink by radiation. For the case of heat being rejected to a heat transfer fluid of known properties flowing through the cooler, the heat rejected may be found from an energy balance on the flowing fluid which reduces to

$$Q'_R = m' C \Delta T,$$

where

Q'_R = heat rejected

m' = mass flow rate of fluid

C = specific heat of fluid

ΔT = temperature change in fluid.

For the case where the heat is ultimately rejected by radiation, a heat pipe may be desirable as an intermediate transport device. The measurement of the heat rejected for this case could be accomplished by measurement of the temperature drop across a known thermal resistance as described in 4.2.1.

4.2.4 Engine Losses

The dominant losses in the FPSE can be classified as thermodynamic irreversibilities, viscous friction, or heat transfer losses. The losses which are important to the performance calculations are those resulting in an energy transfer across the engine boundaries. Other losses are important to understanding the engine performance and are necessary in the optimization of its performance, however, these will not be discussed here. The two basic mechanisms by which energy can be transferred across the engine boundaries are heat transfer and mechanical.

Heat transfer losses are the result of temperature differences across the system boundaries. In order to obtain an accurate energy balance, the heat loss via each path across the boundary must be accurately known, or the magnitudes of the loss must be small.

The mechanical energy transferred across the boundary is the result of unbalanced forces imposed on the cylinder structure by the normal operation of the engine. The ERDA engine was spring mounted allowing the casing to move. If there is a damping component in this support, energy will be dissipated by that component. The magnitude of this loss will depend on the operating frequency, amplitude of the cylinder motion and the effective damping factor of the supporting structure.

Due to the number of connections between the engine and ground (plumbing, electrical, etc.) direct measurement of the work dissipated by the moving engine cylinder would be difficult at best. Therefore, the shaft output power correction was developed to account for the moving cylinder. The derivation of this correction is given in Appendix II.

4.2.5 Engine Energy Balance

From the measurements described in Sections 4.2.1 and 4.2.3, an energy balance on the engine may be completed; i.e.,

$$Q'_I + Q' + Q'_R + Q'_L = 0,$$

where

Q'_I = heat input rate

Q' = output power from engine (Q'_s) or system (Q'_o)

Q'_R = heat rate rejected

Q'_L = losses,

and inputs are positive and outputs negative. In most cases, the equation will not equal zero due to experimental error; i.e.,

$$Q'_I + Q' + Q'_R + Q'_L = \varepsilon$$

where

ε = energy balance residual,

and $|\varepsilon| > 0$.

For the test to be considered valid, the magnitude of ε should be within the uncertainty of the measurement errors propagated by the calculations.

4.2.6 Engine Efficiency

Various efficiency terms may be defined in terms of the inputs and outputs of the engine, components, and system. The engine indicated efficiency is then

$$\eta_T = \frac{Q'_T}{Q'_I},$$

where

η_T = indicated total efficiency

Q'_T = thermodynamic power output,

Q'_I = power input

this is a measurement of the thermodynamic engine performance. The engine mechanical efficiency may be defined as,

$$\eta_m = \frac{Q'_S}{Q'_T},$$

where

η_m = mechanical efficiency

Q'_s = engine shaft power.

This is a measure of how well the energy developed by the thermodynamic cycle is transferred to the output shaft. The net engine efficiency is then

$$\eta_e = \eta_T \eta_m = \frac{Q'_s}{Q'_I}$$

where

η_e = net engine efficiency.

The efficiency of the engine load such as a linear alternator may be found from

$$\eta_l = \frac{Q'_o}{Q'_s},$$

where

η_l = load efficiency

Q'_o = net system power.

The net or overall system efficiency is then

$$\eta_n = \eta_T \eta_m \eta_l = \frac{Q'_o}{Q'_I}$$

where

η_n = net system efficiency

It is this net efficiency which is of primary importance to the system designer, since it indicates the relative quantity of fuel required for a particular mission.

4.3 ENGINE PERFORMANCE PARAMETERS

The performance parameters described in the preceding section are of interest to the system designer and engine developer. The engine developer is also

interested in the details of the engine operation and how they impact the macroscopic performance. This section describes the important engine design parameters and the reasons for their importance.

4.3.1 Engine Geometry

Piston area, displacer rod area, and cylinder area are critical geometric design parameters. These and other parameters, including working space dimensions, heat transfer surface areas, flow cross sections, limits of component motions and component masses are important since they affect engine performance and are essential for engine simulation.

4.3.2 Bounce Space Pressure

The bounce space pressure determines the average working space pressure and therefore impacts the work output per cycle, frequency, and power output. Schmidt analysis indicates that the work output per cycle is proportional to the mean cycle pressure, i.e.,

$$Q \propto P_{\text{mean}}$$

where

Q = Thermodynamic cycle work

P = Mean pressure.

Further the engine frequency is a function of the cycle pressure when the gas spring effects are large compared to the mechanical springs. In this case, the dependence of efficiency on pressure is approximately represented by:

$$f \propto (P_{\text{mean}})^{1/2}$$

where

f = engine frequency.

The power output may, therefore, be approximated by

$$Q' \propto P_{\text{mean}}^{3/2},$$

where

$$Q' = \text{Power output.}$$

It is apparent that the charge or bounce space pressure, which is approximately the mean cycle pressure, is an important parameter.

4.3.3 Heater/Cooler Temperature

The power output and thermal efficiency of the Stirling Engine are functions of the ratio of heat source to heat sink temperatures. The work output per cycle is, according to the Schmidt analysis, related to the temperature limits by,

$$Q \propto (1 - T_c/T_e)$$

where

$$Q = \text{Work output}$$

$$T_c = \text{Heat source temperature}$$

$$T_e = \text{Heat sink temperature,}$$

and of course the thermal efficiency for the Schmidt cycle is

$$\eta_{\text{Ideal}} = (1 - \frac{T_c}{T_e})$$

where

$$\eta_{\text{Ideal}} = \text{Ideal thermal efficiency.}$$

Though real engine performance deviates significantly from these functions, they do indicate the importance of the cycle temperature limits. The appropriate source and sink temperatures used are difficult to define when there is a special temperature distribution in the heat exchangers. In this case, the maximum wall temperature and minimum sink temperatures should be used for

general correlation of results. The temperature of the heat exchanger wall will oscillate to some extent, however, since the Biot number is small, the oscillation should be small, and the wall temperature may be taken as the heat source temperature. During testing, as many temperature measurements as possible should be made in order to characterize the spacial distributions in the heat exchangers.

4.3.4 Component Dynamics

The dynamics of the displacer and power pistons are important for several reasons. First, they determine the volumetric boundaries for the thermodynamic system and therefore, exert a major influence on the overall performance of the system. Second, they feed directly into the determination of the indicated and shaft work outputs (Section 4.2.1). Third, they indicate the system frequency and, therefore, in combination with the cycle work, the power outputs. Since the component displacements are useful for physical interpretation of engine performance, position transducers are required for these measurements. Considerable care must be exercised with this measurement to insure that the measurement system faithfully reproduces the component amplitudes and phase relations. A check on the position measurement may be made by measuring the component velocity and integrating with respect to time to obtain displacement.

Traditionally, the piston motion has been measured relative to the cylinder wall (fixed in most cases) and the displacer motion has been measured relative to the piston motion. The piston motion is, therefore, an absolute position, while the displacer absolute position must be derived from its relative position and the piston position; i.e.,

$$X_D = X_{D/P} + X_P,$$

where

X_D = Absolute displacer position

X_P = Absolute piston position

$X_{D/P}$ = Displacer position relative to piston

For tests and simulation outputs, the relative position of the displacer is frequently plotted against the absolute position of the power piston. The resulting plot is an easily interpreted indicator of the engine performance. A typical plot of the displacer phasing is shown in Figure 4.5, and analyzed in Appendix III.

4.3.5 Working Space Pressure

The instantaneous working space pressures are important for two reasons. First, they are the basic driving force for the displacer and power piston dynamics. Second, they can be used in conjunction with the component dynamics to calculate the thermodynamic work output.

The pressure and resulting force vector relationship for an ideal thermodynamic cycle and its corresponding component dynamics is shown in Figure 4.6. Another good indicator of the overall health of the engine is a P-V diagram, an example of which is shown in Figure 4.7. The shape of this diagram for practical engines must be obtained from a detailed simulation, however, a Schmidt approximation with a sinusoidal pressure wave results in an elliptical P-V diagram. In an actual engine the peaks are sharper and the valleys are flatter, resulting in a kidney shaped diagram. For a real time indication of the P-V relation, the pressure is plotted against the piston position during testing. The indicated work delivered to the piston by the gas can be calculated from the area of this diagram.

4.3.6 Heat Exchanger Pressure Drop

The pressure drop in the heat exchangers has two major effects on the performance of the engine. First, it alters the ideal thermodynamic cycle and second, it is the major damping term in the displacer dynamic equations. Though it is a relatively important factor in the free piston engine, there is no report of it being experimentally determined in the current literature. This is especially important for optimization of the heat exchangers and validation of the simulation code. This measurement will also allow improved

Fig. 4.5 Displacer Phase Diagram

Fig. 4.6 Vector Diagram of a Single Cylinder FPSE

DEFINITION OF TERMS

$V_c = A_c X_{D/P}$, Amplitude of compression space volume

$V_E = A_e X_D$, Amplitude of expansion space volume

X_p , Amplitude of piston motion

X_D , Amplitude of displacer motion

$X_{D/P} = X_D - X_p$, Amplitude of displacer motion with respect to piston motion

ϕ , Angle which displacer leads piston motion

α , Angle which expansion space leads compression space volume

θ , Angle which the pressure amplitude lags the displacer

A_E , Area of cylinder (expansion space)

A_R , Area of displacer rod

$A_c = A_E - A_R$, Area of piston head (compression space)

F_P , Amplitude of piston force

F_D , Amplitude of displacer force

Fig. 4.7 Typical P-V Diagram for a FPSE

measurement of the total (indicated) cycle work, since the actual pressure in the expansion and compression spaces will be measured. The present engine geometry constraints did not allow for this measurement. Modification to allow its measurement will be made when appropriate.

4.4 DESCRIPTION OF EXPERIMENTAL APPARATUS

This section describes the free piston Stirling model engine and instrumentation used to experimentally determine the performance characteristics.

4.4.1 Model Engine

The engine used in the test is shown in Figures 4.8 to 4.11. The auxiliaries required to operate the engine are: helium supply, electrical power and cooling (see Figures 4.12 to 4.15). Helium is supplied from high pressure bottles through a control circuit shown in Figure 4.16. Pressure gauges are included in this system to provide the operator with the status of the bottle pressure, supply pressure for starting, and a back reading of the bounce chamber pressure.

Cooling water used to cool the engine was supplied from city water, through the control circuit shown in Figure 4.17. This water is filtered to remove particulate material, then regulated to reduce supply pressure fluctuations. The flow to the engine cooling jacket was monitored with a rotometer and manually controlled with a valve. Flow to the load cooling loop and the transformer cooling was arranged to absorb heat dissipated. Since the heat dissipated in these components is not important for calculation of an energy balance, the flow was not carefully measured or controlled.

The electrical power supplied to the heater was controlled by the circuit shown in Figure 4.18. Power input is normally controlled by adjusting a variac to obtain a predetermined heater tube temperature. The instrumentation is relatively important since the power input is a fundamental component of

Fig. 4.8 Free-Piston Stirling Engine on Test at MTI

Fig. 4.9 Assembled Model Engine

Fig. 4.10 Electric Heater and Generator Assembly

Fig. 4.11 Displacer and Power Piston Assembly

Fig. 4.12 Coolant and He Supply Panel

Fig. 4.13 Power Input Control

Fig. 4.14 Instrumentation Rack

Fig. 4.15 Readout Instrumentation

Fig. 4.16 Helium Circuit

Fig. 4.17 Cooling Water Circuit

Fig. 4.18 Schematic of Heater Electrical Measurements

the energy balance. Details of this measurement are given in the next section.

4.4.2 Engine Instrumentation

The engine is instrumented such that the operating conditions, output power, energy balance, and efficiency can be calculated. Figure 4.19 shows an overall schematic of the engine instrumentation. The readout equipment is shown in Figures 4.14 and 4.15.

The power input is calculated by measuring the voltage and current supplied to the heater at several locations. The current in each circuit is stepped down with a 400 to 1 current transformer. The secondary is in series with a 0.1Ω , 1% resistor across which the voltage drop is measured with a digital AC voltmeter. Voltages across the heater and/or thermal guards, indicated in Figure 4.18, are also measured with a digital AC voltmeter. The power input is then calculated as described in Section 4.6.

Heater and heater guard temperatures are measured in locations shown in Figure 4.20. These temperatures are measured with ISA type K (Nickel-Chromium vs Nickel-Aluminum) thermocouples. The thermocouple to be measured is selected by a multi-point thermocouple switch. The output EMF of the thermocouple is then digitized and converted to a linearized output proportional to temperature in a digital temperature indicator (Doric Trenicator, Model 410A-K-C.)

The bounce space pressure is measured with a Kistler piezoresistive pressure transducer (Model 4041A50) located on the engine. The output of this instrument is converted to a voltage proportional to pressure and displayed on a digital indicator which is set up to read directly in bars.

The heat rejected to the engine cooling jacket is calculated from the flow and temperature rise of the cooling water. The primary flow measurement is made with a COX LF-6-3 turbine flow meter, the pulse output of the meter

Fig. 4.19 Control and Instrumentation Schematic

Figure 4.20

is amplified and counted. The flow rate is then calculated from the turbine frequency and a calibration curve. The temperature rise across the heat exchanger is measured with differential thermocouples (ISA Type T, copper vs copper-nickel.) The differential thermocouple is selected with a switch and the output EMF measured with a digital millivolt meter (Doric Trendicator, Model 410A-K1.) The heat rejected is calculated as described in Section 4.6.

The dynamic positions, pressure, velocity, and force are measured as shown in Figure 4.19. The positions are measured with special DC-LVDT's which have had the demodulator filter removed. External 2-pole filters are provided which have been phase matched to eliminate any relative phase error in the displacer and piston measurements. The displacer position is displayed vs the piston position on the scope to obtain engine stroke and phase information.

The working space pressure is measured with a piezoelectric pressure transducer (Kistler, Model 701H), which is connected to the working space at the interface between the cooler and regenerator. Since the pressure drop in the cooler is less than 10% of the total heat exchanger system pressure drop which is on the order of 1 to 2% of the average system pressure, this pressure is very close to the compression space pressure. This pressure signal is filtered to phase match it with position signals displayed vs piston position. The area of the resulting diagram is proportional to the work done by the compression space gas on the piston.

The shaft force and velocity are used to determine the shaft power output. The shaft force is measured with a piezoelectric force link (Kistler, Model 9312A), which produces an electrical charge which is converted, in a charge amplifier, to an output voltage proportional to force. The piston velocity is measured with a linear velocity transducer (LVT) (Transtek, Model 111-001). The output of these two transducers are multiplied and filtered electronically to produce an output voltage which is displayed on a digital voltmeter. A block diagram of the system used is shown in Figure 4.21.

Fig. 4.21 Power Output Measurement

4.5 DATA REDUCTION

4.5.1 Energy Balance Boundaries

The control volume for the energy balance on the FPSE may be arbitrarily selected. However, the careful selection of the control volume can in some cases reduce the problems of closing an energy balance.

Several potential control volumes are shown in Figure 4.22. The boundary of primary interest is boundary "A" and the energy flux terms of interest are:

1. Electrical power input to the heater tubes
2. Heater losses to the surroundings
3. Heat rejected to the cooling water
4. Power delivered to the load
5. Mechanical losses to the ground.

For the control volume A, there are three components of the heater loss to the surroundings.

1. Conduction via the thermal guards
2. Convection from the insulation

and

3. Assuming the insulation is transparent, infrared radiation from the heater.

If we confine our attention to the heater, the boundary at the heater end shown in Figure 4.22 can be drawn to include the thermal guards (Boundary "B"). This change in boundary has the advantage that a much better measurement of the losses from the thermal guards is possible by measuring the temperature rise in the cable terminal coolers. The electrical input measurement in this case must, however, be measured at the cold end of the guards. The problem with this measurement technique is that there are potentially more losses from the guards due to convection, and heat generation in the cable connections which will be picked up by the cooling water.

The current approach is to provide the instrumentation necessary to calculate

Fig. 4.22 Schematic of Experimental Boundaries

the net heat input to the thermodynamic cycle at both boundaries and use the independent measurements to confirm the primary calculation.

At the output end of the engine, work delivered to the load by the piston is measured by a load cell and velocity transducer arrangement. The power absorbed by the helium pump load device, results in an increase in the enthalpy of the helium working fluid being circulated by the pump. This is ultimately dissipated in a load cooler. As a confirmation of the power output measurement, the temperature rise of the coolant water through this load cooler may be measured. It is anticipated that some losses to ambient between the helium pump and the cooler occurs, however, the approach is to make sufficient measurements to confirm the basic power calculation.

4.5.2 Heat Input

Heat generation in heater tubes is defined as

$$Q'_{HTR} = 4000.0 V_{HTR} (V_{C1} + V_{C2})$$

while heat generation in the heater system at Boundary "A" is

$$Q'_{IA} = 4000.0 (V_{A1} V_{C1} + V_{A2} V_{C2})$$

Heat generation in heater system at Boundary "B" is defined as

$$Q'_{IB} = 4000.0 (V_{B1} V_{C1} + V_{B2} V_{C2})$$

where

Q'_{HTR} - Heater generation in heater tubes

Q'_{IA} - Heat generation inside boundary "A"

Q'_{IB} - Heat generation inside boundary "B"

V_{HTR} - Voltage across heater tubes

$V_{A1,2}$ - Voltage across circuit 1 and 2 at boundary "A"

$V_{B1,2}$ - Voltage across circuit 1 and 2 at boundary "B"

$V_{C1,2}$ - Voltage output of current transducers for circuit 1 and 2.

4.5.3 Power Output

The power output was calculated by multiplying the shaft force and velocity signals from the transducers electronically and reading out the DC level of the product (see Section 4.2.1).

4.5.4 Heat Rejected To Engine Cooling Jacket

The expression for heat rejected to the cooler is

$$Q'_R = m' C \Delta T$$

where

Q'_R - Heat rejected to engine cooling jacket

m' - Mass flow rate of water through cooling jacket

C - Specific heat of coolant

ΔT - Temperature rise across cooling jacket

The mass flow rate of the cooling water was measured with both a rotometer and a Cox LF-6 turbine meter. The flow rate was then found from either:

$$m' = K_R (RM)$$

or

$$m' = K_T (TM)$$

where

K_R - Rotometer calibration factor

RM - Rotometer reading

K_T - Turbine meter calibration factor

TM - Turbine meter frequency

A calibration curve for this measurement is shown in Figure 4.23.

Figure 4.23 Cox Flow Meter Calibration

The ΔT is found from

$$\Delta T = \left. \frac{dT}{dx} \right|_x \Delta x$$

where $\frac{dT}{dx}$ - slope of the thermocouple linearization curve.

Δx - Output voltage of differential thermocouple.

Substituting for the slope at a temperature of 30 C

$$\Delta T = (24.3 \frac{^{\circ}\text{C}}{\text{mv}}) \Delta x$$

4.5.5 Confirming Power Output

The power output as discussed above can be measured at either boundary "A" or "C". The confirming power output is calculated from measurements of the load coolant flow rate and temperature rise (Boundary "C"). It is expected that this measurement would be slightly lower than the power measured at "A" since there will be some convection loss from the surface of the bounce chamber and the plumbing connecting the load to the load cooler.

4.5.6 Heater Loss

Heat losses from the heater are via two paths, the thermal guards and the insulation. The thermal guards are instrumented with a series of thermocouples to allow calculation of an energy balance between boundaries "A" and "B" on the four individual bars. The heat flux at each end is found from the measured temperature distribution in each guard. A polynomial is fit to the measured temperature to obtain an equation for the distribution. The gradient at each boundary is then calculated from this equation and used with material property and bar geometry data to obtain the heat flux. The temperature distribution is also used to calculate the electrical resistivity and thus, with the measured current, the internal heat generation in the thermal guards. Finally, the temperature distribution is used to calculate the heat conducted to atmosphere through the insulation on the guard. As a further check on this

loss the water flow and temperature rise of the water used to cool the ends of the thermal guards is also measured. This provides an independent measurement of the heat flux at boundary "B" through the guards. There is, however, some additional heat generation in the electrical cables and terminals which will be conducted into this cooling water. Therefore, the gross measurement of the heat into the coolant should be slightly higher than that being conducted out of the guards. The heat loss via the insulation surrounding the heater head is a combination of radiation and conduction through the insulation and convection of the heat from the surface of the insulation to ambient.

The insulation loss is calculated from the measured temperature difference between the heater and the outer surface of the insulation, the geometry of the combination and the properties of the insulation material (Table 4.1). The difficulty with this measurement is that relatively few temperature measurements were performed and the geometry is not a regular shape. An alternative calculation can also be performed from the surface convection off the insulation. This calculation suffers from few measurements of the surface temperature and a rather uncertain convective conductance, since the surface is irregular and oscillating. However, the loss from this surface is rather small with respect to the heat input and, therefore, the effect of any error is also very small.

4.6 ERDA MODEL ENGINE TEST PLAN

Figure 4.2.4 is a flow diagram for testing of the model engine and validation of the computer model. The detailed approach used is outlined as follows.

4.6.1 Preliminary Engine Tests

- Establish gross engine performance for steady-state operation.
 - Measure power output
 - Measure power input
 - Calculate efficiency
 - Measure or estimate losses
 - Calculate energy balance

TABLE 4.1
 CONDUCTIVITY OF "CERAWOOL" BLANKET INSULATION

Density (lb/ft ³)	Thermal Conductivity W/m °K		
	4	6	8
Temperature			
260°C	.078	.062	.055
538°C	.160	.127	.104
815°C	.287	.232	.179

Note: ERDA model engine uses 6lb/ft³ insulation for the heater.

Fig. 4.24 Engine Testing Flow Diagram

- Instrumentation measurement and readout verification and calibration
 - Testing against known standard
 - Cross checking between measurements
- Establish control and operating procedures
 - Establish starting procedure
 - Establish control procedures for heater temperature, engine pressure, and engine load
 - Establish data recording procedures

4.6.2 Establish Performance Predicted By Simulation

- Input
 - Known geometry and dynamic parameters
 - Estimated heat transfer and fluid flow characteristics
 - Estimated piston damping (see Figure 4.25)
- Calculate
 - Component dynamics
 - Elemental temperatures, metal, and gas
 - Heat exchanger flow rates
 - Thermodynamic pressures
 - Power output
 - Heat input
 - Heat rejected
 - Miscellaneous heat flows
 - Efficiency
 - Ideal cycle calculations
- Identify Simplifying Assumption
 - Expansion and compression spaces are represented by one (1) energy node each
 - Heater and cooler are represented by (3) three energy nodes each with a constant temperature wall node
 - Heat transfer coefficients are assumed constant over a cycle
 - Fluid mechanics allow for turbulent flow - resistance concentrated at expansion and compression space entrances

Fig. 4.25 Load Damping Characteristic

- One pressure node each for the expansion, compression, and heat exchange spaces
- Displacer and piston motions are modeled by a velocity and displacement node for each
- Cylinder is fixed
- Regenerator gas and matrix represented by 5 energy nodes each
- Load is an ideal viscous damper
- Validate Simulation by Comparison of the Following Output with Test Data
 - Power output
 - Energy balance
 - Efficiency
 - Component dynamics
 - Dynamic pressure
 - Metal temperatures

4.6.3 Detailed Engine Test

- Test at steady-state conditions
- Data point at 5 levels each (see Table 4.2) of:
 - Pressure
 - Heater temperature
 - Piston stroke
 - Order of points will be random with repetition to detect dependence on total running time

4.7 TEST RESULTS

The model free piston engine for laboratory test was installed at MTI in May of 1976. The apparatus and basic instrumentation had been readied and the engine was mounted and successfully run within two days of its arrival. During these check-out tests, operating pressure and temperature were maintained at levels lower than those used during final check-out tests at Sunpower. Consequently, the power developed was also lower. When the results were

TABLE 4.2
CANDIDATE TEST POINTS

<u>Level</u>	<u>Pressure (Bar)</u>	<u>Temperature (C°)</u>	<u>Stroke (cm)</u>
1	50	650	4.5
2	45	600	4.25
3	40	550	4.0
4	35	500	3.75
5	30	450	3.5

extrapolated to similar operating conditions, there was good agreement between the data. Based on this comparison it was felt that the engine did not change in performance during shipment and consequent installation.

Initial engine runs (Table 4.3) were made to cross check the instrumentation and to identify areas where the instrumentation or measurement techniques could be improved. These improvements are reflected in the description of the engine and instrumentation given in Section 4.4.

4.7.1 Energy Balance Test

An energy balance, indicating that the instrumentation and measurement technique resulted in accurate and consistent results, was obtained during the initial series of tests. The results of the test on which an acceptable energy balance was obtained are summarized in Table 4.4. The contributing items in the energy balance are briefly discussed in the following paragraphs.

The gross heat input was calculated from the voltage drop across each side of the heater input at boundary "A", and the current through the end heater circuit. Measurements indicated a difference of as much as 14 percent of the total input power existed between the two input circuits. This difference was also seen in the temperatures of the thermal guards, which are lower in the low current circuit.

The net thermal guard loss was calculated from the temperature distribution in each bar. The energy crossing boundary "A" was obtained, which in this case was heat flow into the heater. The total internal generation in the guards between boundaries "A" and "B" was then calculated from the temperature distribution, current, and thermal properties of the material. With the additional electrical measurements at boundary "B", the internal heat generation in the guards was checked by direct measurement, rather than indirectly from the temperature distribution and material properties. The difference in power input between boundaries "B" and "A" was 784.2 watts, which agrees reasonably well with a value of 639.6 watts measured as rejected heat from the guard coolers.

TABLE 4.3

INITIAL TEST RESULTS

Seq. No.	Trial No.	Date	Run* Time (Hrs)	Pressure (Bar Abs)	Heater Temp. (°C)	Piston Stroke (cm)	Shaft Power (watts)	Frequency (Hz)	Work (w·s)	Specific Work (w·s) (cm·bar)
1		6/11/76	4.5	41.0	597	4.07	667	18.5	36.05	.216
2		6/14/76	6.25	41.2	600	4.14	640	18.3	34.97	.205
3		6/18/76	9.25	41.0	504	3.40	393	18.5	21.24	.152
4		6/18/76	12.50	41.2	570	4.29	570	19.1	29.84	.169
5		6/28/76	16.00	40.2	509	4.03	491	18.6	26.40	.163
6		6/29/76	18.0	40.3	560	4.22	623	18.7	33.32	.196
7		6/30/76	19.5	41.1	503	4.00	522	19.1	27.33	.166
8		7/8/76	26.33	41.2	505	4.00	547	19.3	28.34	.172
9		7/15/76	28.83	41.0	527	3.85	527	19.0	27.74	.176
10		7/19/76	34.08	41.3	507	3.70	451	19.4	23.25	.152
11		7/21/76	36.08	41.0	516	3.70	508	19.1	26.59	.175
12		8/4/76	39.50	41.4	507	3.61	473	18.5	25.57	.171
13 +		8/7/76	41.5	42.0	501	3.60	479	19.0	25.21	.167

* Cumulative run time from start of testing at MTI

+ Energy Balance test see Table 4.4

Table 4.4

Summary of Engine Energy Balance

● Engine Operating Point

- Heat Temperature 501.5°C
- Heat Sink Temperature 23°C
- Mean Cycle Pressure 42.0 BAR (ABS)
- Frequency 19.0 Hz

● Engine Energy Balance (watts)

● Gross electrical input (boundary "A")	2273.6	
● Thermal guard loss (gain)	22.3	
● Engine heat rejected to cooling water		-1622.8
● Heater loss to ambient		- <u>124.7</u>
TOTAL HEAT LOSS		-1747.5
● Power output to load		- 479.0
● Cylinder acceleration correction		- <u>10.0</u>
TOTAL INPUT	2295.9	
TOTAL REJECTED		-2236.5
UNACCOUNTED FOR ENERGY		59.4

● Engine Thermodynamic Efficiency

- Based on net heat input 22.6%
- Based on heat rejected 23.2%

The heat rejected to the engine cooling water, calculated from the measured flow and temperature rise of the cooling water, was 1622.8 watts.

The shaft power output of the engine was calculated from the shaft velocity and the force transmitted to the load. For this point, the measured output power was 479.0 watts.

A velocity transducer was added to the engine instrumentation which measures the velocity of the engine casing with respect to ground. With this measurement, a better measurement of the angle by which the cylinder motion lags the piston was obtained. This indicated that the angle was about 10 degrees. Therefore, the angle between the piston relative velocity and the cylinder acceleration was 80 degrees. The correction to the test data for the energy balance was 10 watts, compared to a measured power output of 479 watts.

The heat loss from the heater to the ambient through the insulation surrounding the heater was calculated from the average temperature of the heater, the surface temperature of the insulation and the physical properties of the insulation. The conductivity used for this calculation was obtained from the manufacturers specifications for the material used.

The level of unaccounted for energy was 59 watts or 2.6 percent of the total input, which is within the overall uncertainty of the individual measurements.

The engine thermodynamic efficiency was calculated in two ways; efficiency calculated from the shaft power divided by the net heat into the thermodynamic cycle,

$$\eta_{\text{Input}} = \frac{\text{Work}}{Q_{\text{Input}}} = \frac{489}{2171} = .226,$$

and efficiency based on the shaft power and heat rejected

$$\eta_{\text{Rejected}} = \frac{\text{Work}}{Q_{\text{Rejected}} + \text{Work}} + \frac{489}{1623 + 489} = .232$$

The small difference between these two efficiencies is due to the unaccounted energy in the balance.

The piston and displacer dynamic data measured during the energy balance is shown in Figures 4.26 and 4.27. The top diagram in Figure 4.26 presents the relative displacer position with respect to the piston position and the working space pressure as a function of piston position. The area under the pressure curve and swept by the piston (piston displacement) is proportional to the thermodynamic work done by the gas on the piston. The lower figure in Figure 4.26 presents shaft force vs piston position. The area under the force vs displacement diagram is proportional to the shaft work and this work is normally not the same as the measured P-V work due to the piston friction and work transfer from the displacer to the piston. In addition, the force vs displacement diagram (C) indicates that the helium compressor load device is a non-linear damper.

Figure 4.27 shows the displacer/piston phase diagram of Figure 4.26 with respect to the limits of the machine. This diagram is an indication of the position of the components within the cylinder and the absolute stroke of each.

It should be noted that after the energy balance testing, a leaf spring which provides lateral support for the engine was removed since it was not found to be necessary. This change virtually eliminated the angle by which the cylinder lagged the piston, and significantly reduced the power correction.

4.7.2 Performance Mapping Tests

The test points for the performance mapping tests were arrived at by a two-factorial experimental design. This approach should identify the relative importance of the three selected control variables, engine pressure, heater tube temperature, and power piston stroke. The control variables for each test correspond to one corner or the center of a parallelepiped. A summary of the data and the results of the performance mapping experiments are tabulated in Table 4.5.

WORKING SPACE PRESSURE
RELATIVE DISPLACER POSITION

PISTON DISPLACEMENT

PISTON DISPLACEMENT

Figure 4.26 FPSE Oscilloscope Pictures

Strokes

$P = 3.6$ cm

$D = 3.3$ cm

$\phi = 39.5^\circ$

Fig. 4.27 Displacer Phase Diagram

Seq. No.	Trial No.	Date	Run* Time (Hrs)	Pressure (Bar Abs)	Heater Temp. (°C)	Piston Stroke (cm)	Shaft Power (watts)	Frequency (Hz)	Work (w·s)	Specific Work (w·s) (cm·bar)	Efficiency (%)
1	9	8/23/76		40.9	550.9	3.321	426	18.0	23.67	.174	23.0
2	1	8/23/76		31.4	454.2	3.423	264	16.3	16.20	.151	19.3
3	5	8/23/76		31.6	453.2	3.864	301	16.4	18.35	.150	20.1
4	1	8/23/76		31.5	451.9	3.333	258	16.5	15.64	.149	19.8
5	3	8/23/76		31.3	648.1	4.359	530	16.0	33.13	.243	25.0
6	2	8/23/76	49.0	50.0	452.2	2.291	182	20.0	9.10	.079	14.1
7	9	8/24/76		41.8	550.5	3.223	400	18.3	21.86	.162	22.0
8	7	8/24/76	52.0	31.5	647.3	4.327	426	16.3	26.14	.192	22.7
9	6	8/24/76	53.5	50.5	454.2	3.381	436	21.1	20.66	.121	20.5
10	3	8/25/76	55.5	31.5	650.3	4.352	450	16.3	27.61	.201	20.2
11	3	9/9/76	64.0	31.1	651.3	4.414	509	15.8	32.22	.235	26.1
12	6	9/10/76		51.1	497.7	3.241	403	20.9	19.28	.116	19.8
13	5	9/10/76		31.4	475.4	4.035	290	16.0	18.13	.143	19.7
14	2	9/10/76		51.0	478.4	2.760	286	20.5	13.95	.099	18.9
15	9	9/10/76		41.2	550.0	3.284	372	18.2	20.44	.151	20.5
16	4	9/10/76	70.5	51.0	655.4	2.918	400	19.8	20.20	.136	22.3
17	8	9/13/76		51.0	648.7	3.223	506	20.4	24.80	.151	23.9
18	7	9/13/76		31.6	650.4	3.623	340	15.4	22.18	.193	21.5
19	3	9/13/76		31.2	651.4	4.408	463	16.0	28.94	.210	24.8
20	8	9/13/76		51.0	653.7	3.223	505	20.4	24.76	.151	24.2
21	9	9/13/76	78.0	41.1	579.4	3.284	345	18.2	18.96	.140	20.5

* Cumulative runtime from start of testing at MTI

The measured power output for each test point was reduced to a normalized work per cycle by the following equation,

$$sw = \frac{P}{f \cdot s \cdot B} ,$$

where sw - specific work parameter

and P - power output (watts)

f - frequency (Hz)

s - stroke (cm)

B - bounce pressure (Bars Abs),

to eliminate the first order effects of the control variables. The resulting specific work should then be a function of temperature, second order effects of pressure and stroke, and uncontrolled variables. When this parameter is plotted sequentially, Figure 4.28, some idea of performance variation with time can be seen.

The performance mapping tests were run at points corresponding to the corners and center of a three-dimensional cube. Figure 4.29 summarizes the control and dynamic variables corresponding to each test point. Figures 4.30 and 4.31 summarize the experimental results for the conditions given in Figure 4.29 and presents them on a cube diagram for easy comparison.

4.7.3 General Conclusions

Two general conclusions can be drawn from the model free piston Stirling engine tests performed.

First, by careful selection of a system control volume and careful measurement of the energies crossing the thermal boundaries, a system balance can be obtained within the accuracy of the instrumentation used. With the present instrumentation and procedures, the unaccounted for energy was less than 3% of the total energy flow through the system.

Fig. 4.28 Engine Performance as a Function of Time

Fig. 4.29 2³ Factorial Diagram Control & Dynamics Variables

P - Pressure (BARS ABS)
 T - Heater Temperature (°C)
 S - Piston Stroke (cm)
 φ - Displacer Phase Angle (°)

Fig. 4.30 2^3 Factorial Diagram

F - Frequency (Hz)
 P - Power Output (Watts)
 η - Efficiency (%)

Fig. 4.31 2^3 Factorial Diagram

W - Work Per Cycle
 SW - P/(BAR*STRK*FREQ)
 WS - W/STRK

Second, the performance test results showed that the model engine power output deteriorated with running time. The primary factors resulting in this power degradation are believed to be related to the materials selected for the displacer rod sleeve and piston/displacer seal rings in the engine.

It should be noted that this engine was designed and intended for purposes other than long-term life. The engine is an experimental model that was primarily a tool to provide data to validate the computer simulation program. It has provided that data and as such, served the purpose intended. Presently, modifications are being made to improve the engine in terms of operating performance (power output and efficiency) using the available analytical methods as predictive tools. In addition, new designs to improve the areas that may be the cause of performance degradation are also contemplated.

SECTION 5.0

DEVELOPMENT OF A FREE-PISTON STIRLING ENGINE COMPUTER SIMULATION CODE

5.1 INTRODUCTION

This section describes the computer modeling of the FPSE, done by T. Finkelstein under subcontract to MTI, called BSAP. The refinements which MTI has implemented in the BSAP model are also described.

5.2 SUMMARY OF THE BSAP APPROACH

The Beale Stirling Alternator Program (BSAP) was initially developed by Dr. Finkelstein of Trans Computer Associates (TCA) for MTI to analyze the performance of Beale free-piston Stirling engines. The analysis program, an initial problem statement and a preliminary draft of the operating manual were delivered to MTI on April 12, 1976.

The basic approach used in the simulation is to model the energy, momentum, and continuity equations as an analogous thermal network. The continuity and energy equations divide the working space gas into 13 nodes, see Figure 5.1, while the mass/momentum equations divide the working space into three mass nodes, Figure 5.2. In addition, energy equations are solved for many of the working gas boundaries. In the case of the regenerator matrix nodes, the temperature variation over the cycle may have a significant impact on the system performance, since the matrix time constant is on the order of one fifth of the system period. On the other hand, the other boundary nodes have time constants which are significantly greater than the period of the cycle, and these can be set to appropriate constants without significant loss of accuracy.

A complete description of the BSAP program and its use is given in the draft User's BSAP Manual submitted by TCA. At the present time, certain portions of the manual have been identified as proprietary and, therefore, cannot be publicly released, however, the basic analysis is described in a paper by Finkelstein entitled, "Computer Analysis Of Stirling Engines"

Figure 5.1 13 Node Thermal Model

Figure 5.2 3 Node Mass Model

5.3 MODIFICATIONS TO THE BSAP PROBLEM STATEMENT

The definition of the BSAP problem statement as delivered to MTI required several slight modifications in order to obtain a more realistic model of the engine.

5.3.1 ACCURACY IMPROVEMENTS

The transport terms of the energy equation are modeled by a cathode follower or diode function. The initial model of these terms attempted to use an averaging technique to stabilize the numerical solution of the differential equations. It was thought that this stabilizing procedure was causing errors which resulted in the solution "blowing up." When this was replaced with a physically realistic model, the problem was eliminated. The more upstream cathode follower nodes are now set to the upstream gas temperature and the corresponding resistance is set to the product of the flow rate and specific heat. The resistance for the downstream cathode node is set to zero, since the downstream temperature has an insignificant effect on the local gas temperature.

The piston and displacer motions are limited by several physical constraints. The actual constraints for the model engine are shown in Figure 5.3. The interaction of the displacer or piston with these constraints has been treated by assuming they are very stiff elastic limits, i.e., when the position of the piston or displacer exceeds a limit, a restoring force proportional to the distance by which the component has exceeded the limit is imposed on the component. The real system does behave in somewhat this manner, since if a component exceeds a limit, the limit is slightly deformed resulting in a restoring force on the component. In addition, there is a damping effect as the limits are approached which is due to gas flow between the colliding surfaces. A model which accounts for this effect was incorporated and it prevents sharp rebounds by the displacer, the most sensitive dynamic component.

Figure 5.3

POSITION CONSTRAINTS FOR PISTON AND
DISPLACER IN MODEL ENGINE

The initial problem statement included mechanical springs between the piston and cylinder case, and between the displacer and piston. Since there are no mechanical springs of significance in the engine, these spring constants were set to zero, and a gas spring was included between the displacer rod and piston. This gas spring is assumed to be isothermal and to communicate with the bounce space through a variable flow resistance which depends on the relative displacer position.

The initial problem formulation assumed that the total working space mass was constant. An additional mass equation has been added which allows for leakage between the compression space and bounce space. The leakage past the piston allows for inputting a mid-stroke port by specifying the relation between flow resistance and piston position.

5.3.2 Simulation Stability Improvements

The flow and heat transfer constants calculated based on the physical system result in time constants which are significantly shorter than the period of the engine. Since the step size of the simulation is set to a multiple (< 1.0) of the shortest time constant in the system, many steps are required to execute one cycle. This results in a running time of about 50 system seconds to simulate one cycle of the engine operation.

If the simulation is run with specified dynamics in order to check the thermodynamic calculations, the forces acting on the displacer are those shown schematically in Figure 5.4. No collisions are allowed for specified dynamics, therefore, collision forces are not shown. Since the displacer compression side area is about 89% of the total displacer area, the pressure difference across the displacer has a large impact on the displacer motion. The pressure difference across the displacer is due entirely to the drop across the heat exchangers. In early runs of the simulation, the flow resistance was set to an unrealistically high value, in an attempt to stabilize the solution. When a realistic value was used for this resistance, flow and

Fig. 5.4 Displacer Forces

pressure drop oscillations were encountered during part of the cycle. Examination of the forces on the displacer indicated that more work was going into pumping the gas through the heat exchangers than the thermodynamic cycle could input to the displacer.

Since the simulation did not result in a stable calculation of the pressure drop and no error in logic could be found, the fundamental calculation procedure was evaluated next. A simple system consisting of a thermal mass at some initial temperature which is suddenly exposed to an infinite reservoir at a different temperature (Figure 5.5) was studied to check the calculation procedure.

The differential equation for this system is

$$C \frac{dT}{d\tau} = q = \frac{1}{R} (T_{\infty} - T),$$

where

- R = Thermal resistance,
- C = Thermal Capacitance,
- T = Temperature of Mass,
- T_{∞} = Reservoir temperature,
- q = Heat flow rate,
- τ = Time,
- $\tau_c = RC$ = System time constant.

The exact solution for this equation is

$$\frac{T}{T_{\infty}} = 1 - \text{EXP} (-\tau/RC)$$

The actual equation which the BSAP code would solve is

$$T = T' + dT = T' + \frac{1}{RC} (T_{\infty} - T') d\tau$$

Fig. 5.5 Simple Thermal System

where T' is the temperature of the mass at the beginning of a time step and the time interval $d\tau$ is a multiplier ($G < 1$) times the minimum time constant $\tau_c = RC$, i.e.,

$$T = T' + (T_\infty - T')G$$

Stepping this equation through time results in the values given in Table 5.1 for Newton's integration method. While the steady state solution ($t \rightarrow \infty$) for both Newton's and the exact solution will agree, the paths taken for each can differ significantly. The error which occurs is a function of the multiplier used to determine the time step. Since the problem of interest is inherently a transient problem, it was thought that these path errors could significantly influence the results of calculation. An approach for improving the path accuracy, which will be referred to as the "predictor equation", was implemented in the problem statement.

The "predictor equations" assume the average temperature over the forward time step is,

$$\bar{T} = T' + \frac{1}{2} \frac{dT}{d\tau} d\tau$$

substituting $\bar{T}$ for T in equation

$$\frac{dT}{d\tau} = \frac{1}{RC} (T_\infty - T') - \frac{1}{2} \frac{1}{RC} \frac{dT}{d\tau} d\tau$$

Solving for $\frac{dT}{d\tau}$

$$\frac{dT}{d\tau} \left(1 + \frac{1}{2} \frac{d\tau}{RC} \right) = \frac{1}{RC} (T_\infty - T')$$

$$\frac{dT}{d\tau} = \frac{1}{RC + \frac{1}{2} d\tau} (T_\infty - T')$$

$$T = T' + \frac{(T_\infty - T') d\tau}{RC + \frac{1}{2} d\tau}$$

$$T = T' + \frac{(T_\infty - T') G \tau_c}{RC + \frac{1}{2} \tau_c}$$

TABLE 5.1
COMPARISON OF NUMERICAL
INTEGRATION METHODS

τ	Exact soln.	Newton Integration		Predictor Eqn.	
		$d\tau = .8\tau_c$	$d\tau = .4\tau_c$	$\tau = .8\tau_c$	$\tau = .4\tau_c$
.4	.33		.4		.33
.8	.55	.8	.64	.57	.56
1.2	.70		.78		.70
1.6	.80	.96	.87	.82	.80

The result of stepping this equation through time is also included in Table 5.1. Comparison with Newton's and the exact result shows that in this case the "predictor equation" has a much better path accuracy for a comparable time step size. This technique was implemented in the BSAP problem statement first for the mass balance equations, then for the energy equations. The results indicated that the "predictor equation" solution was more stable and allowed use of realistic values for the flow resistance. No difference was detected when this approach was implemented for the energy equations since the time constants for the energy nodes were significantly larger than the minimum time constants.

With a stable mass balance, the solution of the total network could proceed. Several runs were made to determine first if there are any other instabilities, and second if a solution could be obtained. No further instabilities were found, and the system appeared to be converging toward a solution. The basic problem remaining is that nodes such as the regenerator matrix and wall energy nodes have a longer time constant than the gas nodes, and are taking significantly longer than one simulation cycle for the steady state component of the solution to converge.

5.4 PRELIMINARY SOLUTION OF SYSTEM DYNAMICS

The basic approach to solving for the free dynamics in an FPSE is to write the equations of motion for each component and solve them as initial value problems along with the equations defining the thermodynamic system. When this approach is used, any errors in the problem definition or the numerical solution can cause the dynamics to follow an unrealistic path in the time-space plane. This occurred during initial running of the simulation and the results of erroneous equations or parameters were not easily analyzed when undesirable trajectories were followed. Therefore, in cases where the dynamics are not behaving properly it was found to be easier to specify the desired dynamics and compare the calculated forces to what they should

be. The equations can then be made to converge to the proper solution by modifying equations and/or parameters until the forces are made to agree. The agreement will only be approximate since the motion of the displacer and piston are only approximately sinusoidal, however, once approximate agreement is obtained, the actual differential equations may be solved for the coupled dynamics.

The present simulation results are close to a preliminary solution of system dynamics, see Figures 5.6 to 5.9. In Figure 5.9, the "theoretical force" is the force required to provide the desired motion, while the "actual force" is the force the program is currently predicting. From the figure it can be seen that in this run the program is fairly close to matching piston motion, but not displacer motion.

5.5 COMPONENT ANALYSIS

The BSAP model along with all other simulations known to MTI are based on an assumption of quasi-steady flow. In order to evaluate this assumption, a separate analysis of a periodic flow heat exchanger was performed.

The continuity, momentum and energy equations for a two-dimensional heat exchanger (Figure 5.10) were first written. The important parameter of these equations is the periodic flow Reynold's number which is defined as,

$$R_e = g \sqrt{\frac{w}{v}},$$

where

g = gap

w = frequency

v = dynamic viscosity

A closed form solution for these equations can be found where the Reynold's number is either very large or very small. For very small Reynold's numbers, the flow can be assumed to be steady for which solutions are available.

The Reynold's number for the engine and demonstrator are moderate to large numbers. Therefore, the equations were solved for large Reynold's numbers. It should be noted that the equations solved contain no allowance

DISPLACEMENT (IN)

Fig. 5.6 Displacement Vs. Time

Fig. 5.7 Pressure Vs. Time

Fig. 5.8 Temperature Vs. Time

Fig. 5.9 Force Vs. Time

Fig. 5.10 Two-Dimensional Heat Exchanger Model

for turbulence, so the solution is for periodic laminar flow. The results are plotted in Figure 5.11 in terms of the average Nusselt number vs the periodic flow Reynold's number. The steady flow solution is shown for a point of comparison and the hypothesized behavior for intermediate values of Reynold's number is shown.

5.6 DEFINITION OF ENGINE GEOMETRY

The free-piston Stirling engine being modeled in the BSAP computer simulation program is described in Section 4. In addition, the geometry of the dynamic and thermodynamic elements is further defined in Tables 5.2 and 5.3. The physical limits, placed within the engine, of the displacer and piston motions are shown in a plot of relative displacer position vs piston position in Figure 5.3. Also shown on the figure are lines of absolute displacer position. These geometry and physical limit restrictions of the engine are input to the BSAP computer program.

5.7 CONCLUSIONS & RECOMMENDATIONS

The BSAP simulation of the model free piston engine has been evaluated. The original 13 node model was found to result in a rather poor energy balance with 38.5% of the input heat unaccounted for. This is shown in Table 5.4 along with the computer running time per engine cycle. As previously discussed, the step size of the integration is a significant factor with a technique used, therefore, the 13 node model was run with half the base time step resulting in 21.1% of the total input energy unaccounted for. Since the resolution of the network (number of nodes) in finite element models also effects the accuracy of the simulation, the nodes in the heat exchangers were doubled (24 working gas nodes). This improved the energy balance over the 13 node simulation, but still required the short integration step size for an acceptable energy balance. A fundamental reason for this error is thought to be due to the method used to model the mass flows. This hypothesis is supported by the improved accuracy of the energy balance with reduced step size, since there will be a smaller mass change over a step in the heat exchanger nodes with

Fig. 5.11 Periodic Flow Heat Exchanger Analysis

Table 5.2

ENGINE GEOMETRY

Working fluid	Helium
Nominal Working Pressure	40 Bar
Displacer Data	
Mass	.380 kg (.836 lbs)
Body Diameter	56.5 mm (2.225 in)
Body Material	Stainless Steel/Aluminum
Body Seal Diameter	57.1 mm (2.25 in.)
Rod Diameter	17.5 mm (.75 in.)
Rod Material Substrate/Surface	Aluminum/Moly Disulfide
Maximum Stroke	2.85 cm
Displacer Gas Spring Volume	75 $\pm$ 4 cm ³
Piston Data	
Mass	7.628 kg (16.82 lbs)
Seal Diameter	57.1 mm (2.25 in.)
Seal Material	Carbon
Maximum Stroke	6 cm (2.364 in)
Mid Stroke Port (from in stop)	2.47 cm (.972 in.)
Engine Cylinder	
Mass	100.79 kg (222.2 lbs)
Bore Diameter	57.1 mm (2.25 in. nominal)
Bore Material	Chrome
Load Type	Double Acting He Compressor

Table 5.3

ENGINE COMPONENT DEFINITION

<u>Type</u>	<u>Heater</u>	<u>Regenerator</u>	<u>Cooler</u>	<u>Total</u>
Geometry	22 tubes	70 gm, .05 mm dia Random Stainless Steel Wool (Porosity 0.910)	135 .0508 x .376 cm	
Passage Diameter Or Gap	.257 cm	.0506 cm Hyd Dia	.0895 cm (Hyd Dia)	
Frontal Area (Gross)	1.137 cm ²	12.84 cm ² (Net 11.69 cm ²)	2.578 cm ³	
Flow Length	28.40 cm	7.75 cm	8.45 cm	
Volume (Tube)	32.29 cm ³	99.51 cm ³	21.78 cm ³	
(Misc.)	7.60 cm ³	-8.95 cm ³ (matrix)	7.82 cm ³	
(Total)	39.89 cm ³	90.56 cm ³	29.59 cm ³	167.87 cm ³
Surface Area/Length	17.73 cm ² /cm	924.4 cm ² /cm	115.21 cm ² /cm	
Surface Area/Frontal Area	442.9 cm ² /cm ²	612.8 cm ² /cm ²	377.6 cm ² /cm ²	
	<u>Expansion Space</u>	<u>Compression Space</u>		
Area	25.65 cm ²	22.80 cm ²		
"Neutral" Length	2.27 cm	2.0 cm		
Volume (Displaced)	58.23 cm ³	45.60 cm ³		
(Clearance)	7.07 cm ³	15.30 cm ³		
(Total)	65.30 cm ³	60.90 cm ³		126.20 cm ³
				294.07 cm ³

Table 5.4

ENERGY BALANCE ERROR

&

RUNNING "COST"

Number of Working Gas Energy Nodes	Integration Step Size	
	.4	.2
13	38.5% 107 s/cyc	21.1% 234 s/cyc
24	16.2% 143 s/cyc	1.8% 319 s/cyc

the shorter step size and, therefore, a better approximation to the mass flow distribution.

The present status of the simulation with respect to the experimental data is shown in Table 5.5. The simulation indicates very optimistic power and efficiency, though both have similar errors in energy balance. There are several possible reasons for this discrepancy. First, the errors in the mass balance could be affecting the accuracy of the energy equations as described above. Second, heat transfer and friction coefficients are assumed constant over the cycle, when in fact, they are functions of local velocity and temperature. The power output of the simulation is from a P-V integral in the working space so parasitic damping losses have not been accounted for. Finally, the leakage between the expansion and compression spaces has not been included in the model, and the parameters describing the leakage between the compression and bounce spaces correspond to much better seals than those in the model engine.

In addition to using parameters which more closely represent the model engine, the recommendations summarized in Table 5.6 are possible changes which could improve the accuracy of the model.

Table 5.5
COMPARISON OF SIMULATION RESULTS
WITH EXPERIMENTAL DATA

	BSAP MODEL*	EXPERIMENT**
Heat Input	2220	2295.9
Heat Rejected	1376	1747.5
Work Output	876	489
Efficiency	38.9%	23.2%
Unaccounted For	1.8%	2.6%

* 24 Nodes with .2 Time Step & Fixed Dynamics

** See Table 4.4

Table 5.6

RECOMMENDATION FOR IMPROVEMENT OF THE
FREE PISTON STIRLING ENGINE SIMULATION

<u>ITEM</u>	<u>PRESENT MODEL</u>	<u>RECOMMENDATION</u>	<u>COMMENT</u>
Mass/Momentum Equations	3 Mass Nodes	Mass/Momentum Resolution To Equal Energy Equations	
Mass/Momentum Treatment	Laminar Flow Loss Through Tube	Provide For Acceleration Effects	
Variable Area Ducts	None	Provide Equations Which Will Handle Variable Area Ducts	Will Improve Generality and May Be Needed for Demonstrator
Friction & Heat Transfer Coefficients	Constant	Velocity Dependent	
Unsteady Flow Friction & Heat Trans Coefficient	Correction To Constant Factor	Evaluate Other Models	Requires Two Dimensional Model of Flow
Property Values	Constant	Temperature Dependent as Required For Working Gas	
Availability/Entropy Generation Accounting	None	Include In Model	Will Help Identify Areas Where
Gas Spring Model	Constant Stiffness and Damping Coefficient	Evaluate Other Models	
Gas "Squish" In Working Spaces	Disp. Is Damped As It Approaches A Boundary	Evaluate Other Models	
Pre-Processor Subroutine	Minimal	Include	Reduce Burden In Setting Up Problem
Post Processor Subroutine	Minimal	Include	Reduce Burden In Analyzing Results and Clarify Output

APPENDICES

APPENDIX I

ALTERNATOR COMPUTER ANALYSIS

ACUP	FR	S	SS	VIAP	M	PF
1.000E+00	0.000E+01	0.000E-01	0.000E-01	2.200E+02	0.500E-01	1.000E+00
2.000E-02	0.850E+02	5.000E-01	2.400E-01	0.300E+01	0.000E+00	0.000E+00
1.000E-01	0.700E+01	0.500E+04	0.750E+02	0.500E-01	0.500E+01	0.000E+00
0.000E-02	1.850E-02	9.500E+04	1.100E+00	1.100E+00	1.000E-05	1.000E+00
0.000E+00	7.000E+01	1.400E-01	1.000E+00	0.	1.	1.
0.	0.	0.	0.	0.	0.	0.
FLR	FRI	POTIF1	POTIFC	FCR1	FCRC	FCX
1.200E+00	1.000E+00	7.000E-01	7.000E-01	1.000E+00	1.000E+00	1.000E+00
0.	0.	0.	0.	0.	0.	0.
DO	DCUM	DCUF	ALI	DLU	OSH	
3.073E+00	1.939E+00	2.883E+00	3.753E+00	9.003E+00	1.002E+01	
0.000E+00	0.000E+00	0.000E+00	0.000E+00	0.000E+00	0.000E+00	
3.107E+00	2.031E+00	2.085E+00	1.074E+00	5.984E-01	4.733E-01	
AXLD	AXLA	ADC	ALC	CH	CH	DRAF
4.410E-01	4.783E-01	2.692E+00	1.070E+00	8.173E-01	6.925E-01	7.048E-01
SLII	SLI	SLIC	SLC	SATL	SHH	
1.120E-01	1.120E-01	3.200E-02	4.432E-02	4.432E-02	1.918E+00	
DIR	DWIRDC	CF	EFFR	REGR		
0.007E-02	2.270E-02	3.876E+01	9.090E-01	7.502E-02		
NI	N	IRMS	IRML	NDC	IDC	
1.544E+03	2.401E+02	4.545E+00	0.	4.653E+03	3.542E-01	
FELP	FELS	FENIP	FENIS	NIDC	TMMF	SGNIDC
4.943E+00	8.417E+00	2.100E+02	3.268E+02	1.003E+03	1.648E+03	0.
ICUM	IDCUM	MLTC	MLDC	MLT	PMFL	PSGDC
7.917E+02	2.674E+03	7.800E+00	2.025E+01	2.013E+01	5.648E+05	0.
K	PMK	PSUM	QSUM	PSLUT	PSLAT	PSUG
1.306E+00	1.306E+00	2.404E+02	4.808E+02	9.101E+01	2.920E+01	0.
RDC	VGENU	VGEN	ALF	ALFO	EQR	EQX
1.892E+02	2.365E+02	2.335E+02	0.	1.353E-02	2.971E+00	7.275E+01
PCUM	PCAM	QQA	QDCUM	QUDC	B	WHP
8.306E+01	5.461E+01	1.094E-01	7.359E-01	1.555E-01	6.156E+04	7.114E+00
WCU	WCUCU	WCUTC	WPLNG	WSH	WLAM	WTOT
1.594E+01	1.483E+00	1.224E+01	1.057E+01	4.634E+00	2.645E+01	7.133E+01
WATTS	CWAT	CWTCC	CWTDC	PLEC	PLHY	
1.100E+03	3.311E+01	2.819E+01	2.373E+01	2.592E+00	1.248E+01	
BSTAT	BPLNG	BGAPAC	BGAPDC	BGAPMX	LAMS	
7.262E+04	6.962E+04	1.691E+04	4.001E+04	5.692E+04	6.374E+02	
R1	R2	WF2A2	WF2A2H	FCYCL	FSOL	
7.649E-01	2.141E-05	6.091E+03	4.097E+03	1.290E+02	1.263E+02	
CPA	COMA	CAPA	SCUR	VSCOR	PCUR	VPCUR
2.467E+00	2.961E-01	2.897E+00	-3.646E-05	3.307E+02	0.	0.

ALF IS NEGATIVE WHEN VEL LAGS FSOL AND VGEN LAGS I

VGENC		VG		VM	
1.108E+02 0.		3.101E+00 1.504E+02		3.101E+00 7.505E+01	
VXF		VIA		VIB	
0. -1.053E+02	0.	7.530E+01		3.651E+00 1.493E+01	
VTAPC		VE		VMG	
1.100E+02 0.		6.752E+00 1.053E+02		1.655E+02	
BISG		B12G			
0. 0.		4.545E+00 0.			
B11		B12		BIM	
4.545E+00 0.		3.297E+00 5.160E-02		1.249E+00 -5.160E-02	
CIRMS		CIRML		CIC	
Q. 0.	0.	0.	0.	0.	0.
Z2P		ZMP		ZCP	
2.248E-05 3.944E-04	0.	1.042E-03		1.183E-05 2.803E-04	
Z1AP		ZCCP		ZSG	
0. 2.875E-04	1.183E-05 5.738E-04	0.	0.	0.	
ZGP		Z1BP		ZEP	
1.183E-05 5.738E-04	1.393E-05 5.095E-05	2.576E-05 0.307E-04			
Z1A		Z1B		Z2	
0. 3.783E-04	1.326E-05 7.493E-05	2.141E-05 5.190E-04			
ZM		ZEA			
0. 9.061E-04	4.454E-04 0.				
ZXF		ZCC		THTA	
0. -0.307E-04	0. 0.	0.	0.		
VFTC		BARI			
4.190E-04 0.	2.621E+05 0.				
PHIA PHIB	PHI2 PHID	ANGA	FAA		
1.177E+05 2.332E+04	1.173E+05 2.774E+05	0.	7.050E+00		

Z1A,Z1B,Z2,ZM ARE AS COMPUTED
 Z1AP,Z1BP,Z2P,ZMP ARE CORRECTED BY FCR,FCX,& PUTIF

SIDPOL DATA

JI	DELJ(JI)	DEL	FSP 2 POLES
1	2.000E-04	5.000E-03	2.052E+00
2	4.000E-04	1.000E-02	4.105E+00
3	6.000E-04	1.500E-02	6.159E+00
4	8.000E-04	2.000E-02	8.214E+00
5	1.000E-03	2.500E-02	1.027E+01
6	1.200E-03	3.000E-02	1.233E+01
7	1.400E-03	3.500E-02	1.439E+01
8	1.600E-03	4.000E-02	1.646E+01
9	1.800E-03	4.500E-02	1.853E+01
10	2.000E-03	5.000E-02	2.060E+01

PROGRAM STOP AT 5810

CS D 8.17 UNITS

147116 06/04/76

147116 06/04/76

147116 06/04/76 LING NOMENCLATURE (IN ORDER APPEARING ON OUTPUT SHEET)

147116 06/04/76

147116 06/04/76 INPUT

147116 06/04/76

147116 06/04/76 KWHP - HP FOR MOTOR, KW FOR GENERATOR

147116 06/04/76 FR - FUNDAMENTAL FREQUENCY OF TERMINAL VOLTAGE, HZ

147116 06/04/76 S - STROKE (PEAK TO PEAK DISPLACEMENT) OF PLUNGER, IN

147116 06/04/76 SS - FACE WIDTH OF PLUNGER POLES, IN

147116 06/04/76 VTAP - RMS VALUE OF FUNDAMENTAL HARMONIC OF TERMINAL VOLTAGE, VOLTS

147116 06/04/76 M- RATIO OF DC AMPERE TURNS / AC AMPERE TURNS PER COIL

147116 06/04/76 PF - DESIGN (AC) POWER FACTOR, FOR MOTOR IN MOTOR DESIGN

147116 06/04/76 , FOR LOAD IN GENERATOR DESIGN

147116 06/04/76 G - RADIAL GAP BETWEEN PLUNGER & STATOR, IN

147116 06/04/76 IDENS- CURRENT DENSITY IN AC COILS, AMPS PER SQ IN

147116 06/04/76 PKF- PACKING FACTOR IN AC SLOT

147116 06/04/76 CCAR- RATIO OF COMP COIL COPPER AREA TO AC COIL COPPER AREA

147116 06/04/76 IHTT - BASE ANGLE OF COMP COIL SLOT

147116 06/04/76 KCOIL - CODE FOR WINDING SLOT,

147116 06/04/76 1- FILL TOTAL SLOT AREA

147116 06/04/76 2- FILL ONLY RECTANGULAR SECTION ABOVE "V"

147116 06/04/76 KSTAT - CODE FOR STATOR CONSTRUCTION

147116 06/04/76 1- STATOR IRON SOLID (INVOLUTE LAMS)

147116 06/04/76 2- STATOR LAMS RADIAL WITH GAPS

147116 06/04/76 IRIT - MAXIMUM RATIO OF PLUNGER CORE DIA TO POLE DIA

147116 06/04/76 (MAY BE RESET BY PROGRAM TO LOWER VALUE IF REQ'D TO HOLD FLUX LIMITS)

147116 06/04/76 VDC- VOLTAGE TO DC COIL

147116 06/04/76 BDC - DC FLUX DENSITY IN PLUNGER CORE , LINES/IN**2

147116 06/04/76 IDENC- CURRENT DENSITY IN DC COIL, AMPS/SQ IN

147116 06/04/76 PKD- PACKING FACTOR IN DC SLOT

147116 06/04/76 SLTANG - ANGLE OF STATOR TOOTH SLOT FROM BORE SURFACE, DEG

147116 06/04/76 LM - RADIAL DIMENSION AT TIP OF STATOR TOOTH GAP, IN

147116 06/04/76 FLAM - THICKNESS OF LAMINATION STOCK, IN

147116 06/04/76 FLAP- FLUX DENSITY IN STATOR LAMINATIONS

147116 06/04/76 HYSL- HYSTERESIS LOSS FACTOR, LAMINATIONS

147116 06/04/76 ECL- EDDY CURRENT LOSS FACTOR, LAMINATIONS

147116 06/04/76 CL- LAMINATION MATERIAL VOLUME RESISTIVITY, OHM IN 1 " CUBE

147116 06/04/76 KCON - CODE FOR CONDUCTORS: AC, COM COIL, DC

147116 06/04/76 1- COPPER; 2- ALUMINUM

147116 06/04/76 HNAAC- RATIO HEIGHT TO WIDTH OF AC SLOT

147116 06/04/76 TCOPC - TEMPERATURE IN COPPER CONDUCTORS, DEG C

147116 06/04/76 SLTK - RATIO STATOR SLOT GAP TO STROKE

147116 06/04/76 XFIX - PORTION OF REACTANCE TO BE CORRECTED BY SERIES CONDENSER

147116 06/04/76 YFIX - PORTION OF REACTANCE TO BE CORRECTED BY PARALLEL CONDENSER

147116 06/04/76 KMG - CODE FOR MOTOR RUN, -1 ; FOR GENERATOR RUN, +1

147116 06/04/76 KPRINT - CODE FOR PRINT OF CIRCUIT PARAMETERS & SIDEPU L DATA

147116 06/04/76 1 - YES; 0 - NO

147116 06/04/76 SUPG - SUPERGAP -- A NON-WORKING GAP IN THE STATOR LEGS FOR THE PURPOSE OF BLOCKING DOWN THE AC AMPERE TURNS IN THE

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1590 STATOR POLE REGION
 1600 SUPSF - A FACTOR WHICH ADJUSTS FOR TIME AVERAGING EFFECT OF THE
 1610 PERMEANCE OF THE SUPRGAP ON THE REQUIREMENT OF DC AMPERE
 1620 TURNS. (USE .75 UNLESS THE SLOT IS POSITIONED TO BE BY-PASSED
 1630 DURING THE HIGHER PORTIONS OF THE STROKE, IN WHICH CASE,
 1640 THE FACTOR APPROACHES VALUES OF .9)
 1650 FLK - DESIGN FACTOR FOR DC AMPERE TURNS IN IRON OF PLUNGER & STATOR
 1660 ADJUSTS FOR ACTUAL RELATIVE TO TABULAR DATA FOR 1010 STEEL &
 1670 M19 LAMINATION STOCK
 1680 FNL - DESIGN FACTOR FOR ADJUSTING EFFECT OF AC AMPERES ON DC AMPERE
 1690 TURNS RELATIVE TO THE THEORETICAL BASIS IN THE PROGRAM
 1700 POTIFM - DESIGN FACTOR FOR ADJUSTING PROGRAM EVALUATION OF AC COIL
 1710 SLOT REACTANCE
 1720 POTIFC - DESIGN FACTOR FOR ADJUSTING PROGRAM EVALUATION OF COMPENSATING
 1730 COIL REACTANCE
 1740 FCRN - DESIGN FACTOR FOR ADJUSTING PROGRAM EVALUATION OF AC COIL
 1750 RESISTANCE. THIS FACTOR CAN ALSO REPRESENT THE LOCAL IRON
 1760 LOSSES IN THE STATOR
 1770 FCRP - DESIGN FACTOR FOR ADJUSTING PROGRAM EVALUATION OF COMP COIL
 1780 RESISTANCE. THIS FACTOR CAN ALSO REPRESENT LOCAL
 1790 IRON LOSSES IN THE PLUNGER
 1800 FCX - DESIGN FACTOR FOR ADJUSTING PROGRAM EVALUATION OF POLE REGION
 1810 FLUX LEAKAGE FACTOR
 1820
 1830 OUTPUT
 1840
 1850 DG - OD OF PLUNGER POLES, IN
 1860 DCRM - ID OF COMPENSATING COILS, IN
 1870 DOPR - PLUNGER COPE DIA, IN
 1880 DZI - STATOR BORE DIA, IN
 1890 DLO - OD OF AC COIL, IN
 1900 DSH - OD OF STATOR, IN
 1910 SPAN- AXIAL DISTANCE BETWEEN COIL CENTERLINES, INCHES
 1920 WIDE - WIDTH OF COIL PLUS STATOR LEGS AT EACH SIDE, IN
 1930 H - HEIGHT OF AC SLOT, IN
 1940 (SPECIFICALLY FROM APEX OF EXTENDED SLOPED SIDES OF SLOT
 1950 RPO, TO SLOT OD)
 1960 AX - AXIAL DIMENSION OF AC SLOT, INCHES
 1970 SR - RADIAL DISTANCE FROM STATOR BORE TO START OF RECTANGULAR PORTION
 1980 OF AC SLOT, IN
 1990 TST - RADIAL THICKNESS OF LAMINATION OVER OD OF AC COILS
 2000 AXLO - STATOR LEG SIZED FOR MAXIMUM DC FLUX, IN
 2010 AXLA - STATOR LEG SIZED FOR HALF DC FLUX PLUS MAXIMUM AC FLUX
 2020 CDC - RADIAL DIMENSION OF DC COIL, INCHES
 2030 ACC - AXIAL DIMENSION OF DC COIL, INCHES
 2040 CH - RADIAL DIMENSION OF COMPENSATING COIL, INCHES
 2050 CL - AXIAL DIMENSION OF COMPENSATING COIL, INCHES
 2060 DPAT - RATIO OF DOPR TO DG
 2070 SLIT - GAP WIDTH OF AC SLOT, INCHES
 2080 SLT - MAXIMUM OF SLIT OR SAIL

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100 SLIC - GAP WIDTH OF COMPENSATING COIL SLOT, IN
 110 SATL - MAXIMUM OF SLIC OR SATL
 120 SATL - LIGHT OF RECTANGULAR PORTION OF AC SLOT, IN
 130 SATL - SATURATION GAP WIDTH OF COMPENSATING SLOT, INCHES
 140 DWIP - DIA OF AC WIRE, INCHES
 150 DWDC - DIA OF DC WIRE, INCHES
 160 C - ESTIMATE OF MATERIAL COST, \$
 170 EFF - MOTOR OR GENERATOR EFFICIENCY
 180 REG - VOLTAGE REGULATION FROM DESIGN POINT TO NO LOAD (GENERATOR)
 190 PFC - POWER FACTOR INCLUDING DC POWER
 2000 AT - AMPERE TURNS PER COIL, AC
 210 N - TURNS PER AC COIL
 220 IRMS - RMS CURRENT DRAWN BY MOTOR, AMPS
 230 IRML - RMS CURRENT AT LINE
 240 NDC - TURNS IN DC COIL
 250 IDC - CURRENT DRAWN BY DC COIL, AMPS
 260 FELP - LENGTH OF DC FLUX PATH IN PLUNGER IRON, IN
 270 FELS - LENGTH OF DC FLUX PATH IN STATOR IRON, IN
 280 FENIP - A.T. REQ'D OF DC COIL TO MAINTAIN BDC IN PLUNGER PATH
 290 FENIS - A.T. REQ'D OF DC COIL TO MAINTAIN BLAM IN STATOR PATH
 300 NIDC - A.T. REQ'D OF DC COIL TO MAINTAIN BGAPDC IN GAP
 3100 TWMF - TOTAL AMPERE TURNS REQ'D OF DC COIL INCLUDING THAT REQUIRED TO
 320 OFFSET ARMATURE REACTION & SUPERGAP, IF USED
 3300 SGNIDC - A.T. REQ'D OF DC COIL TO MAINTAIN DESIGN FLUX THRU SUPERGAP
 340 ICCM - CURRENT IN COMPENSATING COIL, AMPS
 350 IDCOM - CURRENT DENSITY IN COMPENSATING COIL, AMPS/SQ IN
 360 MLTC - MEAN LENGTH OF TURN, COMP COIL, INCHES
 370 MLDC - MEAN LENGTH OF TURN, DC COIL, INCHES
 380 MLT - MEAN LENGTH OF TURN, AC COIL, INCHES
 390 PMFL - TOTAL DC FLUX IN CORE, LINES
 400 PSGDC - PERMEANCE OF SUPERGAP TO DC CIRCUIT
 410 K - FLUX LEAKAGE FACTOR AT GAP, AC
 420 PMY - FLUX LEAKAGE FACTOR AT GAP, DC
 430 PSUP - PERMEANCE AROUND MAIN COIL, PLUNGER CENTERED
 440 OSUP - PERMEANCE AROUND DC LOOP
 450 PSLAT - AC NOT PERMEANCE LESS PSLAT
 460 PSLAT - PERMEANCE OF AC TEETH AT TIP
 470 PSUB - PERMEANCE OF TWO SUPERGAPS IN SERIES CIRCUIT AROUND AC COIL
 480 RDC - RESISTANCE OF DC COIL, OHMS
 490 VGEN - OPEN CIRCUIT VOLTAGE (GENERATOR), VOLTS
 500 VGEN - I MEV OF GENERATED VOLTAGE, VOLTS
 510 ALF - ANGLE BY WHICH VGEN LEADS CURRENT
 520 ALFO - PREVIOUS ITERATIVE STEP SOLUTION FOR ALF
 530 REQ - EQUIVALENT RESISTANCE OF MG, OHMS
 540 EX - EQUIVALENT REACTANCE OF MG, OHMS
 5500 PCOM - PERMEANCE OF COMPENSATING SLOT LESS PCAM
 560 PCAV - PERMEANCE OF COMPENSATING SLOT TEETH AT TIP
 570 CMA - WATTS PER SQ IN OF SURFACE AC COIL
 5800 QDCM - WATTS PER SQ IN OF SURFACE COMP COIL
 590 QDC - WATTS PER SQ IN OF SURFACE DC COIL

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2450 R- DESIGN AC FLUX DESIGN IN GAP (UNCOMPENSATED)
 2500 WHP - WEIGHT OF A HOLLOW PLUNGER
 2510 (CORP T WITHIN .02*DG OF DCM)
 2520 WCU- TOTAL LB OF COPPER IN AC COILS
 2530 WCUAC- TOTAL LB COPPER IN COMP COILS
 2540 WCUIC- LB OF COPPER IN DC COIL
 2550 WPLNG- WEIGHT OF PLUNGER, LB
 2560 WSH- WEIGHT OF STATOR LAMINATION SPACER, LB
 2570 WLAM- WEIGHT OF STATOR LAMINATIONS, LB
 2580 WTOT - WEIGHT OF MOTOR, LB
 2590 CWAT- COPPER LOSS, AC, WATTS
 2600 CWTC - COPPER LOSS IN COMPENSTING COILS, WATTS
 2610 CWDC- COPPER LOSS, DC, WATTS
 2620 PLEC- EDDY CURRENT LOSS IN STATOR LAMS, WATTS
 2630 PLHY- HYSTERESIS LOSS IN STATOR LAMS, WATTS
 2640 WFEPL - LOCAL IRON LOSSES IN PLUNGER, WATTS
 2650 WFEST - LOCAL IRON LOSSES IN STATOR, WATTS
 2660 BSTAT - LOCAL FLUX DENSITY IN STATOR TOOTH AT SLOT EDGE, LINES/IN**2
 2670 BPLNG- FLUX DENSITY IN NECK OF PLUNGER TEETH, LINES/SQ IN
 2680 BGAPAC- AC FLUX DENSITY IN GAP, LINES/ SQ IN
 2690 BGAPDC- DC FLUX DENSITY IN GAP, LINES/SQ IN
 2700 BGAPMX- COMBINED FLUX DENSITY IN GAP, LINES/SQ IN
 2710 LAMS- NUMBER OF LAMINATIONS IN CIRCUMFERENCE OF STATOR
 2720 PIN2 - RESISTANCE OF ONE AC COIL, OHMS
 2730 P2 - RESISTANCE OF ONE COMPENSATION COIL, OHMS
 2740 WF2A2 - PLUNGER WEIGHT *FR**2*AMPL**2
 2750 (A CRITERION FOR SPRING STIFFNESS REQUIREMENT)
 2760 WF2A2H - WF2A2 FOR A HOLLOW PLUNGER
 2770 FCYCL - HARMONIC FORCE REQ'D TO PRODUCE HORSEPOWER AT GIVEN STROKE
 2780 & FREQUENCY
 2790 FSOE - HARMONIC FORCE OUTPUT OF MOTOR (NOT NECESSARILY IN PHASE
 2800 WITH VEL)
 2810 CPA - AREA OF AC COIL SLOT, IN**2
 2820 CPA- AREA OF COMPENSATING COIL SLOT, IN**2
 2830 CAPA- AREA OF DC COIL SLOT, IN**2
 2840 SCOR - SERIES CAPACITOR CORRECTION, FARADS
 2850 VSCOR - VOLTAGE ACROSS SERIES CAPACITOR, VOLTS
 2860 PCOR - PARALLEL CAPACITOR CORRECTION, FARADS
 2870 VPCOR - VOLTAGE ACROSS PARALLEL CAPACITOR, VOLTS

APPENDIX II

CALCULATION OF TOTAL POWER DELIVERED TO PISTON

A momentum balance on the piston assembly requires: (See Figure A.2.1)

$$F_p = F_m + m_p x_{p/c}''$$

or in terms of observable quantities

$$F_p = F_m + m_p (x_p'' + x_c'')$$

where

- F_p - Force input to piston from thermodynamic cycle
- F_m - Measured force between power piston and load
- $x_{p/c}$ - Position of power piston with respect to cylinder
- x_c - Position of cylinder with respect to ground
- $', ''$ - First and second derivatives with respect to time
- m_p - Mass of piston
- x_p - Position of the power piston

Integrating over a cycle

$$\oint F_p dx_p = \oint F_m dx_p + \oint m_p (x_p'' + x_c'') x_p' dt$$

$$W_T = W_M + m_p \oint d\left(\frac{x_p'^2}{2}\right) + m_p \oint x_c'' x_p' dt$$

where

W_T - Total work

W_M - Measured work

Since $\oint d\left(\frac{x_p'^2}{2}\right) = 0$

this reduces to:

A.2.1 Phase Relation Between Piston Relative Velocity Acceleration

A.2.2 Model of Piston Assembly for Calculation of Total Power Output

$$W_T = W_M + m_p \int \dot{\phi} x_c'' x_p' dt$$

or if we define the correction term

$$W_C = m_p \int \dot{\phi} x_c'' x_p' dt$$

then

$$W_T = W_M + W_C$$

Now assuming both piston and cylinder motions are sinusoidal or: (See Figure A.2.2)

$$x_p' = V_p \sin Wt$$

$$x_c'' = A_c \sin(Wt + \phi)$$

then the correction term reduces to

$$W_C = \frac{1}{2} m_p V_p A_c \frac{\cos \phi}{f}$$

where

m_p - Piston mass

V_p - Piston velocity amplitude

A_c - Cylinder acceleration amplitude

ϕ - Lead angle between piston velocity and cylinder acceleration vectors

f - System frequency

The total power output is then found by multiplying the work by frequency,

$$P_T = P_M + \frac{1}{2} m_p V_p A_c \cos \phi.$$

APPENDIX III

GEOMETRIC ANALYSIS OF PHASE DIAGRAM

Assume $X_P = \frac{P}{2} \sin \omega t$

and $X_D = \frac{D}{2} \sin (\omega t + \phi) + D_O = (\frac{D}{2} \cos \phi) \sin \omega t + (\frac{D}{2} \sin \phi) \cos \omega t + D_O$

Then $X_r = X_D - X_P = (\frac{D}{2} \cos \phi - \frac{P}{2}) \sin \omega t + (\frac{D}{2} \sin \phi) \cos \omega t + D_O$

where

X_P = Piston position

P = Piston total stroke (peak to peak)

ω = Angular frequency

t = Time

X_D = Displacer position

D = Displacer total stroke (peak to peak)

ϕ = Phase angle between displacer and piston motion

D_O = Distance between mid points of displacer and piston motion

X_r = Displacer position relative to piston position

The parameters X_P and X_r are the abscissa and ordinate of the phase diagram shown in Figure A.3.1. It may be noted, therefore, that

Point		ωt	X_P	$X_r = X_D - X_P$
1	A	$(2n + 1)\pi$	0	$-\frac{D}{2} \sin \phi + D_O$
2	B	$2n\pi$	0	$\frac{D}{2} \sin \phi + D_O$
3	D	$(2n + \frac{1}{2})\pi$	$\frac{P}{2}$	$\frac{D}{2} \cos \phi - \frac{P}{2} + D_O$

Fig. A.3.1 Displacer Phase Diagram

The mean ordinate of Points A and B is the ordinate of point C and is equal to

$$X_{r_c} = \frac{X_{r_A} + X_{r_B}}{2} = D_o$$

The length of CB in Figure A.3.1 is equal to the ordinate of B minus the ordinate of C or

$$\begin{aligned} |CB| &= X_{r_B} - X_{r_C} \\ &= \left(\frac{D}{2} \sin \phi + D_o \right) - D_o \\ &= \frac{D}{2} \sin \phi \end{aligned}$$

The length of element FD is the ordinate of F minus the ordinate of D or

$$\begin{aligned} |FD| &= X_{r_F} - X_{r_D} \\ &= X_{r_C} - X_{r_D} \\ &= D_o - \left(\frac{D}{2} \cos \phi - \frac{P}{2} + D_o \right) \\ &= \frac{P}{2} - \frac{D}{2} \cos \phi \end{aligned}$$

The length of element CF is the abscissa of point F minus the abscissa of Point C

$$\begin{aligned} |CF| &= X_{PF} - X_{PC} \\ &= X_{P_F} - X_{P_A} \\ &= \frac{P}{2} - 0 \\ &= \frac{P}{2} \end{aligned}$$

Point E is located by swinging an arc of radius FD about point F. The length of EF is, therefore,

$$|EF| = |FD| = \frac{P}{2} - \frac{D}{2} \cos \phi$$

The length of element CE is the difference between length CF and length EF hence

$$\begin{aligned} |CE| &= |FD| - |EF| \\ &= \frac{P}{2} - \left(\frac{P}{2} - \frac{D}{2} \cos \phi \right) \\ &= \frac{D}{2} \cos \phi \end{aligned}$$

The elements CB and CE from a triangle with side $\frac{D}{2} \sin \phi$ and $\frac{D}{2} \cos \phi$ respectively, i.e.,

As a consequence $\angle CEB = \phi$ and $|EB| = \frac{D}{2}$.