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DESIGN STUDY OF A 15 kW FREE-PISTON STIRLING ENGINE — LINEAR ALTERNATOR FOR DISPERSED SOLAR ELECTRIC POWER SYSTEMS

George R. Dochat
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August 1979

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
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NOTICE

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16. Abstract

This study resulted in a conceptual design of a free piston solar Stirling engine-linear alternator which can be designed and developed to meet the requirements of a near-term solar test bed engine with minimum risks. The conceptual design was calculated to have an overall system efficiency of 38% and provide 15kW electric output. The free-piston engine design incorporates features such as gas bearings, close clearance seals and gas springs. This design is hermetically sealed to provide long life, reliability and maintenance free operation. An implementation assessment study performed as part of this study indicates that the free-piston solar Stirling engine-linear alternator can be manufactured at a reasonable price cost (direct labor plus material) of $2,500 per engine in production quantities of 25,000 units per year. Opportunity for significant reduction of cost is also identified.

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1.0 INTRODUCTION

This final report contains the results of a six-month study contract awarded by the NASA-Lewis Research Center to develop a conceptual design of a nominal 15 kW electric solar Stirling engine. Conceptual designs were evaluated and developed for both a free-piston and kinematic-type Stirling engine for small dispersed solar powered applications in identical parallel path programs. This volume contains only the results for the free-piston Stirling engine study. A separate volume contains similar information for the kinematic-type Stirling engine.

The study performed configuration definition studies, including a detailed parametric evaluation of the selected concepts, with a final ranking of all attractive configurations. Upon selection of the best configuration from an overall system viewpoint, the study developed a conceptual design of a near-term solar engine capable of providing a nominal 15 kW electric output. Paralleling the conceptual design, an implementation assessment of the selected design was performed defining the producibility (cost), durability and growth potential.

It is concluded that a Stirling engine offers high conversion efficiencies over other heat engines (>.40 percent) for small solar powered applications. This high efficiency will enable system costs to be reduced. The free-piston Stirling engine driving a linear alternator offers the potential for long life, inherent reliability and maintenance-free characteristics of a hermetically-sealed power unit.

1.1 Background

As part of the Solar Thermal Power Program in the Division of Solar Technology U.S. Department of Energy, studies and experiments are being conducted for Central Power Systems application and Dispersed Power Systems Applications. In support of these systems, the Advanced Technology Branch of the above division is conducting research on advanced systems and subsystems for increased efficiency and decreased cost of electricity from solar energy input. One of these systems is referred to as a point-focusing, parabolic dish collector-receiver. This type of solar collector-receiver, when coupled with a heat engine and generator set, can provide electrical energy for low power applications (kilowatt range). When coupled together in a much larger array they can produce power in the megawatt range. The NASA through the Jet Propulsion Laboratory (JPL) and the Lewis Research Center (LeRC) has the responsibility to conduct research and develop the concept for the Division of Solar Technology of DOE. The LeRC has the responsibility of developing the engines and power conversion equipment for the point-focus systems.

This study evaluates the concept definition of both free-piston and kinematic-type Stirling engines for single parabolic dish applications with a nominal output of 15 kWe. The Stirling engine system offers the potential for high solar energy to electrical power conversion efficiencies and therefore reduced collector size resulting in lower total system costs.

1.2 Objectives

The objective of this study program is to develop the concept definition of both a free-piston and kinematic-type solar Stirling engine/alternator for a single parabolic dish application. The study identifies the most attractive configurations for a 15 kWe Stirling engine with an electrical alternator having either three-phase or single-phase 120-208 volts 60 Hertz output. Based on intended use, the major objective is to develop a Stirling engine/alternator that meets above requirements with the highest efficiency since this will reduce overall system cost. The main goals of this study are as follows:

- Parametric analyses of attractive engine/alternator concepts
- Identification of an advanced solar Stirling engine configuration and its implementation assessment
- Conceptual design of near term solar Stirling engine/alternator

1.3 Scope

Four major tasks were established to accomplish the stated objectives. The following identifies these four tasks and a brief description of the scope of effort:

Task 1: Configuration Definition and Parametric Analyses: Parametrically assess various Stirling engine configurations for a range of heater and cooler temperatures over a broad-load profile and identify the most attractive engine configurations.

Task 2: Implementation Assessment: For the one configuration selected from Task 1, an assessment is made to determine its development and production implementation potential. This task evaluates present state-of-the-art, producibility, durability and growth potential.

Task 3: Conceptual Design: In a parallel effort with Task 2, a conceptual design for a near-term test bed engine is established.

Task 4: Engine-System Interface: While this study program concerns only the Stirling engine/alternator conceptual definitions, the requirements of the remainder of the solar power system (i.e., collector, heat receiver, system controls) are provided by NASA to obtain an optimum overall system configuration.

A more complete description of the program scope is contained in the respective sections of this report.

1.4 Requirements

The study requirements are divided into two areas: parametric analyses and conceptual design. It is desirable to have a Stirling engine with a system efficiency as high as possible to reduce collector size and hence overall system cost. The major program objective, therefore, is to obtain a engine/alternator system with highest possible overall system efficiency while meeting all other requirements.

The program specifications for the Task 1, Parametric Analysis and the Task 3, Conceptual Design are as follows:

	Parametric Analyses	Conceptual Design
Configuration:	Consider Three Concepts	Single 15 kW Size
Thermal Input:	Consider Both Direct and Indirect	General Electric Heat Pipe Receiver Design
Heat Metal Temperature:	Evaluate 650° to 1100°C	815°C
Cooler Water Temperature:	Evaluate 21° and 65°C	43°C
Working Fluid:	Not Specified	Helium
Heat Storage:	15 Minutes at Full Power	70 Minutes at Full Power Within Receiver

	Parametric Analyses	Conceptual Design
Engine Power Modulation:	Not Specified	Constant Power Input
Engine/Alternator Power Output:	Various Power Levels 25, 50, 75, and 100 Percent of 15 kW	15 kW Constant + Parasitics

1.5 Summary of Results

1.5.1 Engine Conceptual Design

This study resulted in a conceptual design of a free-piston solar Stirling engine-alternator which can be designed and developed to meet the requirements of a near-term solar test bed engine with minimum risks. The conceptual design developed under this study represents an evolution of previous engine designs that have been fabricated, assembled and tested by MTI.

The major design changes and significant differences from past designs are:

1. The heater head represents a novel heat exchanger specifically intended for sodium vapor condensation heat transfer within a heat pipe heat receiver. The heat exchanger is designed to be produced in castings and avoids braze joints which represent potential failure modes.
2. The linear alternator is an inversion of the conventional alternator design. The stator is inside the reciprocating magnetic structure which itself is attached to the power piston. This design reduces weight substantially by shrinking the diameter and wall thickness of the pressure vessel.

The conceptual solar engine-alternator design is shown in Figure 1-1. Conceptual design parameters are presented in Table 1-1. This engine was calculated to have an overall system efficiency of 38 percent and provide 15kW electric output. The free-piston engine design incorporates the following features to provide long life, reliability and maintenance-free operation:

- Gas Bearings - Eliminate Wear
- Close-Clearance Seals - No Lubricant, No Contamination, No Potential Failure Mechanism
- Gas Spring - No Mechanical Failure Mode
- Posted Displacer - Eliminate Need for Close Tolerances Between Piston and Displacer
- Internal Supplied Bearings - Eliminate External Compressor

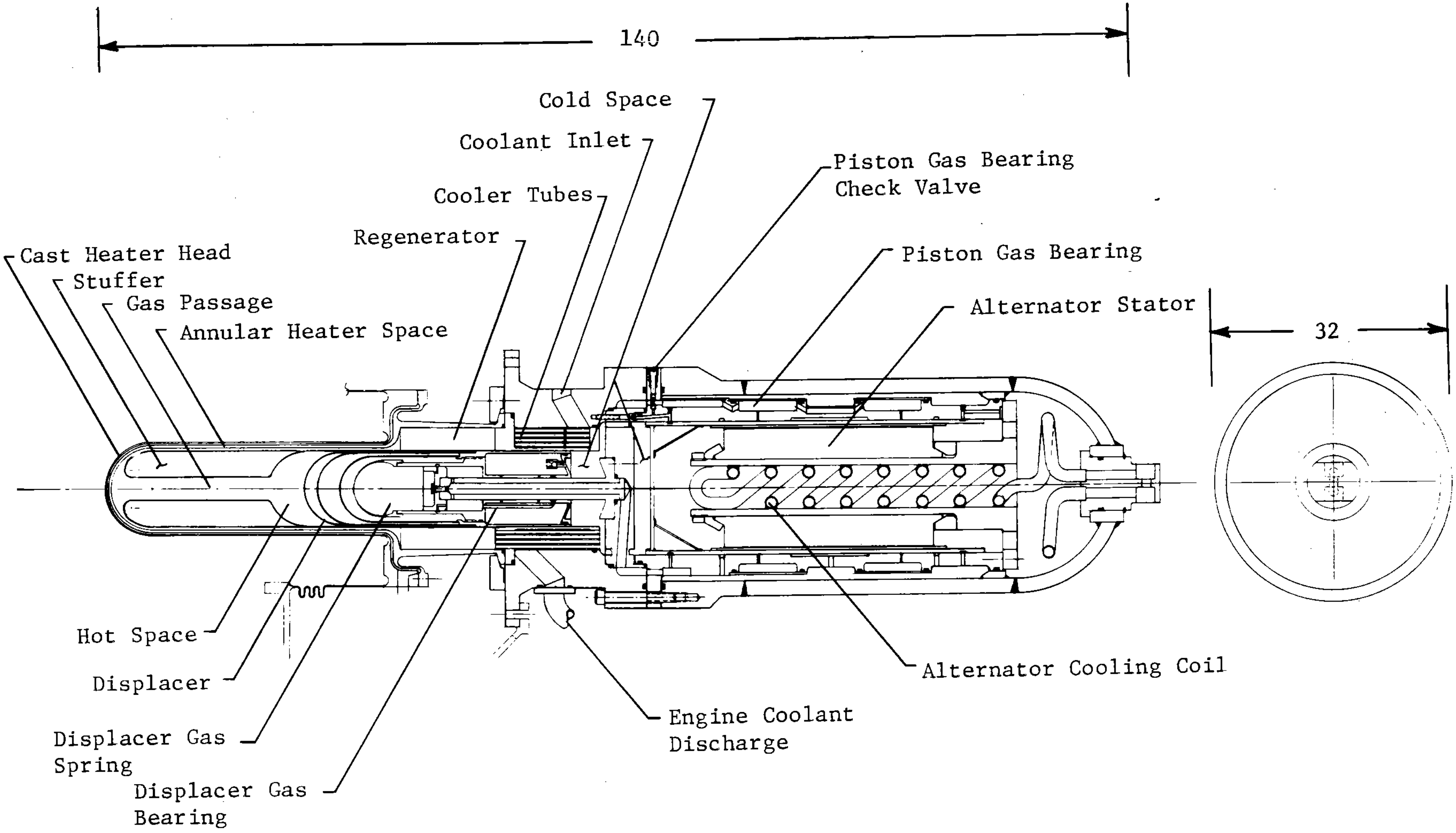

Fig. 1-1 15 kW Free-Piston Stirling Engine - Alternator Conceptual Design (Dimensions in cm)

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TABLE 1-1

15 kW FREE-PISTON SOLAR STIRLING ENGINE CONCEPTUAL DESIGN PARAMETERS

Engine power output	16,500 watts
Charge pressure	58.2 bars
Displacer amplitude	3.828 cm
Displacer phase angle	36.1°
Displacer rod diameter	2.83 cm
Cooler tube ID	3.49 mm
Cooler pumping loss	247 watts
Average cooler delta T	27.8°K
Regenerator length	12.7 cm
Regenerator void volume	1308 cm^3
Regenerator effectiveness	.99
Regenerator wire diameter	1.17×10^{-3} cm (.003)"
Heater tube number	42
Average heater delta T	64°C
Heat from heater to gas	39,760 watts
Overall engine (before alternator) thermal efficiency	41.5
Piston external spring	3.1×10^5 N/m^2
Displacer external spring	21×10^5 N/m^2
Cooler NTU	.52
Heater NTU	.76
Conduction and heat losses	1012 watts

The free-piston Stirling engine has no preferred orientation and will operate at any attitude. In addition, the engine interfaces readily with the given GE heat receiver concept as shown in Figure 1-2.

The results of the implementation study indicate that the near-term conceptual design is within the current or is an adaptation of state-of-the-art (Table 1-2). Areas where development is required and, in some cases, is already in progress at MTI are:

- Alternator - Permanent Magnet
- Controls - Engine-Alternator to Grid
  - Engine-Alternator to Receiver
- Regenerator - Cost Effective Regenerator Materials
- Heater Head - Interface with Condensing Sodium

The implementation assessment based on production quantities of 25,000 units per year indicated that the engine can be manufactured at a reasonable prime cost (direct labor plus material) of $2,500 per engine as shown in Figures 1-3 and 1-4. Extrapolation to 100,000 units show a cost of slightly over $2,000 per unit or about $140 per kW. It is clear that the alternator design utilizing samarium cobalt permanent magnetic material represents about 1/2 of the total system cost. MTI has explored alternative magnetic materials that may substantially reduce engine-alternator costs. A high magnetic flux permanent magnet material (Mn-Al-C) has recently been identified as a good substitute for the high cost samarium cobalt. Costing performed for this material is presented in Figure 1-5 and shows costs for the system to be $1300 per unit or less than $90 per kW.

In conclusion, the study developed a design of a free-piston solar Stirling engine-linear alternator that can provide required power, operate at high overall system efficiency, have potential for long life and high reliability, and can be produced at a reasonable cost with development effort. It is considered that the free-piston solar Stirling engine with the above advantages will accelerate the commercialization of small dispersed solar thermal power systems.

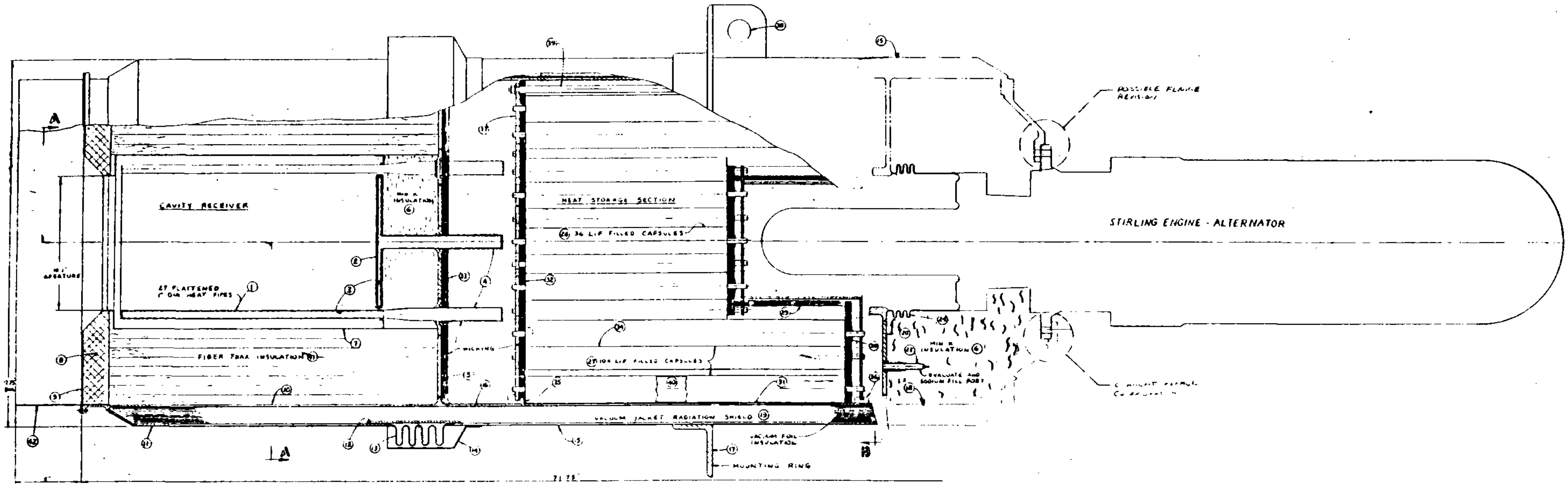

Fig. 1-2 Stirling Engine/Heat Receiver Interface

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TABLE 1-2

STATE-OF-THE-ART EVALUATION

Critical Component	Key Technology	Technology Status*
1) Heater Head	• Condensing Liquid Metal Heat Transfer	• Significant Improvement
	• Cast Heater Head	• Significant Improvement For High Heat Transfer
2) Regenerator	• High-Volume Processing With Effective Material Utilization	• Adaptation of State-of-the-Art
3) Bearing System	• Internally Supplied Bearing Gas	• Adaptation of Current Technology
	• Surface Coatings Techniques	• State-of-the-Art
4) Seals	• Close Tolerance Seal	• Extension of State-of the-Art for Life
5) Displacer Drive	• Posted Displacer and Gas Spring	• State-of-the-Art
6) Alternator		
- Plunger	• Rare Earth Permanent Magnet Manufacturability	• Adaptation of Current Technology
- Stator	• Manufacturing Technique (Microlamination, etc.)	• Adaptation of Current Technology
7) Control	• Engine/Alternator Stability Matching	• Adaptation of Current Technology
	• Displacer Gas Spring Volume Control	• Adaptation of Current Technology
	• Engine/Receiver Interface Control	• Significant Improvement

* Terms are defined in Section 3.1.2

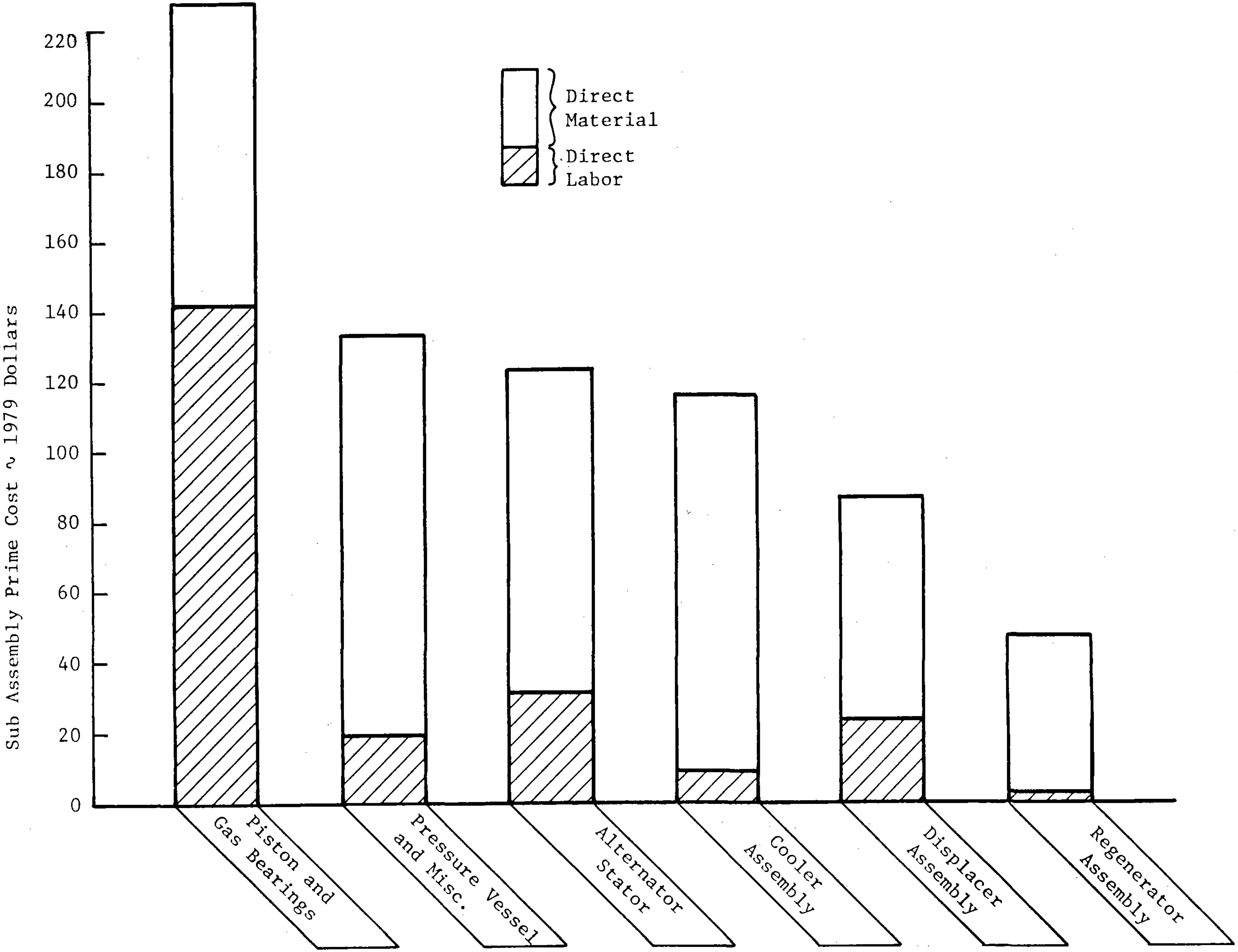
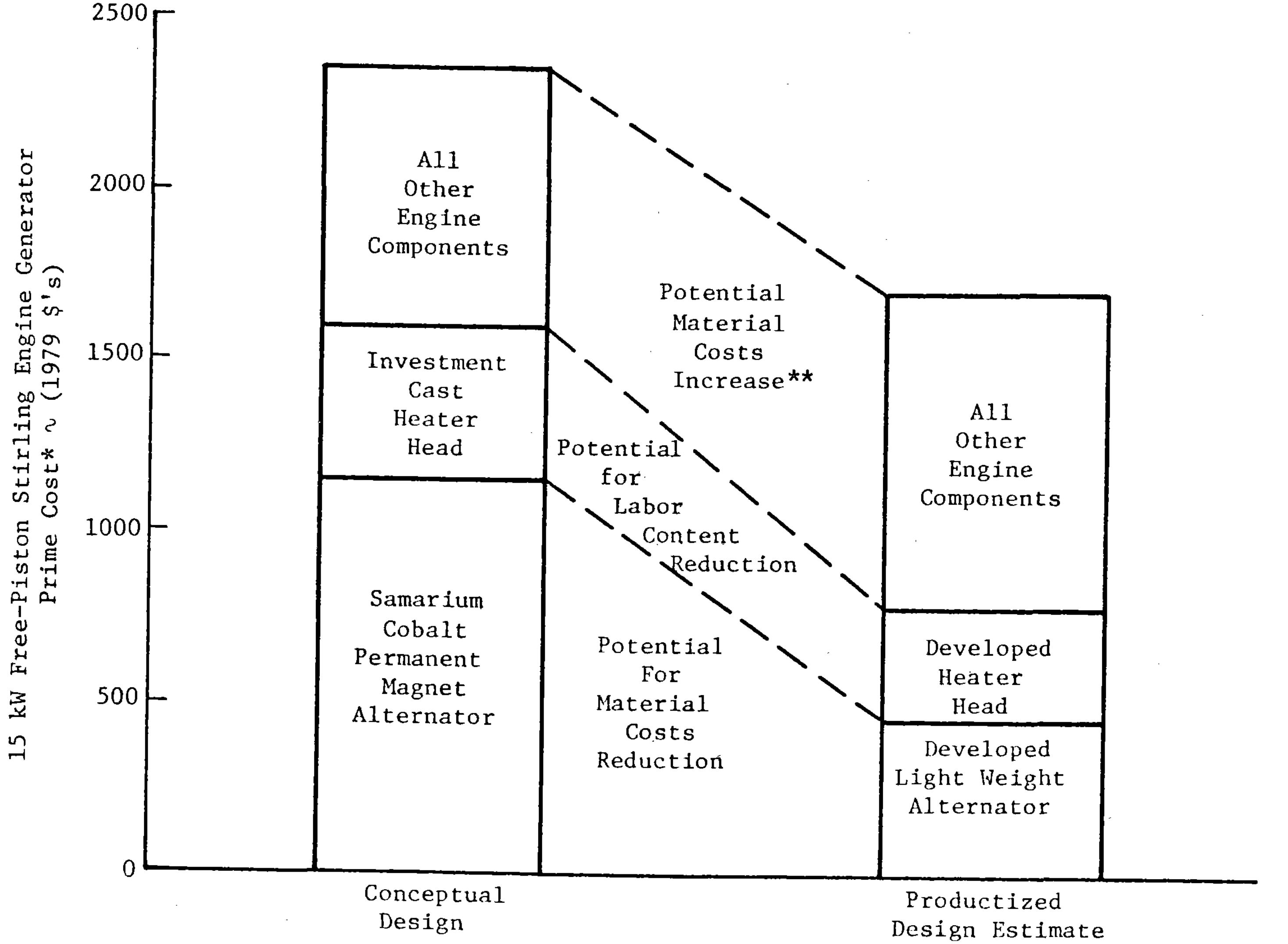


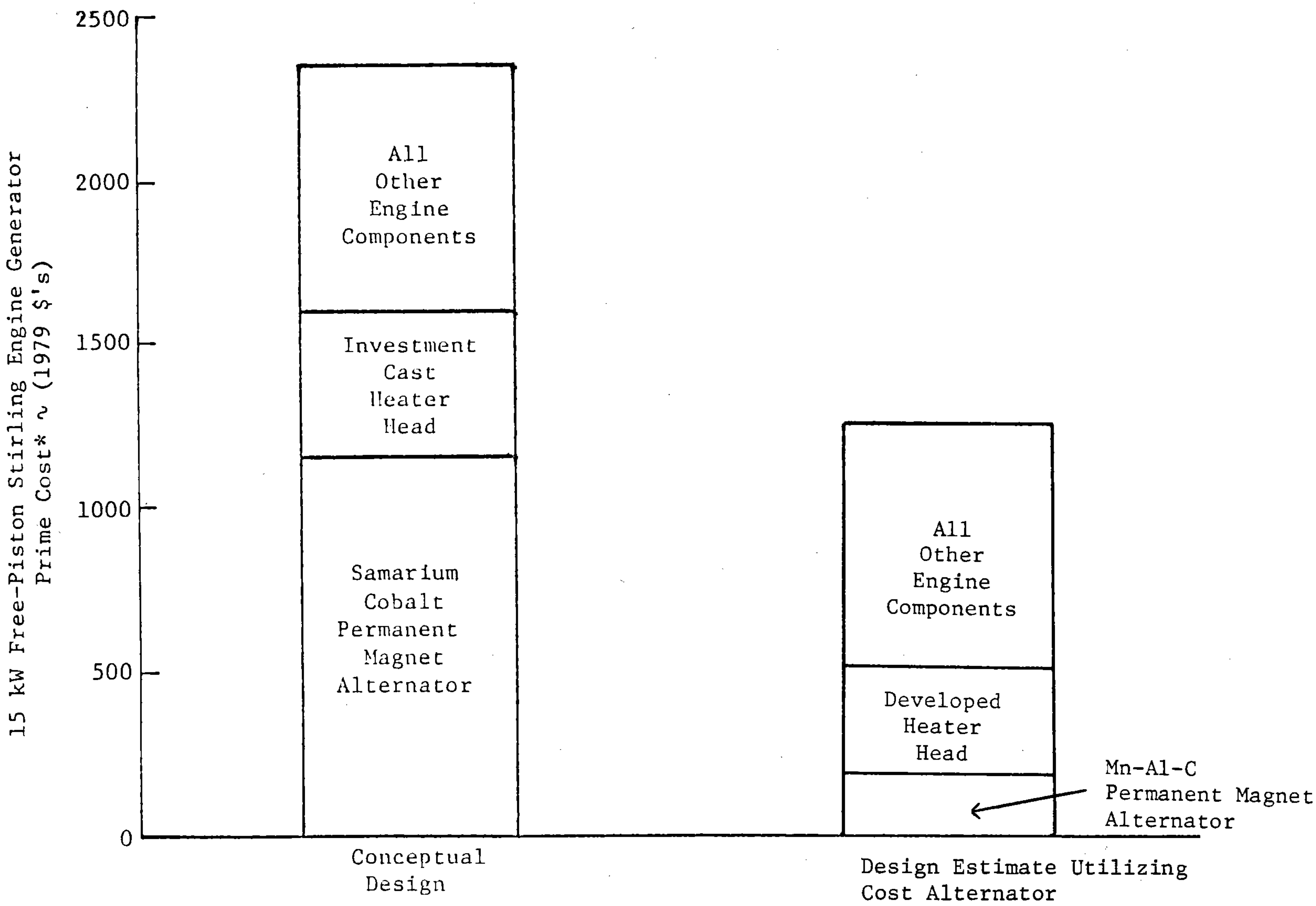
Fig. 1-3 Prime Cost Estimates for the 15 kW Solar Free-Piston Stirling Engine/Alternator Component Conceptual Design



*Prime Cost Includes Direct Labor and Materials Only

**Possible Compromise to Accommodate Potentially Heavier Alternator Plunger Design

Fig. 1-4 Prime Costs Estimates for the Conceptual Design and a Developed Engine/Alternator System



*Prime Cost Includes Direct Labor and Materials Only

Fig. 1-5 Prime Cost Estimates for the Conceptual Design and a Developed Engine/Alternator System for Production Quantities of 25,000 Per Year

2.0 CONFIGURATION DEFINITION STUDIES AND PARAMETRIC ANALYSES

2.1 Task Objective

The primary objective of this major task was to identify potential system configurations, parametrically analyze unique engine concepts, and determine the most attractive configuration for a 15 kWe Stirling engine with an electric alternator having a three-phase 120-208 V 60 Hz output.

2.2 Technical Approach

The technical approach to accomplish the task objective is to evaluate heat input concepts, determine design points, identify potential configurations, select three attractive concepts, perform parametric analyses of these concepts, and provide a ranking of the attractive configuration and recommendations.

To evaluate heat-input concepts, MTI solicited information from Minneapolis-Honeywell in the area of heat collectors/heat receivers and Dynatherm in the area of heat transport loop. Mutual agreement as to the best concept is determined based on the requirements for the Task 1 study; the major influence on choice of heat input concept is system capability for 15 minute thermal storage.

The solar insolation profile provided for the study is reviewed and evaluated to determine the percentage of peak power (i.e., engine design point) that would result in maximum kilowatt-hour output over an annual period.

The concept definitions are based on past experience regarding engine-alternator systems that have the potential for maximum overall system efficiency. Before initiation of the parametric analyses, the configurations identified are reduced to three (via an initial ranking of configurations). The parametric analysis is performed to provide the necessary performance data of each concept in comparison with others such that each configuration could be ranked and a recommendation made.

2.3 Heat Input Concepts

2.3.1 An Integrated Heat Receiver, Thermal Storage and Heat Transport System Concept

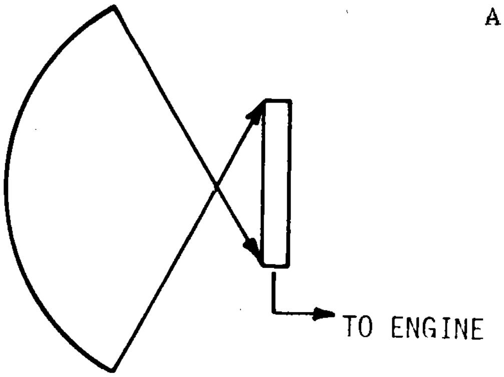
The objective of this subtask was to conceive a heat input concept which will transfer the heat from the collector to the Stirling engine working fluid minimizing any penalty on the engine's overall thermal efficiency. The heater head configuration of a Stirling engine is designed to accomplish its thermodynamic function of adding adequate heat to the working fluid with minimum dead volume and minimum pumping losses. Any deviation from the optimum configuration will result in degradation of engine performance. Therefore, the ideal heat receiving and heat transport system operates in such a manner that the optimum heater head configuration is maintained.

A major requirement of the system for Task I study was the capability of the thermal storage system to provide 15 kW output for 15 minutes. The thermal storage can be either built as an integral part of the engine/heat receiver system or at a remote location. Because of the high operating temperatures (around 871°C to 927°C) a remote heat storage will introduce additional losses and system complexity. Also in a system configuration where the thermal storage is remote and fixed, and the engine is mounted on the collector, flexible joints and liquid metal pumps at high temperatures would be required.

Based upon the aforementioned considerations and initial assessments, efforts were directed at conceiving a system which integrates heat receiving, heat transport and thermal storage as a single unit, as well as being sufficiently compact to be mounted on the collector. No further work was performed on the remote heat storage system because of anticipated losses, complexity and non-availibility of liquid metal pump in desired temperature range.

2.3.2 Heat Input Concept Selection

Several heat input concepts were examined and identified as follows. The initial assessments of advantages/disadvantages were reviewed with Minneapolis-Honeywell and Dynatherm in an effort to identify the most attractive heat input concept. The heat input concept selected is a liquid metal system (essentially a sodium pool boiler) using a molten salt for heat storage. The system is compact, lightweight and operates without a liquid metal pump. A summary of concepts considered with advantages/disadvantages leading to the selected concept are listed on the following pages.



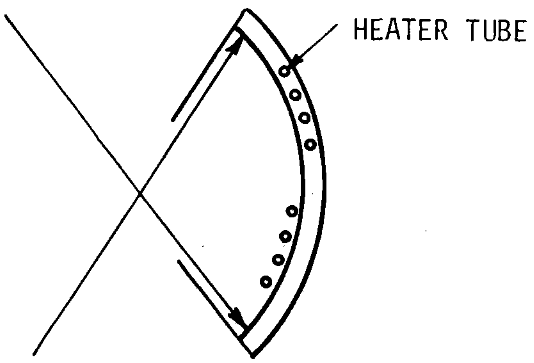
DIRECT RADIATION

A. External Absorber

This idea was abandoned due to the following disadvantages:

- Small heat storage capacity
- High heat losses
- Uneven heat flux
- Reradiation to nearby structures
- Secondary heat transport medium required
- Heater head would require modification

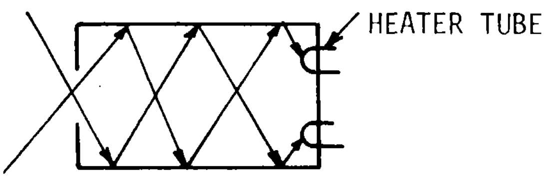
B. Internal Absorber



RADIATION - CAVITY RECEIVER

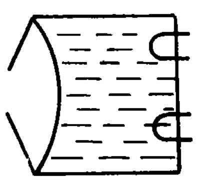
1. Direct Radiation: this idea was not pursued due to the following disadvantages:

   - small heat storage
   - uneven heat flux
   - hard to control heat flux
   - more complex heater tube geometry
   - longer heater tube - less engine efficiency



2. Cavity Receiver: this concept was rejected because of the following disadvantages:

   - low storage heat capacity to weight ratio
   - less efficient utilization of reradiation-tube area comparable to aperature area
   - increased heater tube area required



3. Liquid Immersed System: this system has the following advantages over the foregoing systems as follows:

   - nearly uniform temperature
   - high-heat transfer rate
   - higher heat storage capacity (higher liquid specific heat than metal)

- natural convection possible, no pumps required

No further considerations were given to the concept, however, due to the following disadvantages:

- low heat storage compared with heat of fusion or heat of vaporization system
- system temperature drops as heat is abstracted, resulting in low engine efficiency
- large mass required to maintain allowable temperature drop
- large mass and heavy system result in more heat loss area

C. <u>Two-Phase Heat Input System</u>

1. Liquid Vapor System: utilization of latent heat of vaporization as heat storage.

Several liquid metals have properties that are ideal for thermal storage such as:

- high latent heat of vaporization
- low vapor pressure
- high thermal conductivity
- high specific heat

Advantages of this system are:

- high heat transfer rate, optimum heater head configuration
- high critical heat flux for boiling, no hot spot in cavity
- temperature modulation possible
- power drain regulation possible

System disadvantages are:

- use of vapor-state results in poor volume to heat storage capability ratio
- use of liquid state results in poor mass-to-heat storage capability ratio

It is apparent that the liquid-vapor system is very attractive from various viewpoints, particularly its potential for operating temperature modulation and power drain regulation, both of which are not possible with any other system. Following several design iterations based on this concept, a close-coupled, two-phase, two-temperature sodium system was conceived, as shown in Figure 2-1. Sodium is used for both heat transport and thermal storage. By allowing a temperature drop from 982°C (container material limit) to 760°C, 10.4 kW-hr of heat can be stored in 132 kg of sodium. The liquid sodium is forced to flow to the expansion chamber by vapor pressure. This liquid sodium vaporizes in the expansion chamber and then yields heat to the heater head through condensation. The temperature at the expansion chamber is constantly maintained at the design point by keeping its pressure constant somewhere below the corresponding vapor pressure at the storage temperature. This can be accomplished with components similar to those used in a pressure regulator.

This system is simple, compact and lightweight. Additionally, this concept has the advantage that the flow of liquid sodium into the expansion chamber can be shut-off during the system shutdown period. This allows the temperature in the storage chamber to remain high while the temperatures in the expansion chamber and the engine remain low. No cooling of the engine is therefore required during the shutdown period. Unlike the other systems, temperature separation cannot be easily accomplished. Keeping the cooling water running during the shutdown period not only consumes power, but also transfers heat energy away from the storage which could otherwise be utilized.

The major disadvantage with this heat input concept is that a high-temperature liquid pump is required to pump the condensed sodium back to the storage at a higher pressure. A liquid metal pump with demonstrated capability of working at a temperature around 1400°F is presently not available. The time involved in developing such a pump lead to the decision whereby this heat input concept was not considered further.

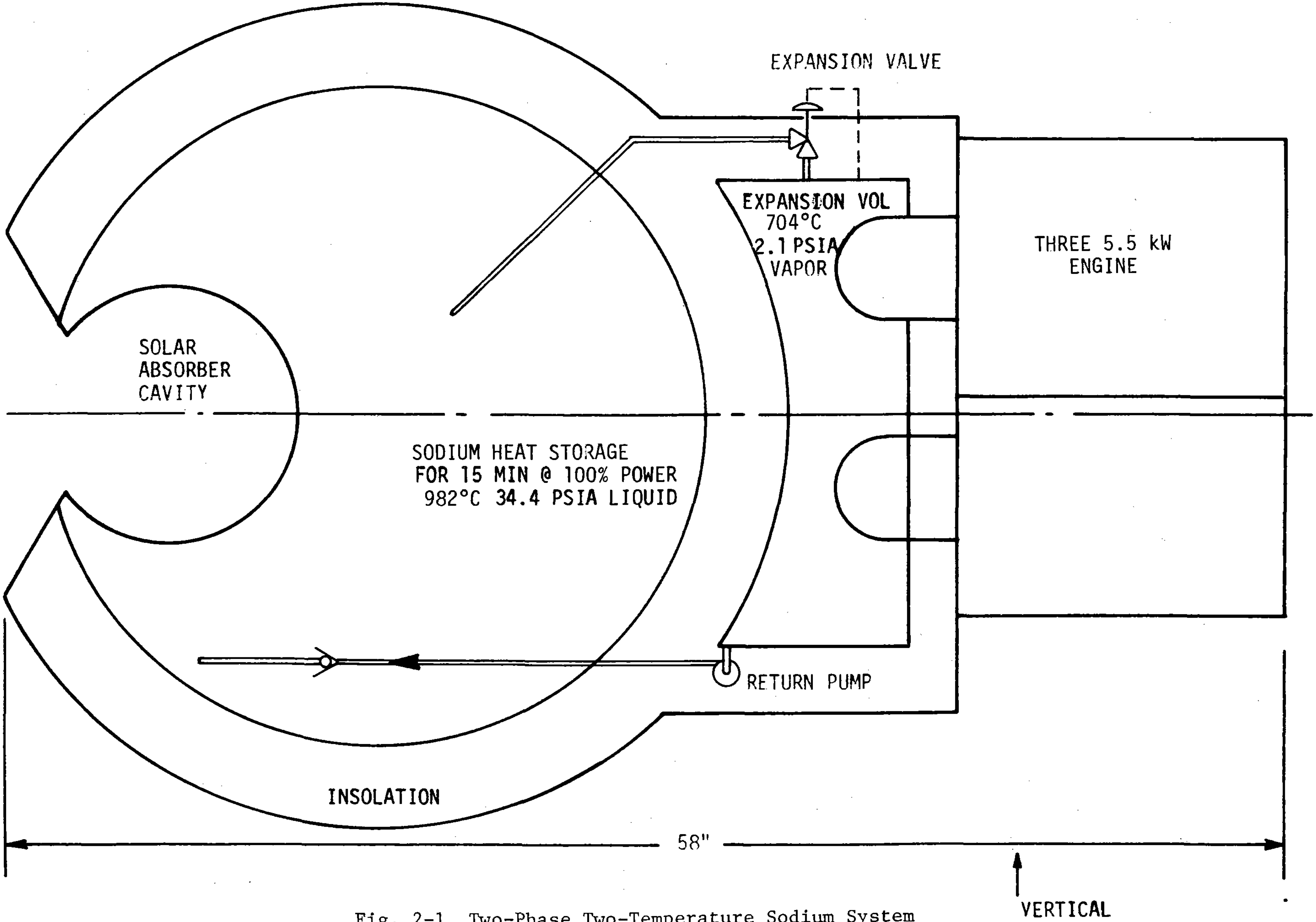


Fig. 2-1 Two-Phase Two-Temperature Sodium System

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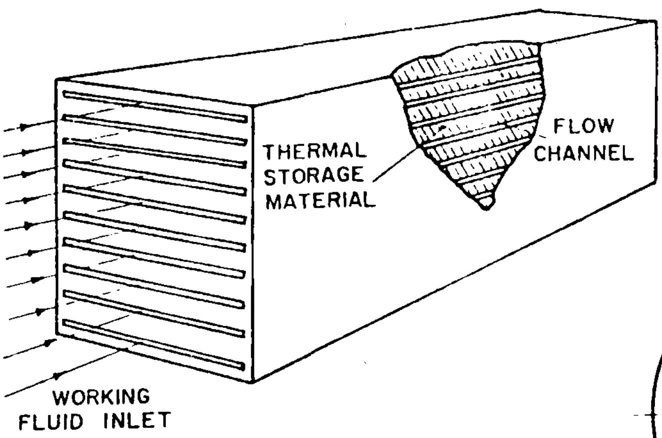
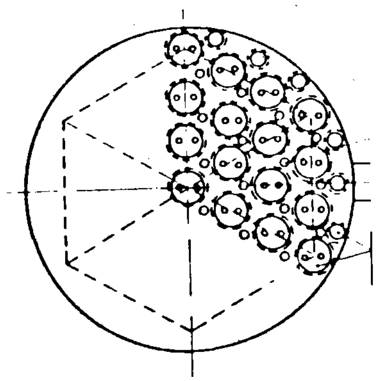
2. Solid-Liquid System or Molten Salt System: many salts have the properties that are ideal for thermal storage such as:

- high melting point for high temperature application
- high heat of fusion
- high specific heat

The disadvantages of such a system are:

- low thermal conductivity of salt; heat of fusion utilization requires a large heat transfer surface
- a second high-heat transfer working medium is required
- complex and costly system as shown in the following sketches





These disadvantages were overcome, and the related problems were solved leading to the pursuance of the recommended concept.

2.3.3 Recommended Heat Input Concept

The molten salt system has a high heat storage density. It also has the same advantages as the liquid-vapor system if liquid metal is used for the heat transport medium. The main disadvantage of this system, as discussed previously, is the low thermal conductivity of the salt. The ability to transfer heat into or out of the salt requires that a large heat transfer area has to be provided. To increase this heat transfer area, the salt is distributed in a large number of small thin-walled containers, usually small diameter cylinders. The numerous amount of cylinders makes this system complex and costly.

2.3.3.1 General Description of Recommended Concept. For the anticipated wide application of solar power systems, it is clear that the cost of each component is an important consideration. To make the molten salt system simple and economical, a method has been conceived whereby the system structure is simplified. Instead of using individual small-diameter cyclindrical containers, the proposed system utilized a single, continuous tube coiled in an enclosure as shown in Figure 2-2. An enclosed 2.54 cm outer diameter (OD) by 2.22 cm inner diameter (ID) tube, 51.8 meters in length provides 4.13 m^2 outer surface area for heat transfer. The heat of fusion of 35.8 kg of LiF alone is adequate to run the engine-alternator for 15 minutes at 15 kW full power output.

Liquid sodium operating at 871°C, 28°C above the melting point of LiF, was selected as the heat transport medium. The vapor pressure of sodium at 1600°F is 13.28 psia and the net pressure acting on the system is only 1.42 psi. This low pressure makes a thin wall structure possible. As a result, the temperature drop across the wall is low for the entire heat transfer surface thereby allowing a higher engine operating temperature and less thermal stress for the same receiver material temperature limitation. This system is compact, lightweight and economical and operates without a liquid metal pump and high receiver system efficiency. The system specifications are listed in Table 2-1.

2.3.3.2 Receiver System Heat Losses. The heat loss of the receiver system was calculated and results are as follows:

a. Heat Loss During Steady-State Operation:
   The total heat losses during steady-state operation is 4.2 kW.

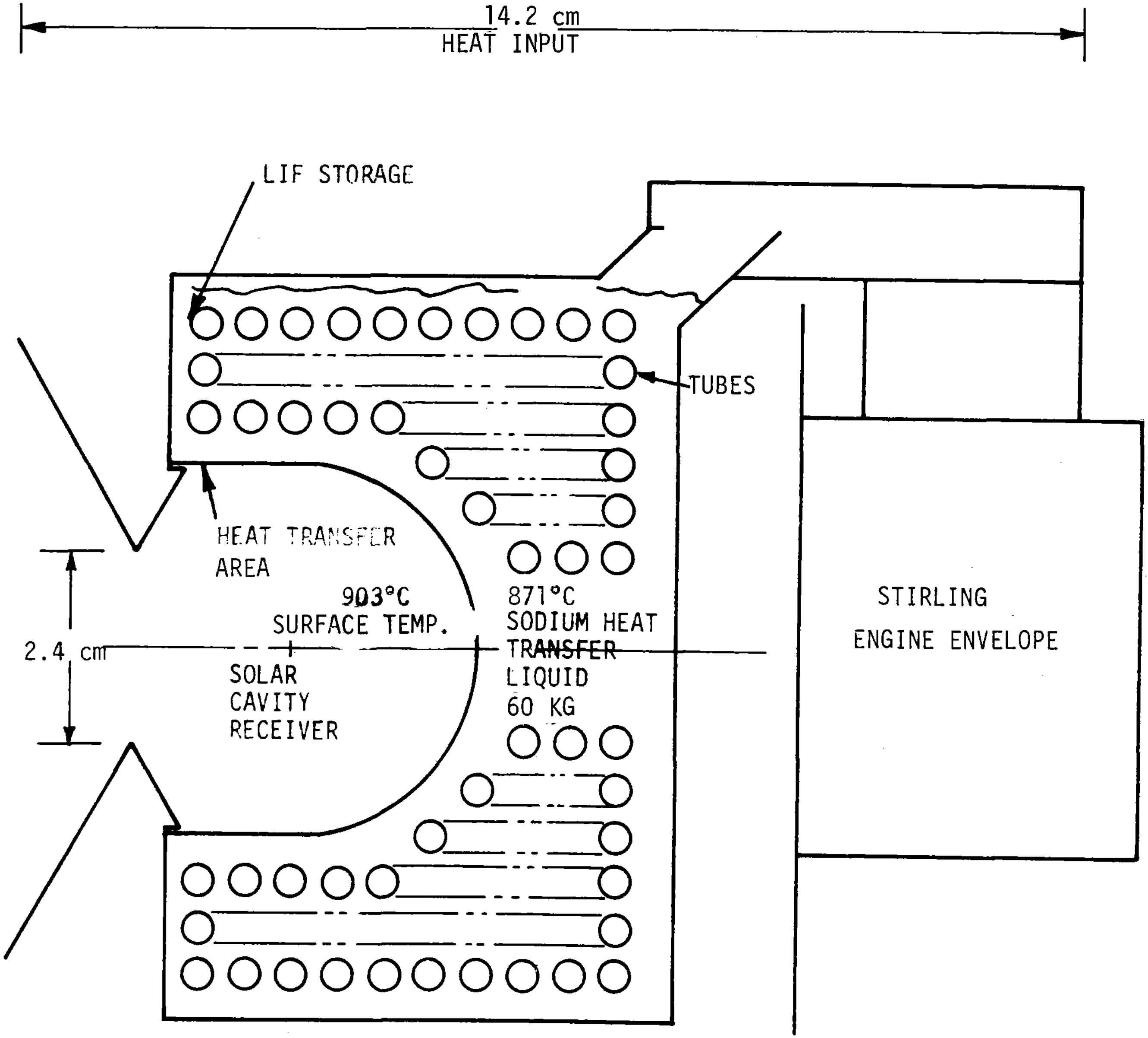


Fig. 2-2 Recommended Concept

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TABLE 2-1

SPECIFICATION OF A LiF-Na SYSTEM

LiF Weight	35.8 kg
LiF Volume at Liquid State	.02 m^3
Container Tube	2.54 cm OD x 2.22 cm ID x 51.8 m long
Tube Surface Area	4.13 m^2
Tube Weight	48 kg
Sodium Weight	75.3 kg
Operating Temperature	871°C
Heat Flux at Cavity	189,779 w/m^2
Temperature Drop Across Cavity Wall	31.8°C
Insulation	10 cm Knowoll 12 lb density
Receiver Efficiency	91 percent

b. Heat Loss During Shutdown Period:

During the shutdown period, no heat is added to the system but the heat loss persists as long as the system temperature is higher than the ambient temperature. To ensure that the sodium in the receiver system will not solidify (for a reasonable time period), system temperature as a function of time was calculated and plotted as shown in Figure 2-3.

2.3.3.3 System Efficiency.

a. Collector Size:

The collector size was determined as 8.2 meters in diameter, based on the maximum alternator output of 15 kW at maximum solar insolation and the following parameters:

• Maximum solar insolation	1.03 kW/m^2
• Receiver shadowing area	.557 m^2
• Stirling engine overall thermal efficiency (assumed)	40%
• Alternator efficiency (assumed)	90%
• System operating temperature	871°C
• Total receiver system heat loss	4.2 kW
• Collector reflectance	85%

b. System Efficiency:

With the available heat storage, the engine operates at a constant design temperature during cloud covering periods. The heat loss remains constant due to constant system temperature. As the solar insolation decreases, the heat input decreases but the heat loss is constant. System efficiency therefore decreases as the insolation decreases. The system efficiency as a function of solar insolation was calculated and is shown in Figure 2-4.

2.3.3.4 System Performance. Using the June 21, 1976 solar insolation profile in Lanchester, California as an example, the receiver temperature variation, power output, insolation outer surface temperature and receiver system heat

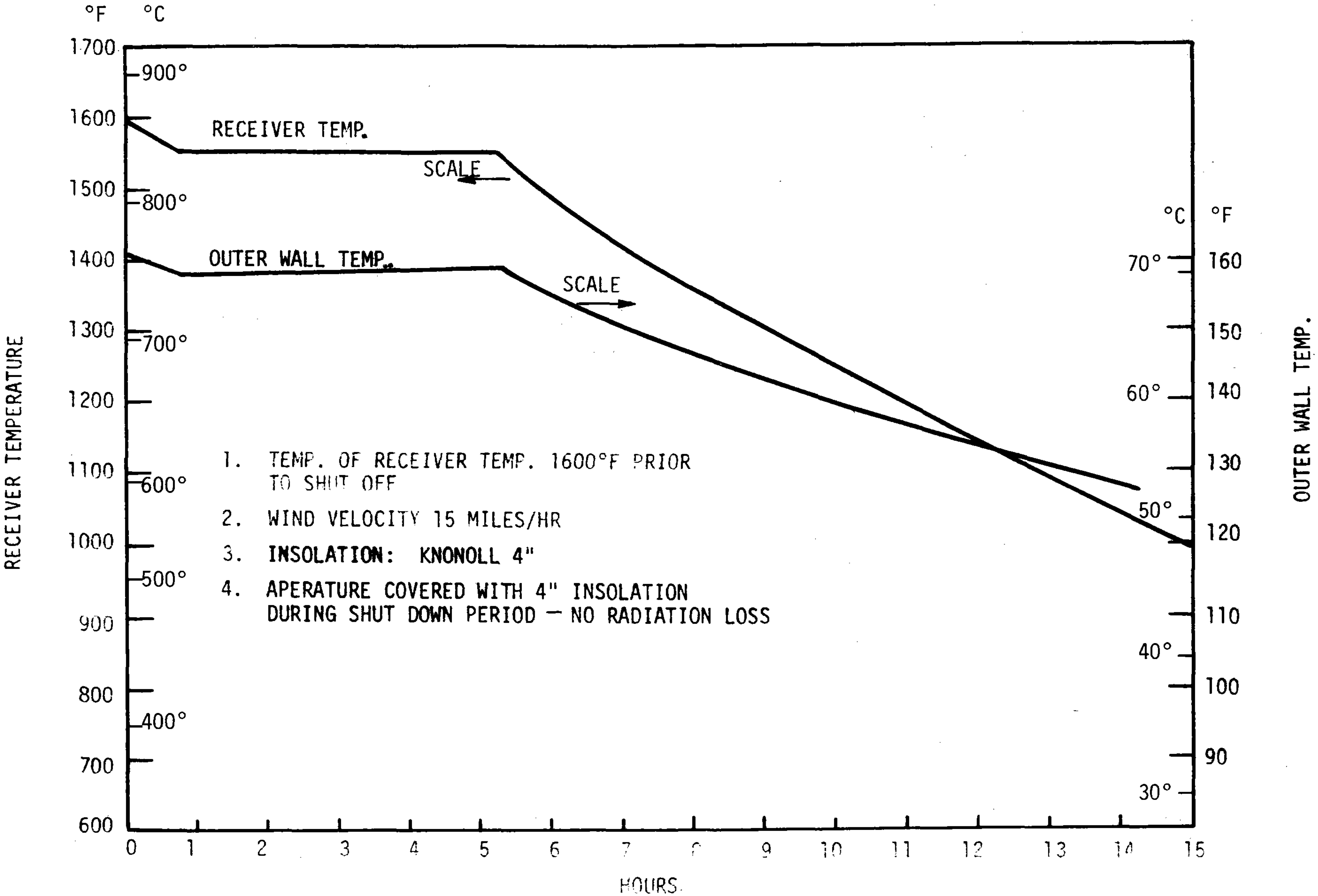

Fig. 2-3 Received Temperature Versus Hours After Shutdown

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1. COLLECTOR REFLECTANCE 85%
   TRACKING ERROR IGNORED
2. RECEIVER LOSSES
   4" KNOWALL INSULATION 12 LBS DENSITY
   15 MILE/HR WIND
   AMBIENT TEMPERATURE 70°F

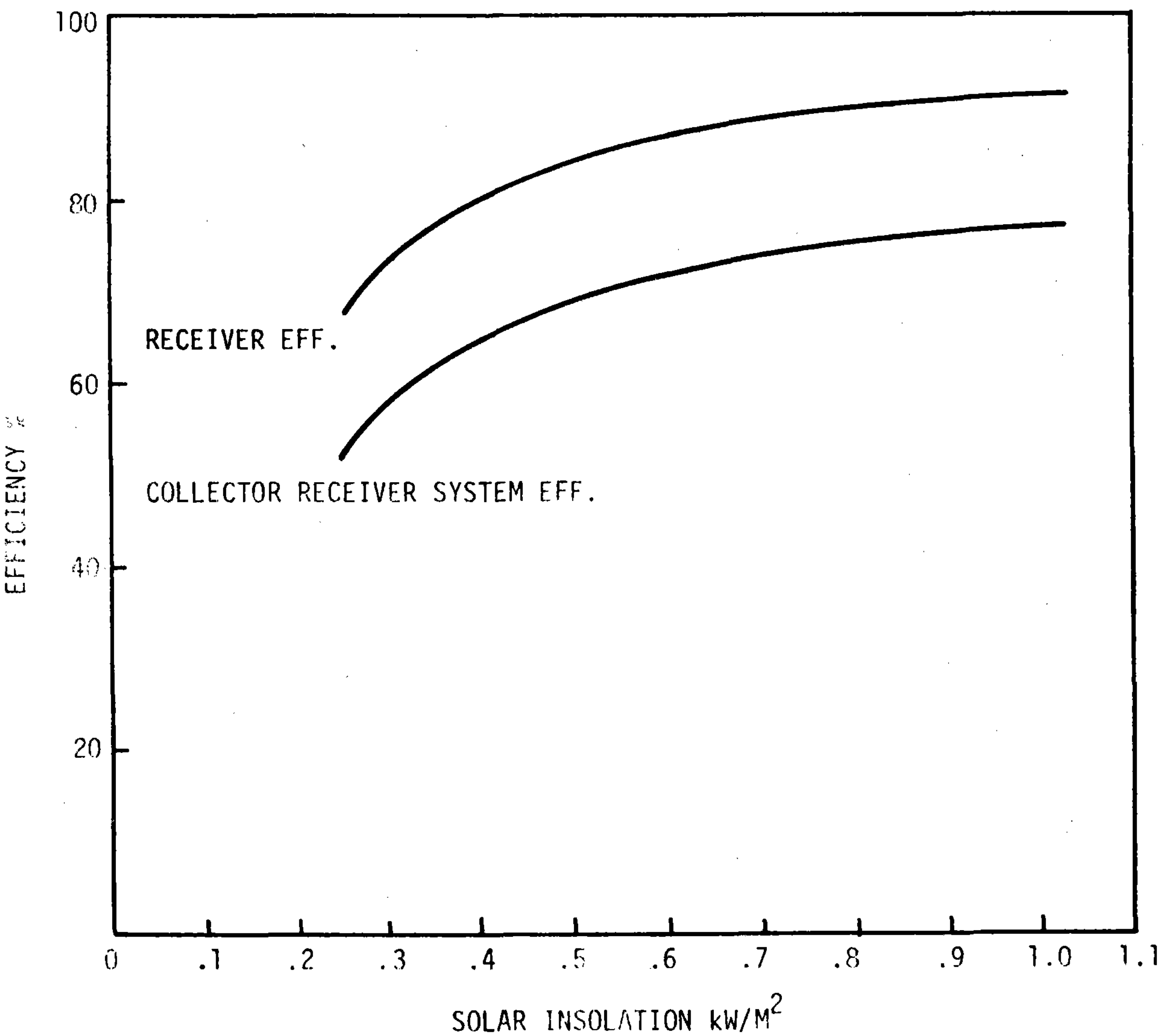

Fig. 2-4 Collector-Receiver Efficiency Versus Solar Insolation

loss were calculated and are plotted versus time in hours as shown in Figure 2-5.

2.4 Engine Design Point Selection

The engine-alternator maximum output capacity of 15 kW at maximum solar insolation is the given design condition. Regulating output at lower power levels according to the insolation profile was the original design philosophy. For a given maximum output, the engine overall thermal efficiency can be maximized at any narrow band of power level but cannot be maintained across the full power range. It is most important to select the engine design point at a power level where the engine's highest thermal efficiency coincides with the solar insolation level of the highest available annual energy. Although the solar energy is free the lower engine efficiency will require a larger collector and receiver system as well as the engine itself (assuming fixed alternator efficiency). It is considered that a low capital cost solar power generating system is the key to its commercialization. Since a major portion of solar system cost is within the collector/receiver system, it is important to maximize engine efficiency.

The major considerations for selecting the engine design point are:

- Insolation distribution over the year
- Collector-receiver system characteristics
- Engine characteristics
- Load characteristics

The collector-receiver characteristics have been discussed in the foregoing section. The alternator output is assumed to be connected to a utility grid system. The effect of this type of loading on engine performance is discussed in Subsection 4.7.1, System Stability. Only engine characteristics and solar insolation require the following considerations:

Solar Insolation Distribution. A copy of a detailed statistical summary for the solar insolation distribution in Lanchester, California was provided to MTI. This summary was the basis for the selection of the engine design point. The information needed is the accummulated hours in a year for each insolation

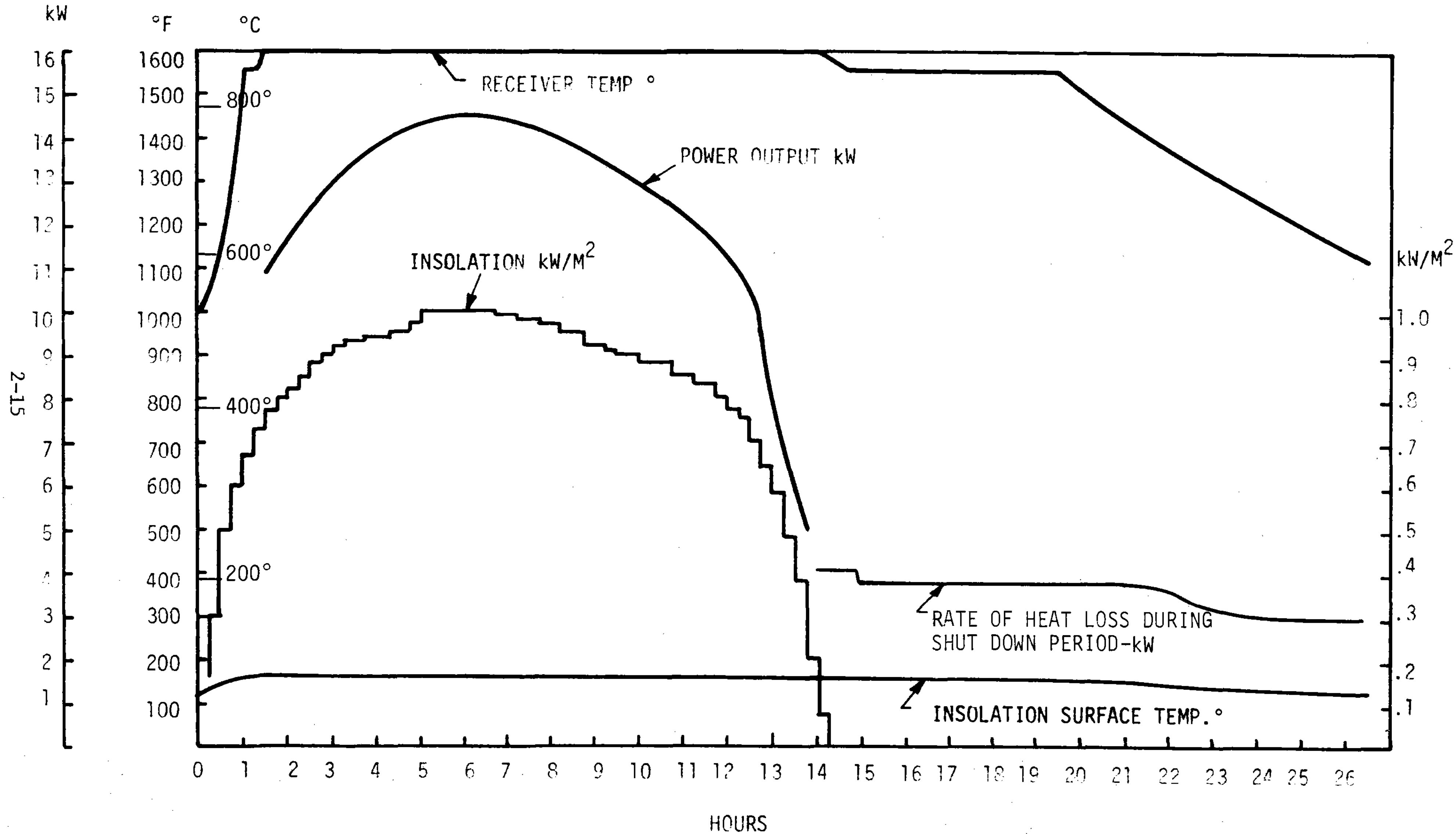

Fig. 2-5 Receiver System Performance

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level and is not directly available but can be obtained by utilizing the plot of insolation flux versus average hours per day for each month as shown in Figure 2-6. For the purpose of analysis, the insolation flux density was divided into ten levels between 0.5 kW/m^2 and 1.0 kW/m^2 by an increment of .05 kW/m^2 and three levels between .2 kW/m^2 and .5 kW/m^2 with an increment of 0.1 kW/m^2 plus those above 1 kW/m^2.

At each insolation level, the average hours per day were scaled from the plot and then multiplied by the number of days in that month. For example, in the month of March, the total hours at insolation between .9 kW/m^2 and .95 kW/m^2 is 1.64 hours x 31 days = 50.84 hours, as shown in Figure 2-6. This process was repeated for the fourteen insolation levels for twelve months and the results are summarized in Table 2-2. With the annual total hours at each insolation level so calculated, the annual kW hours available to the collector per each square meter of the collector area is obtained by simply multiplying the total annual hours by the average insolation in that range. For example, in the month of March available kW-hr to the collector at insolation levels between .9 kW/m^2 and .95 kW/m^2 is 563.52 hours x $\frac{.9 + .95}{2}$ = 521.25. The available energy to the receiver is reduced to 443.06, which is 85 percent of 521.25, representing reflectance of the collector.

The operating temperature of the recommended receiver system is fixed at 1600°F (as well as the heat loss) by adjusting the power output level according to the solar insolation level. For unit consistency in Table 2-2, kW hr/m^2 of collector area, the heat loss due to radiation and convection at the receiver is converted into loss of energy in kW hr/m^2 at the collector area. The total receiver heat loss is 4.202 kW-hr/hr equivalent to 4.202 kW-hr/hr ÷ 52.34 m^2 or .08028 kW-hr/hr m^2 at the collector independent of insolation level. In the above example, the 443.06 kW-hr available at the receiver are collected in 563.52 hours. Therefore, the energy available to the engine in the month of March is 443.06 kW-hr/m^2 minus (563.12 hours x .08028 kW hr/hr m^2) = 397.83 kW hr/m^2. As previously stated, the engine alternator system is designed to develop 15 kW output at maximum solar insolation. At each lower insolation level, the alternator output is listed in Table 2-2.

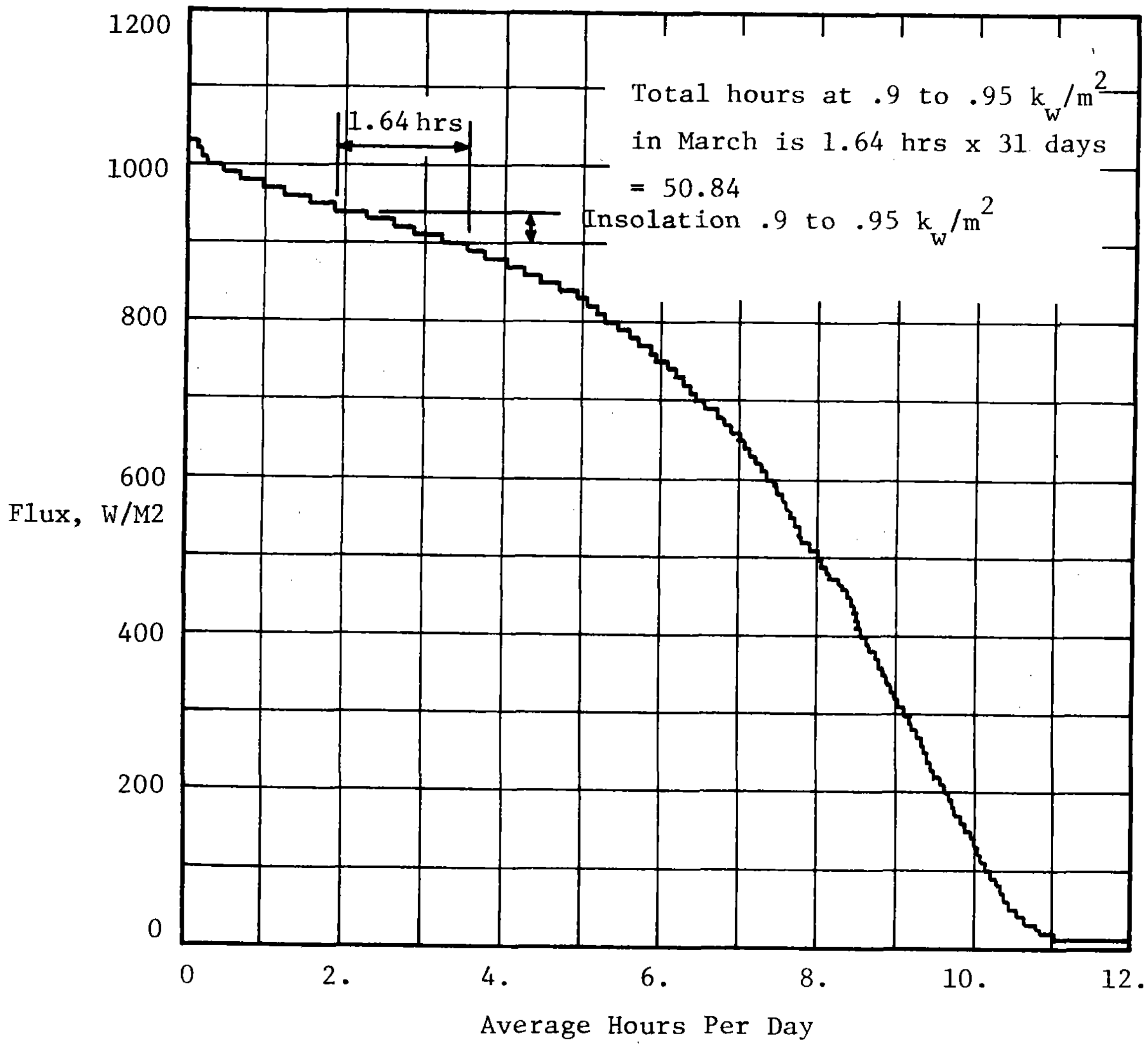

Fig. 2-6 Typical Insolution Versus Hours Per Day

TABLE 2-2

ANNUAL DIRECT INSOLATION IN LANCASTER, CALIFORNIA

Insolation / Hours / Month	Over 1 kW/m² Flux	hours	.95-1	.9-.95	.85-.9	.8-.85	.75-.8	.7-.75	.65-.7	.6-.65	.55-.6	.5-.55	.4-.5	.3-.4	.2-.3	Total Hours
JAN.	1.01	2.17	47.12	53.01	38.75	26.35	19.84	17.98	13.02	7.13	10.23	6.82	13.64	13.33	9.61	279.
FEB.	1.012	5.80	45.82	26.39	19.72	14.21	13.92	11.89	10.02	11.31	7.25	6.67	18.85	15.37	13.05	220.27
MARCH	1.031	7.75	49.91	50.84	35.65	23.25	21.08	14.57	16.43	10.85	8.68	8.99	19.22	16.43	15.50	299.15
APRIL	1.01	.9	41.10	42.30	50.40	32.10	24.60	21.60	17.40	9.60	18.00	7.50	18.30	16.20	17.40 EST.	317.40
MAY	1.01	1.24	28.21	24.80	72.85	43.40	35.65	24.49	22.94	17.98	15.19	13.33	9.92 EST.	22.01 EST.	21.70	353.71
JUNE	1.01	2.1	44.70	73.20	56.70	38.70	24.60	22.80	18.60	15.90	12.6	10.8	18.60	21.30	21.00 EST.	381.60
JULY		0	24.18	37.82	48.98	44.02	37.20	28.21	24.49	17.05	17.05	15.50	21.70	18.60 EST.	18.60 EST.	353.40
AUG.		0	93.00	88.97	50.53	31.00	31.31	13.64	13.64	12.71	7.75	6.82	12.40	11.78	11.16	383.71
SEPT.		0	3.60	26.40	36.60	30.00	22.50	16.20	14.70	15.30	11.70	10.50	16.80	12.90	16.50	233.70
OCT.		0	52.39	49.29	36.89	29.14	18.29	18.91	15.50	11.47	10.85	9.30	17.98	15.19	9.30	294.50
NOV.		0	58.50	56.40	35.10	24.90	18.60	11.40	11.70	9.00	5.70	6.00	11.40	12.90	9.90	271.50
DEC.		0	62.0	34.10	29.45	16.74	14.62	16.43	13.02	10.23	8.37	10.23	12.40	11.16	12.09	250.84
TOTAL HOURS		19.96	550.53	563.52	511.62	353.81	282.21	218.12	191.46	148.53	133.37	112.46	191.21	187.17	175.81	3639.78
kW hr available/m² at the Collector		20.33	536.76	521.25	447.67	291.99	218.71	158.14	129.24	92.83	76.88	59.04	86.04	65.51	43.95	2748.06
kW hr available/m² at the receiver Reflectance N_c = .85		17.23	456.24	443.06	380.52	249.19	185.90	134.42	109.85	79.91	55.35	50.18	73.13	55.68	37.36	2336.09
kW hr available/m² at the engine with radiation & convection losses		15.89	412.05	397.83	339.44	219.71	163.25	116.90	94.48	66.98	54.48	41.16	57.79	40.66	23.24	2043.86
Power output kW at the alternator 1.031 kW/m²		15.00	14.10	13.30	12.50	11.70	10.90	10.10	9.30	8.50	7.70	6.90	5.69	4.09	2.49	

Engine Design Point Selection. The information listed in Table 2-2 was plotted in Figure 2-7 which shows annual hour distribution and energy available to the engine versus insolation flux. The engine design point was selected at the peak of energy available to the engine, rather than the hour distribution peak. This is because the quantity to be maximized is the annual energy production. At the peak of the energy curve, the engine output is 14.1/.9 or 15.7 kW (alternator efficiency of .9 assumed). Therefore the engine efficiency must be maximized at 15.7 kW level to obtain the maximum kilowatt-hour over an annual period.

It is also important to note in Figure 2-7 that the 90 percent of the total energy available to the engine occurs at power levels greater than 50 percent of the peak insolution. This implies therefore that the Stirling engine only requires power modulation over the range of 50-100 percent power to utilize 90 percent of energy available.

2.5 Concepts Selected for Parametric Analysis

2.5.1 Preliminary Concepts

Several engine-alternator concepts were defined and evaluated to determine the three best configurations to be carred into the parametric analysis. The concepts that were selected based primarily on overall potential for high efficiency and reliability are:

- 3-5 kW engines with gas bearings, close clearance seals
- 1-15 kW engine with gas
- 1-15 kW - same as the above except incorporation of the ceramic heater head

Following is a discussion of some general concepts that were subsequently rejected and the associated trade-offs of design features.

2.5.1.1 3-Five kW Engine with Wear Seals. A five kilowatt engine intended as one of a triad of engines to be mounted on a single collector and receiver is shown in Figure 2-8. The triad approach was considered since there was a requirement for three-phase electrical output. Linear alternators are not able to generate more than one phase of alternating current, consequently,

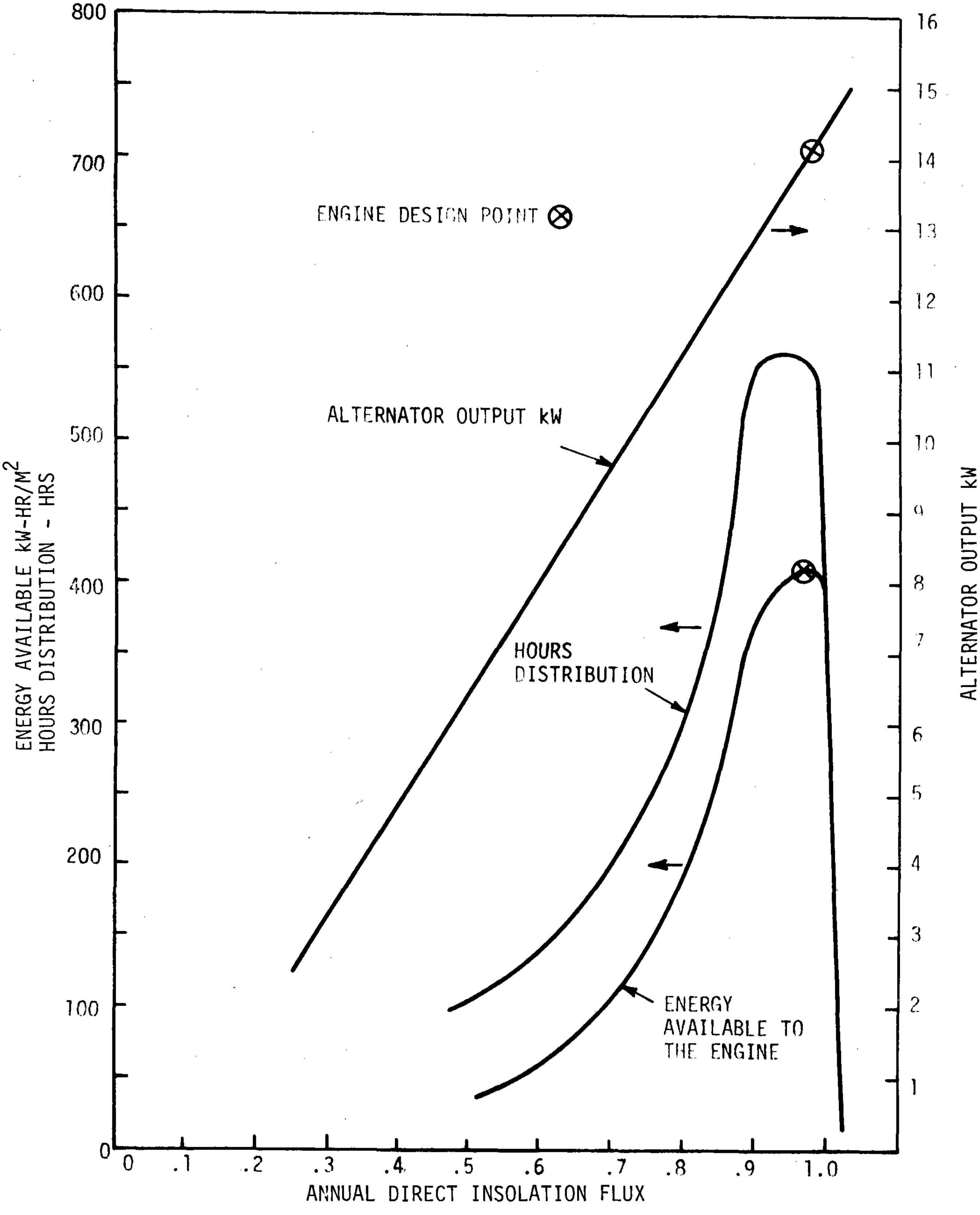

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Fig. 2-7 Energy Available to the Engine versus Insolation Flux

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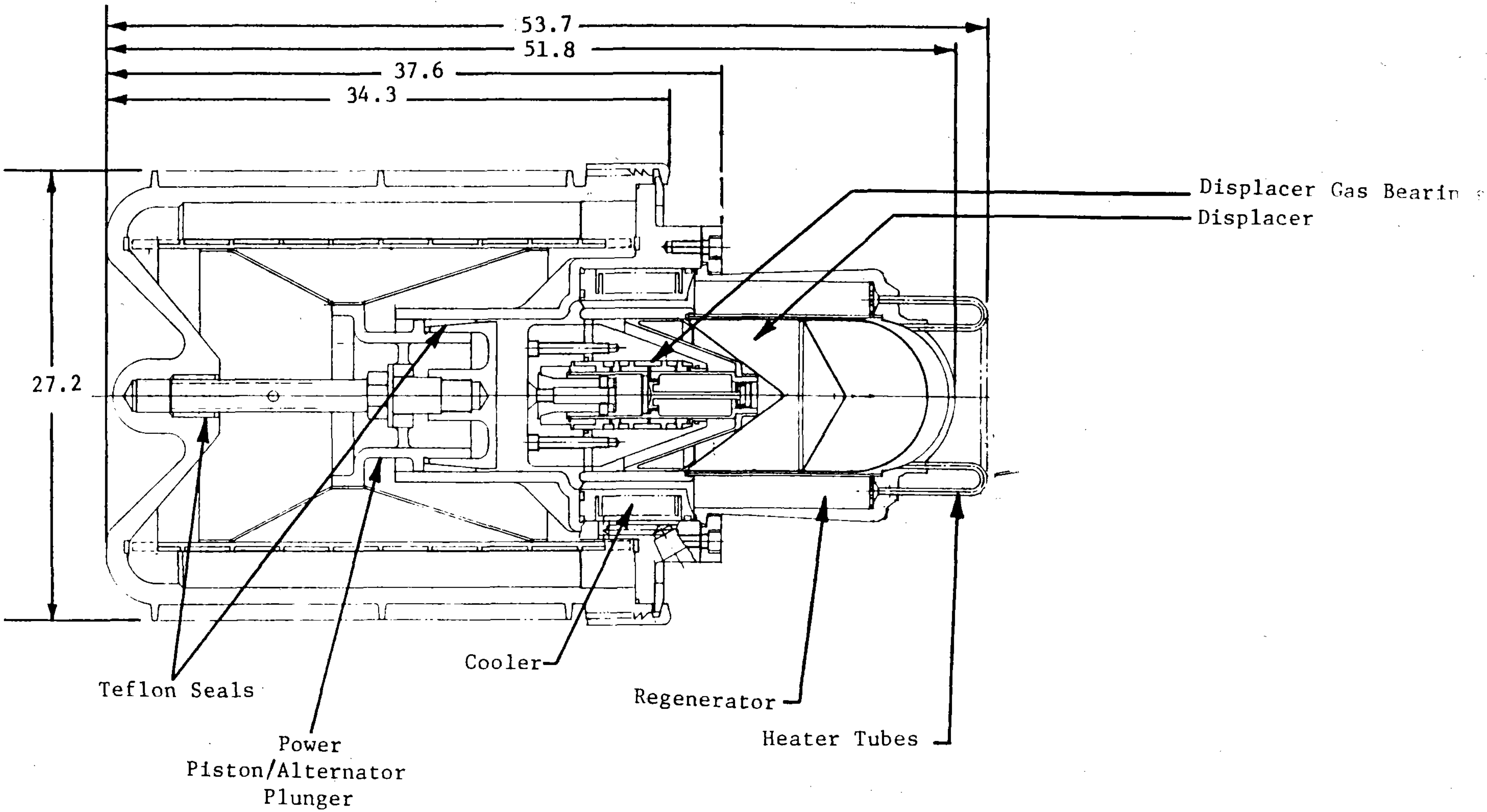

Fig. 2-8 5 kW Engine Utilizing Wear Seals
(dimensions in cm)

three alternators are required for three-phase output. The engine embodied the following design features:

- Rubbing contact filled Teflon bearings and seals on the plunger
- Gas bearings and close clearance seals on the displacer
- Tubular heater head
- Gas springs to tune both displacer and plunger
- Barium ferrite permanent magnet alternator plunger
- External stator alternator

Resulting from these features, the design has certain inherent advantages and disadvantages:

Advantages

- Contact seals and bearings are simple, low cost and adjustable
- Tubular heater head leads to efficient heat transfer and low mass
- Highly reliable gas springs
- Seals and bearings easily adjustable and replaceable
- Alternator of low cost design

Disadvantages

- Performance deterioration due to seal wear
- Starting friction due to rubbing contact seals
- Bearing alignment problems due to locating rear bearing in pressure vessel
- Contamination of gas bearing and regenerator due to wear particles from bearings and seals
- Difficulty in attaining sufficient piston gas spring due to large internal volume of alternator
- Excessive weight due to pressure vessel size

2.5.1.2 One Fifteen kW Single-Cylinder Engine with Dual Gas Springs. A fifteen kilowatt engine has the same general characteristics as the five kW engine except scaled to a higher power. This engine was initially envisioned for use with an external phase converter to supply the specified three-phase output.

No additional advantages would appear to result except perhaps some slightly smaller thermal losses to the surroundings due to the higher volume to surface ratio of the engine.

Additional disadvantages would be a more severe gas spring problem caused by the larger ratio of alternator volume to engine displacement, and higher specific weight. (A larger, thicker pressure vessel is required because power output from the alternator is essentially a surface function while engine output is essentially a volume function causing alternator size to grow at a more rapid rate than the engine.)

2.5.1.3 One Fifteen kW Single-Cylinder Engine with Mechanical Spring. This is an engine with the same general characteristics of the previous engine except it utilizes a gas bearing on the piston and a mechanical spring on the power piston/plunger. The mechanical spring turned out to be quite large and one-third of its mass must be added to the moving mass of the plunger. Also, any asymmetry in the spring or its mounting surfaces could impose side loads on the piston gas bearing that are large enough to exceed a reasonable design capacity. Experience with other high-cycle springs has indicated that the necessary material and manufacturing standards are so high thay they preclude the manufacture at reasonable cost.

2.5.1.4 One Fifteen kW Torsional Engine. A 15-kilowatt engine with two parallel cylinders and a torsional oscillating alternator coupling, with two rocker-like pistons, is shown in Figure 2-9. It was thought that this configuration might simplify the gas bearing problem since there was no gas spring per se. Expansion in one engine resulted in compression in the other engine. Thus there were no gas spring losses and the alternator chamber had a constant volume. Also, the effective moving mass would be lower since in rotary motion only a portion of the mass moves at the full velocity, while mass elements

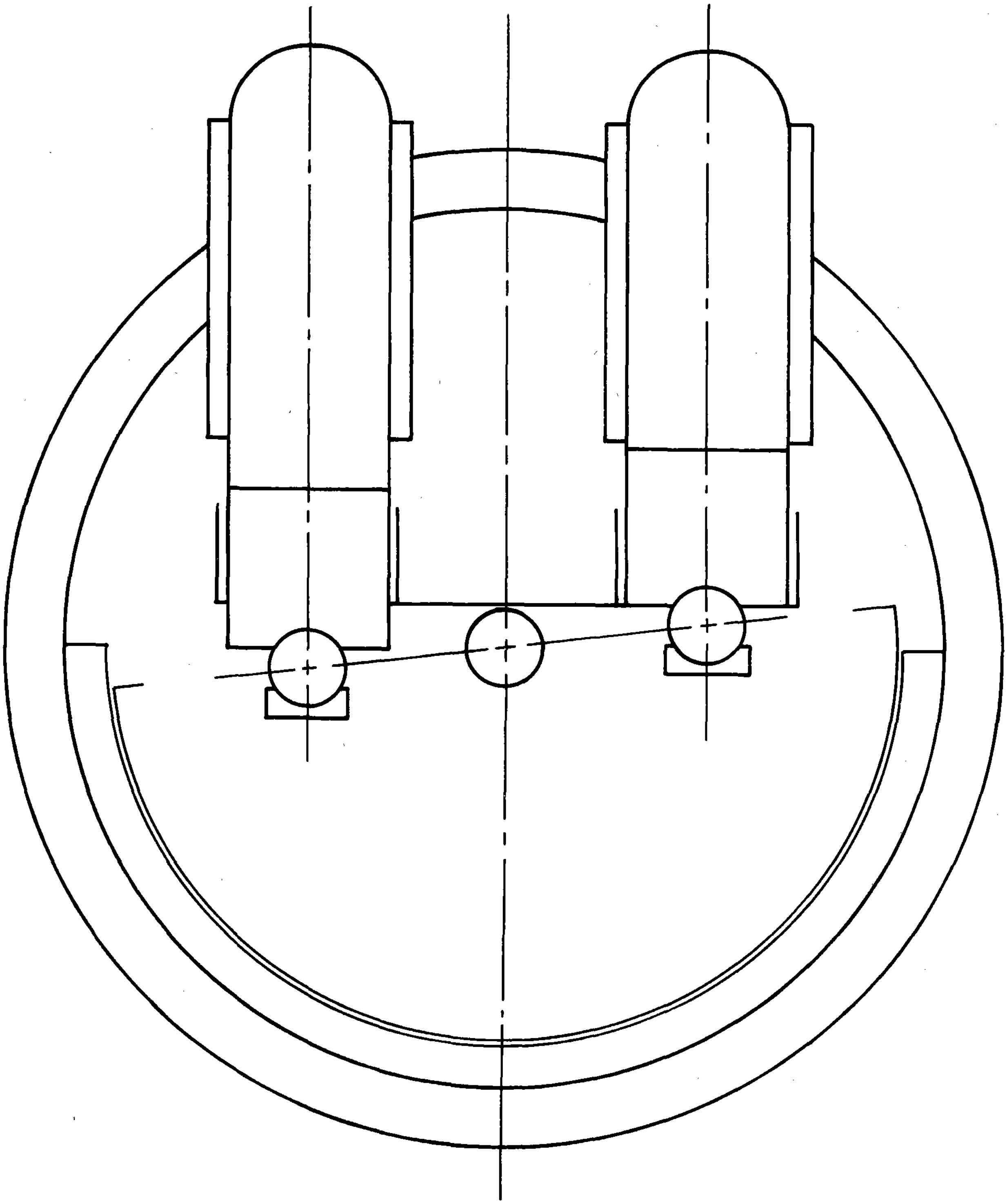

Fig. 2-9 15 kW Torsional Engine

closer to the centerline move slower and possess less kinetic energy. However, the mechanical losses from the large number of required bearings seemed as though they would outweigh any gains made by eliminating the gas spring. Consequently, the concept was rejected.

2.5.2 General Trade-offs

The necessary trade-off/review of the general concepts presented in Subsection 2.5.1 and the configurations finally selected for the parametric analysis are discussed herein.

2.5.2.1 Direct versus Indirect Coupled Alternators. Consideration was given to various means for coupling the free-piston Stirling engine (FPSE) to the electrical generating means. The basic decision was first between direct coupled (i.e., FPSE (Figure 2-10) mechanically attached to an appropriate electrical machine) and indirect coupling. Indirect coupling, e.g., hydraulic or pneumatic, would allow rotating electrical machinery to be powered by the variable stroke free-piston Stirling engine (Figure 2-11).

Supplementary benefit could be derived from the engine concept shown in Figure 2-11 the area of frequency control. Since the generating equipment could be induction type, the frequency would be solely dependent on the bus the unit is attached to and three-phase output is thereby possible.

However, the indirect coupled systems suffer from a double-efficiency penalty since the work done by the engine must first be transformed from a linear to a pressure function, then retransformed to a rotational system. The inherent loss at each transformation indicates that the number of transformations should be held to a minimum. Consequently, it was decided to pursue concepts using direct coupled engine-alternators.

2.5.2.2 Gas Bearing versus Wear Seals. Attempts were also made to incorporate higher efficiency subsystems. Gas bearings were selected over rubbing contact systems for lower parasitic losses, lower wear rates, less internal engine contamination from wear particles and higher inherent reliability. It is also possible to externally pressurize the engine gas bearings before the engine is started to reduce bearing wear and facilitate startup.

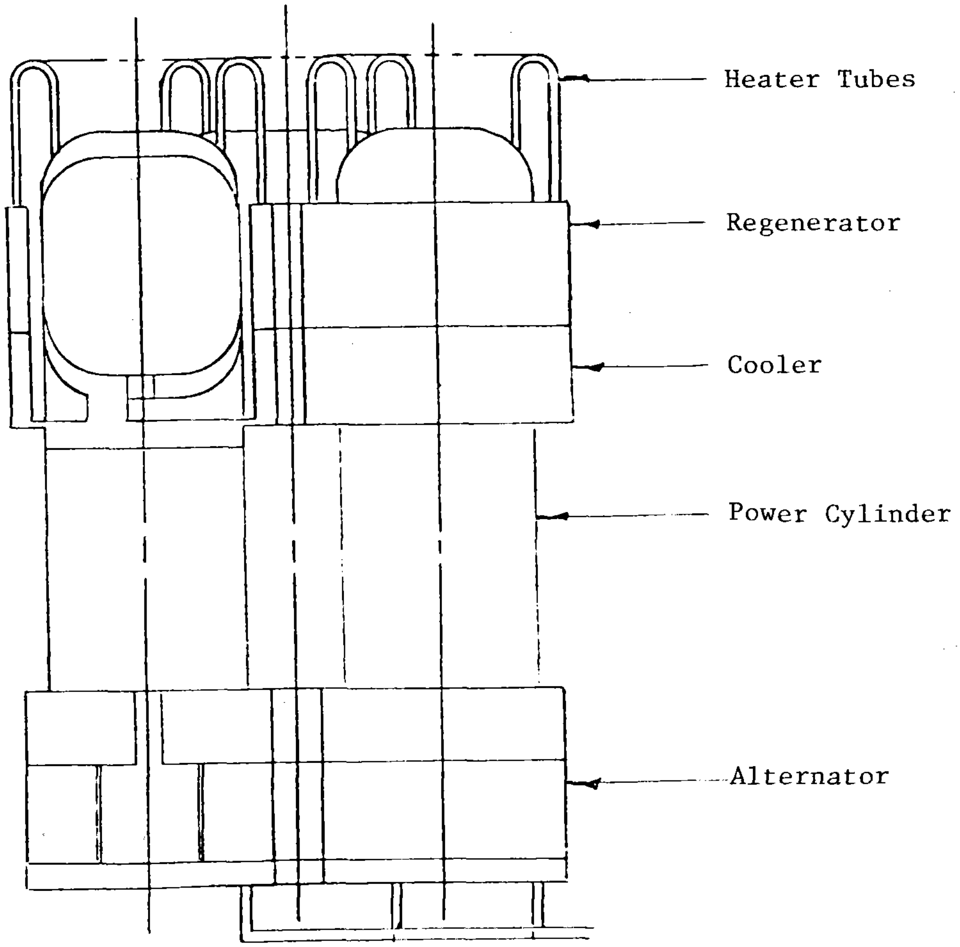

Fig. 2-10 Three Free-piston Displacer Engines Driving Three Single-phase Linear Alternators

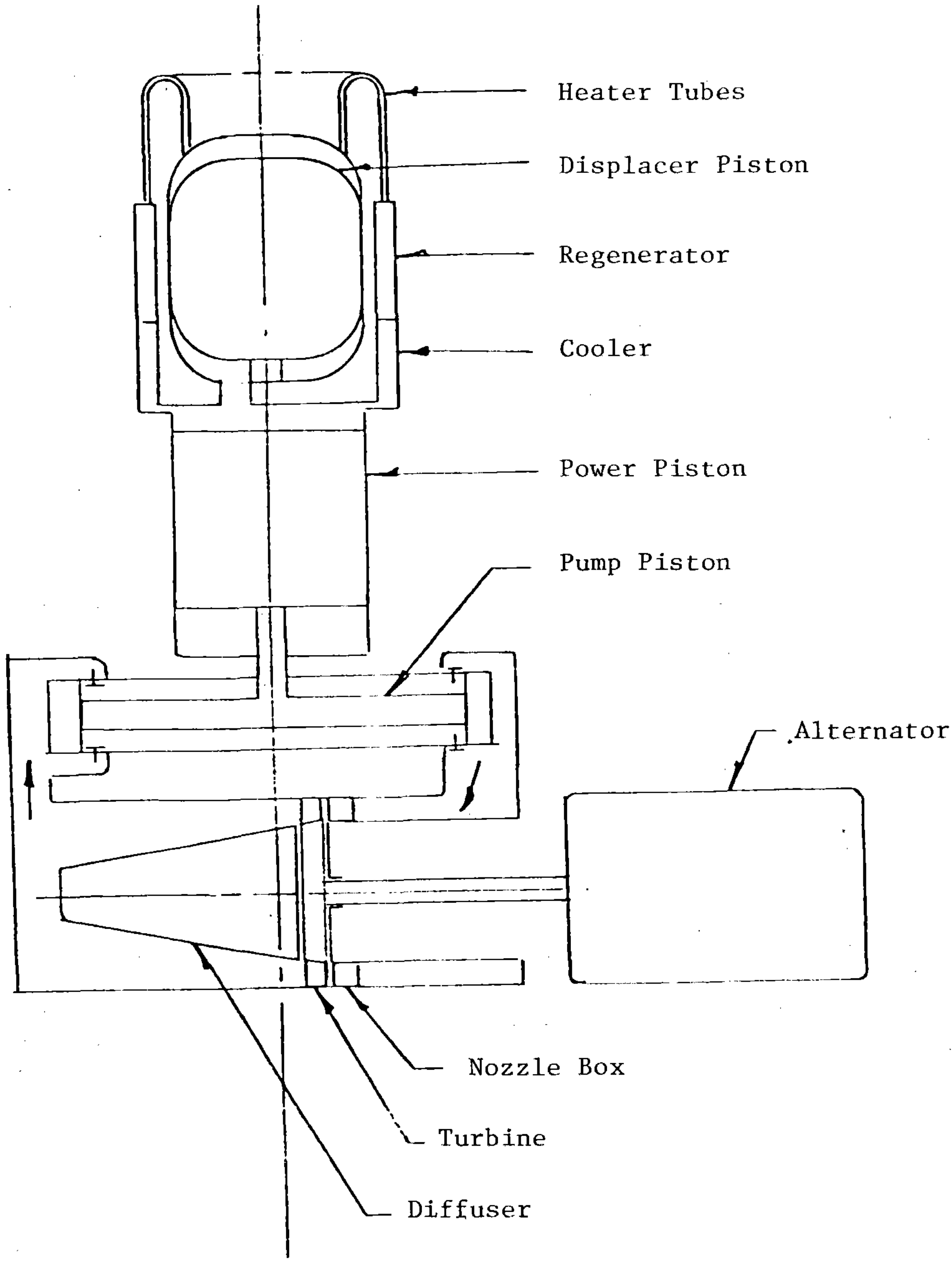

Fig. 2-11 Linear Engine with Fluid Drive

2.5.2.3 Mechanical Spring versus Gas Spring. The design of a mechanical spring was investigated in order to take advantage of its potentially higher efficiency. However, a detailed design effort showed serious problems of high-cycle fatigue would seriously impair engine life. Furthermore, one-third the weight of the mechanical spring contributes to the moving mass and this, in turn, raises the bearing load. Additionally, any nonsymmetry in manufacturing the spring or its mounting surfaces will result in sizable side forces applied to the piston bearing due to the very large axial forces required. Because of the small magnitude of gas spring losses, it was decided to restrict the proposed configurations for parametric analysis to those which incorporated gas springs and close clearance seals. The magnitude of the performance gain utilizing mechanical spring would not justify the sacrifice in reliability.

2.5.2.4 Inside versus Outside Alternator Stator. The alternator configuration was also subject to some scrutiny before a particular approach was settled upon. Detailed discussion of linear alternator operations and trade-off parameters are presented in Section 2.6. The early concepts involved the so-called external stator construction where alternator stator surrounded the plunger and the moving magnets were mounted on the plunger outside surface. The requirement for gas bearings and close clearance seals meant that there would be considerable length consumed by those systems. The pole area required for the permanent magnets also required additional length.

The inside of the plunger was hollow and essentially wasted space. Also, the excessive length of both the plunger and stator led to manufacturing and operating difficulties. Internal diameters with length/diameter ratio of more than three become very difficult to grind in a single operation without special tooling. Furthermore, the plunger is difficult to hold straight and concentric from end-to-end.

The alternator was then redesigned with the thought of reducing the complexity and relaxing production tolerances. It was noticed that if higher flux density magnetic materials such as samarium-cobalt were used for the moving magnets, the pole area could be reduced so it could fit within the bore of the plunger. This led to the creation of the concept of internally mounted moving magnet rings coupled with a cantilevered internal stator.

When this configuration was adopted, other design simplifications followed. The gas bearings and seals could be one continuous diameter and "telescoped" over the alternator thereby allowing the cylinder to be shortened to provide a length to diameter ratio of about three; it could also be manufactured in one piece, eliminating alignment problems. Because of the continuous diameter, the piston/plunger could also be constructed and machined as a single piece. The high flux density magnetic material which allowed the compact construction also has a very high resistance to magnetization, making it virtually impossible to demagnetize the alternator during operation. This is particularily important during the development stages when many unpredictable situations often arise.

Other benefits came from the internal stator too. The piston/plunger weight was reduced to a minimum since the material was "used twice". The steel piston also provides the return flux path for the magnetic circuit, making additional back iron unnecessary. Finally since the alternator is so compact, it is possible to use a much smaller, lighter pressure vessel to contain it, and the internal volume of the vessel is closer to the desired optimum for effective gas spring action.

2.5.2.5 Cast versus Wrought Heater Head. The cylinder head construction was also subject to considerable thought. Because of the decision to use indirect heating of the engine heater head, coupling it to the receiver with a heat pipe and thermal energy storage module, it was felt that the number of joints within the heat pipe envelope should be minimized. This decision was based on the extremely active nature of hot sodium. Weld joints, braze joints and interfaces of dissimilar metals are all subject to possibly serious corrosion. Consequently, an attempt was made to minimize the number of joints, and limit those absolutely necessary to nonstructural areas. In this manner it was considered that the joint designs could be very conservative and the chances of failure would be minimized. For the above reasons a cast heater head would be desirable, since the entire surface inside the heat pipe would be of a single material, the only joint being at the point where the head joins the heat pipe. To guarantee reliability, a redundant joint design could be used at that location.

Ceramic heater heads were considered, but structural ceramics compatible with high temperature sodium are not readily available at this time. Other difficulties result from the difference in thermal expansion between ceramic and metal parts, but many of these can be overcome if joint design is done with care. Additionally, the principal advantage of ceramic, i.e., high temperature strength, is of doubtful value since the increased radiation losses from the receiver may wipe out any gains in engine cycle efficiency. However, a ceramic heater was included in the parametric analysis so that the effect on efficiency could be assessed quantitatively.

2.5.3 Layout/Description of Selected Concepts

Three (3) engines were defined for parametric analysis:

1) 3-5 kW engines with gas bearings, close clearance seals
2) 1-15 kW engine-alternator with gas bearings, close clearance seals
3) 1-15 kW - same as the above, except incorporation of the ceramic heater head

The first engine is a 5 kilowatt design (Figure 2-12) using a cast cylinder head with internal slots to serve as heater tubes. The head is then lined with a "stuffer" which closes the slots and forms passages which travel from a small port in the center of the head radially, then axially back to the top end of the regenerator. The regenerator is annular, packed with layers of 200-mesh screen woven from .001 in. diameter wire. The cooler is defined as a number of round passages parallel to the engine axis, allowing the actual hardware to be either fabricated from tubes or drilled from solid. The displacer and power piston are both carried on gas bearings, using close clearance seals. The displacer and power piston are tuned using gas springs. The large bore/stroke ratio in this engine, along with the volume within the plunger occupied by the alternator stator, allows the entire pressure vessel volume to function as the gas spring. Because of the pressure variation in the whole cavity a plenum must be provided for both the high and low sides of the gas bearing system. The gas bearing supply is ported from the piston gas spring.

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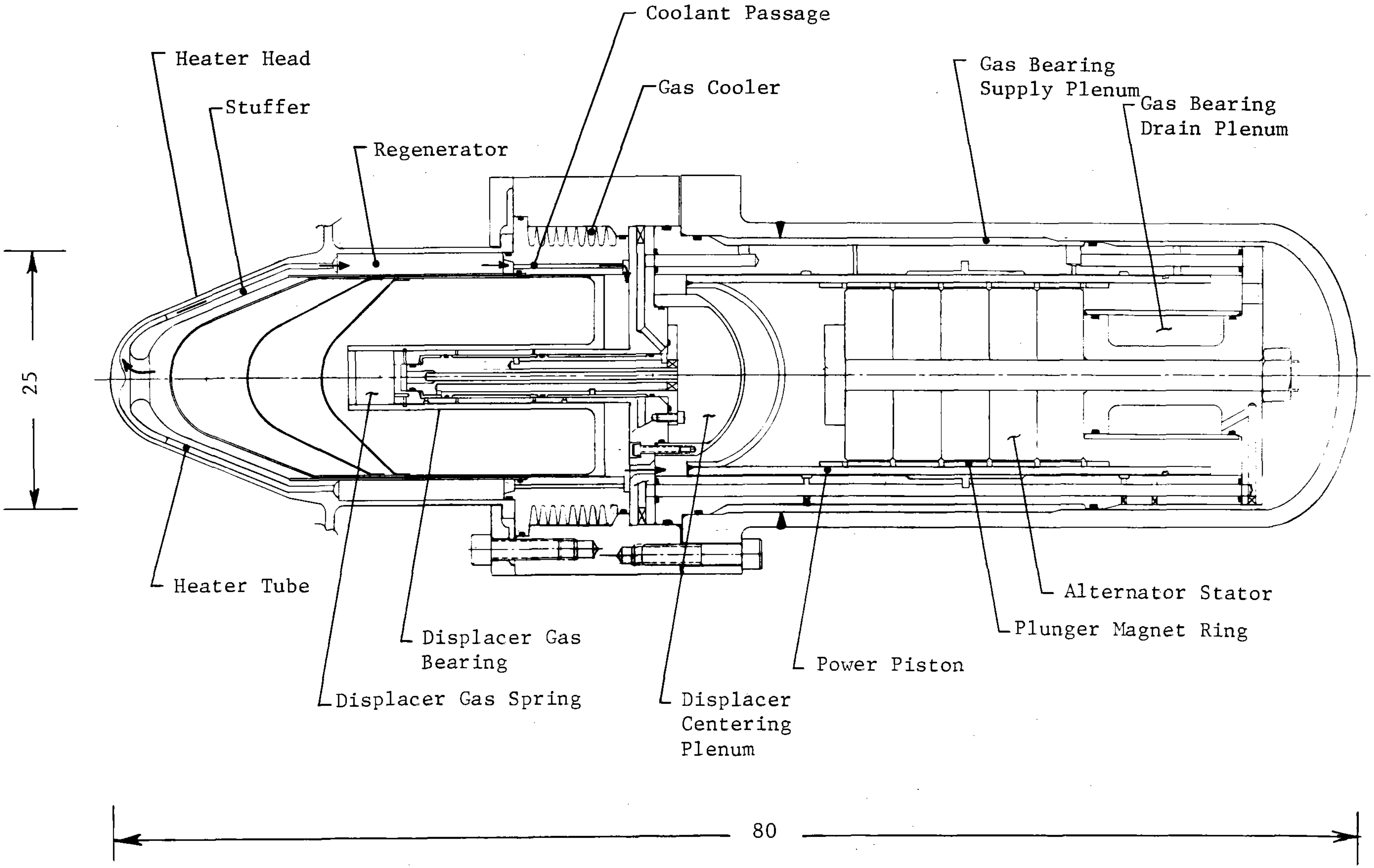

Fig. 2-12 Five kW Engine-Alternator
(Dimensions in cm)

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The alternator is a permanent magnet internal stator design of the type described in Section 2.5.2.4 and is cantilevered from the end of the cylinder.

The second configuration (Figure 2-13) was schematically the same as the 5 kW design, but with the power scaled up to 15 kW. The third design was considered dimensionally the same as the 15 kW engine with the assumption of a ceramic heater head and any other parts required to raise the temperature limit of the engine.

2.6 Linear Alternators

The free-piston Stirling engine produces a reciprocating motion. An energy converter is required which will convert the mechanical energy in this reciprocating motion into electric energy. Such an energy converter, called a linear alternator, typically has one member directly connected to the piston of the free-piston Stirling engine and the other member is stationary. Any electric energy converter obeys the principle of reciprocity, i.e., it converts mechanical energy into electric energy and vice versa. Various forms of linear electric motors can therefore be used as linear alternators.

The principle of operation of these various motors can be divided into two main categories: flux switching and flux reversing. In the flux switching type, the DC excitation coil, which is the source of the main flux, is on the stationary member as are the AC coils. The flux linkage of the AC coils is changed (switched) from high to low level by the movement of the plunger between its two extreme positions. In the flux reversing type, the source of magnetomotive force (MMF) is on the moving member or plunger. The coils are on the fixed member or stator and the flux linking then goes through a positive maximum to a negative maximum as the plunger moves between its extreme positions. In either case, the change in the flux seen by the AC coils causes a voltage to be induced in the coils of the same frequency as the oscillation of the plunger. When used in the alternator mode, this induced voltage delivers current and power to the outside load. The source of the exciting flux (MMF) may be a coil; i.e., electromagnetic or it may be a permanent magnet. Use of a permanent magnet for flux reversal machines avoids sliding or flexible current collectors.

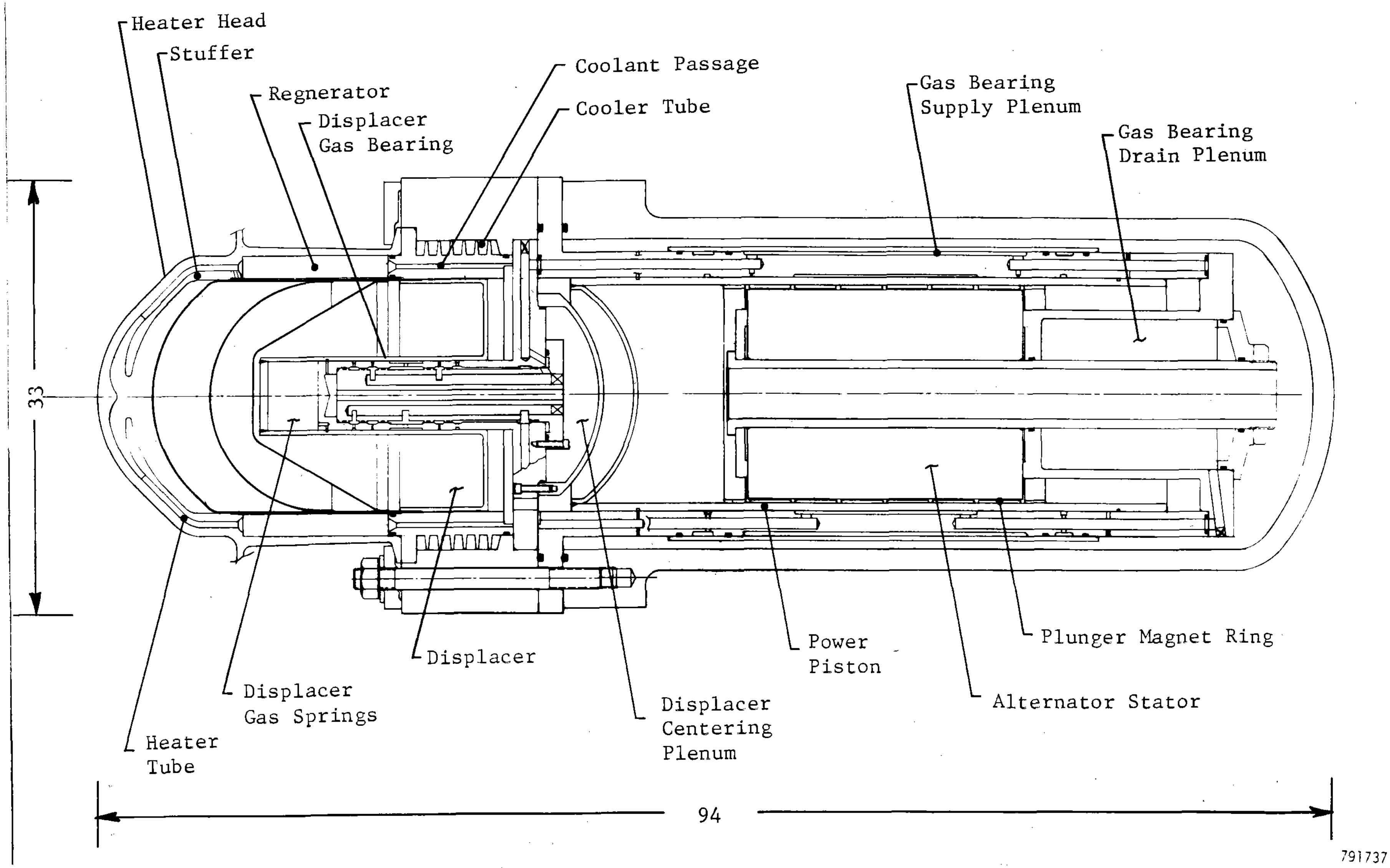

Fig. 2-13 15 kW Preliminary Conceptual Design
(Dimensions in cm)

Variants of these motors can be classified from yet another viewpoint, depending on whether or not the moving member is used as a flux carrier. In either the flux switching or reversing type only a part of the magnetic circuit needs to move relative to the other part. In some configurations this is achieved and results in an extremely light plunger. However, in such configurations the flux has to cross twice as many air gaps as required in the configurations where the flux carrier also travels with the moving part of the magnetic circuit. In the flux switching type, there are two modes of operation, namely: saturated and unsaturated plunger.

The following characteristics are common to all these alternators:

- Oscillating plunger
- Stationary stator
- No moving contacts anywhere
- Single-phase AC power

From a systems viewpoint, the following are desired characteristics:

- Low plunger weight
- Low side pull to reduce requirements in bearing design
- High efficiency
- Low cost
- Ability to scale to high powers, etc.

2.6.1 Selection of Types of Alternators

To make a comparison between various types of alternators, it is desirable to design each concept to a given specification of power, weight of plunger and efficiency, etc. and then evaluate the competing designs. The choice is made on the basis of these qualitative judgements with a reasonable certainty of having selected the best alternator type for the given application. Table 2-3 summarizes the characteristics of these various types. Some of the properties, like the magnitude of the side pull, etc., are inherent in the concept. Certain others, like efficiency and weight of the moving plunger, are design related and dependent on other considerations. For example, one could design to obtain similar efficiency in the various concepts, but the weight of the overall system in these different concepts will be considerably different from each other. The qualitative statements used in Table 2-3 approximately represent the design of any particular concept.

Table 2-3

Comparison of Three Different Concepts of Linear Oscillating Machines

Machine Type	Flux Switching		Flux Reversing
	Unsaturated Plunger	Saturated Plunger	
Flux Variation	Maximum to low (does not change sign)	Maximum to low (does not change sign)	From positive maximum to negative maximum
Total Weight	2	4	1
Plunger Weight	High	Very low (if moving member is not used as a flux carrier)	Medium
Side Load	High	Very low	Medium
Efficiency	Medium	Medium	High
Ability to scale to higher powers	By increasing diameter only	By adding additional sections	By adding additional sections
Control	Four control variables i) α angle ii) stroke iii) D.C. excitation iv) applied voltage	Four control variables i) α angle ii) stroke iii) D.C. excitation iv) applied voltage	Three control variables i) α angle ii) stroke iii) applied voltage

2.6.2 Reasons for Selecting Flux Reversing Type Permanent Magnet Alternator

The permanent-magnet reversing flux-type alternator was selected, based on the previous considerations, for further study. It offers the following advantages over the other types considered:

- Light-plunger weight
- Potential for high efficiency
- Excellent geometrical match with the engine. (The diameter of the alternator plunger and the diameter of the piston of the engine can be the same which results in considerable simplifications.)
- Overall light weight of the alternator
- Low side pull making gas bearing design easier.

The lack of one of the variables for control (viz., the DC excitation) is really the only penalty associated with this choice.

2.6.3 Two Configurations of Permanent Magnet Alternator

The permanent-magnet reversing flux-type alternator can be implemented in many configurations. A configuration with the plunger in the form of a hollow cylinder lends itself to interfacing with the engine as shown in Figure 2-14. Here the hollow space inside the plunger can be used for engine piston, gas spring, etc.

From the heat transfer consideration, the outside stator arrangement of Figure 2-14 is desirable since almost all of the heat is generated in the stator. Magnetically, the design of the stator is straightforward. The highest flux densities occur at the airgap (i.e., at the plunger diameter). This allows the use of flat laminations.

An 'inside stator' configuration was also considered since it offers the following advantages (refer to Figure 2-15):

1. Smaller overall diameter for the same plunger diameter (and, hence power)

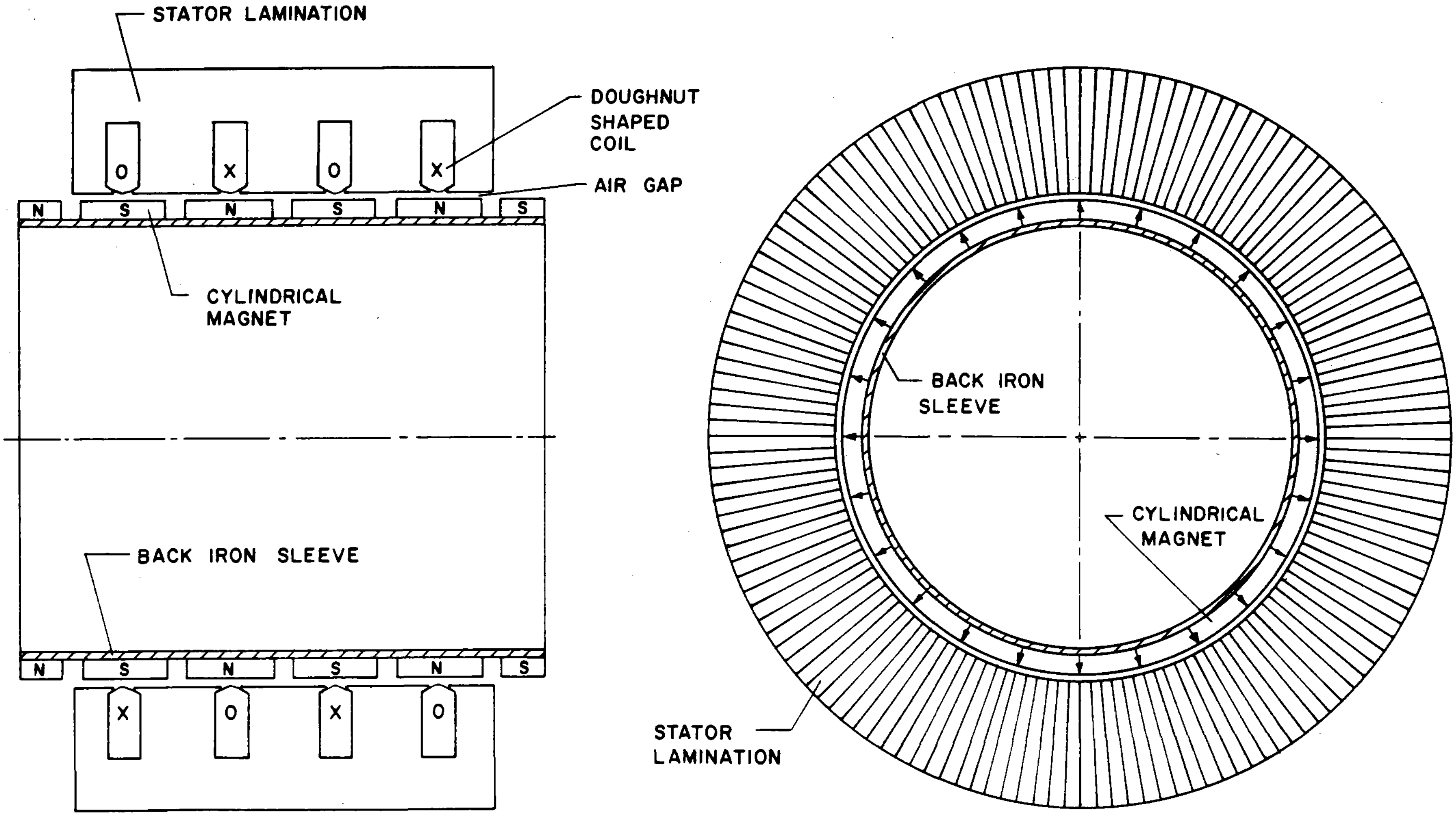

Fig. 2-14 External Stator Configuration

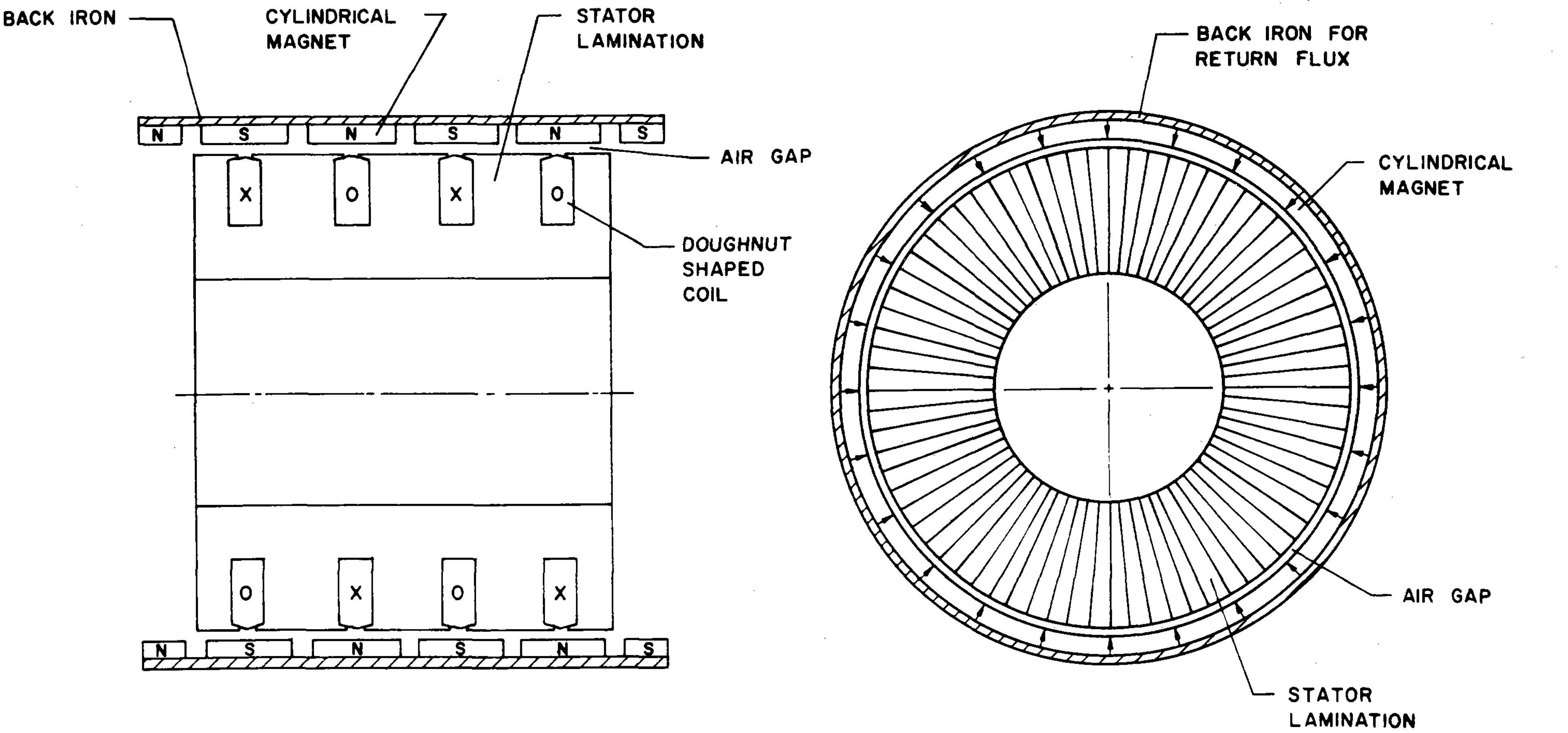

Fig. 2-15 Inside Stator Configuration

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2. Smaller overall weight

3. Smaller steel weight implying less core losses

4. Smaller diameter resulting in the smaller mean diameter of the copper wires producing smaller resistance. (The resistance could be as much as 30 percent lower than the outside stator configuration for otherwise identical alternators.)

5. Considerations (3) and (4) imply the potential for higher efficiency.

The difficulties associated with this design are:

1. Heat transfer area is small and, hence, alternator cooling imposes more difficult requirements on the cooling system.

2. The stator has to be cantilevered from one end. This would require higher clearance between the plunger and the stator. Furthermore, this limits the size of the alternator for a given plunger diameter.

2.6.4 Parameters of Design

The airgap power in a permanent magnet alternator is given as follows:

$$P = (4.8 \times 10^3)\ (\pi\ D''\ \ell''\ lm'')B_r^2 \times \left(\frac{V}{E}\right) \left\{1 - \frac{1}{(V/E)^2}\right\}^{1/2} \left\{\frac{lm''}{lm''+g''}\right\} \frac{f_1}{(1-S_o/\ell)\ C_f}$$

where

P	= Airgap power (watts)
D"	= Diameter (in inches)
l"	= Half length of each section (inches)
l"	= Amplitude of maximum stroke
lm"	= Thickness of the magnet (inches)
g"	= Radial length of the airgap (inches)
Br	= Residual flux density of the magnet (Tesla)
(V/E)	= Ratio of applied voltage to induced voltage
E	= R.M.S. value of the fundamental component of the induced voltage with slot opening considered
E_o	= R.M.S. value of the induced voltage with no slot opening

then

$f_1 = E/E_o$

S_o/ℓ = Ratio of slot opening to the length of each section

if

X = Airgap reactive of the machine with slot opening and fringing flux considered

X_o = Airgap reactive of the machine with no slot opening and assuming the AC coil current flux to have only the radial component

then

$$C_f = \frac{X}{X_o}$$

The above equation takes into consideration the stability requirements discussed in detail in Subsection 4.7.1. Figure 2-16 presents the permanent magnet alternator nomenclature.

2.6.4.1 Selection of the Magnet Material. The magnet properties of importance are shown below:

- Br, residual flux density
- Hc, coercive force
- μ, recoil permeability
- ρ, resitivity
- d, density.

The previous equation shows that for a given magnet volume (2π D" 1" lm") the power output increases as Br^2. However, for stability reasons, V/E has to be greater than unity, and this can also be seen from the reference to the above power equation. Thus, if the plunger is not moving and the alternator is connected to the power grid, the current in the AC coil will cause a reverse flux in the magnet. It is important that the magnet should not demagnetize under such conditions. This implies that 'Hc' the coercive force of the magnet, should be high. Alternatively, the rated voltage should be limited to such a value that the flux due to current in the AC coils will not demagnetize the magnet. Thus, higher values of coercive force will result in higher power density.

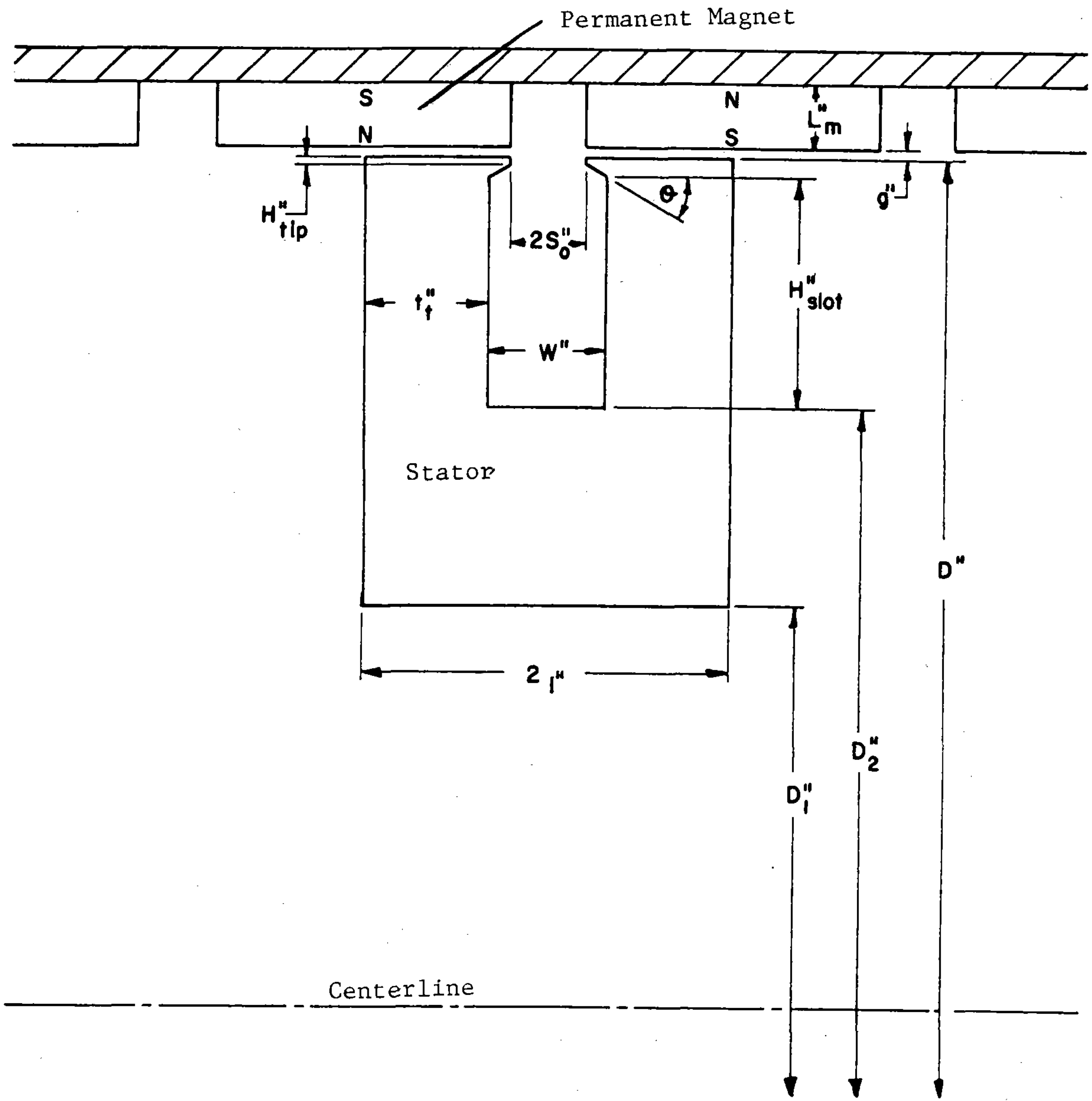

Fig. 2-16 Permanent Magnet Alternator Nomenclature

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As the plunger oscillates back and forth, the flux density at its surface varies due to the stator currents and also the permeance variations caused by the effects of slot openings. This flux variation will cause surface losses due to the eddy currents in the surface of the magnet. To keep surface losses to a minimum it is then desirable to have as high a resistivity as possible. The density of the magnet material should be as low as possible to obtain the lightest possible plunger weight.

Other physical properties such as temperature stability, coefficient of thermal expansion, etc. also have to be considered during the design phase.

Properties of some commercially available magnets are listed in Table 2-4.

Based on these properties a "Samarium-Cobalt" ($SmCo_5$) material was selected for the designs to be parametrically analyzed.

2.7 Parametric Analysis

2.7.1 Method

The receiver system characteristic has been discussed in Subsection 2.3. The engine power level at which the maximum thermal efficiency should be optimized has also been determined in Subsection 2.4. With a number of physical constraints, such as material property limitation and alternator requirements, a parametric analysis for the optimization of the engine under different operating conditions was performed. The objective of this analysis is to determine the operating condition at which the highest possible thermal efficiency can be achieved. The analysis consists of ten operating conditions, five heater-head temperatures (1200, 1400, 1600, 1800, and 2000°F) and two coolant water temperatures (70 and 150°F). After the operating condition for the highest thermal efficiency has been selected, the same engine is checked for partial load and performances.

2.7.1.1 Optimization of Engine Configuration. The processes involved in a real Stirling cycle are complex. The interaction between these processes makes the problem even more complicated. Simulation of a real Stirling cycle by computer technology offers in-depth knowledge of the cycle and

Table 2-4

Properties of Some of the Commercially Available Permanent Magnet Materials

Material	Residual Induction Br (Gausses)	Coercive Force Hc (oersteds)	Recoil Permeability μ	Resitivity P (micro ohm Cm)	Density (pounds per cubic inch)
Cast ALNICO Grade 5-7	13000	730	3.8	47	0.264
Grade 8B	7800	1850	1.9	50	0.264
Ferrites Grade 8	3850	3150	1.07	$>10^6$	0.175
Grade 7	3450	4000	1.06	$>10^6$	0.166
Rare earth Cobalt type Sm CO_5	8600	>25000	1.05	50	.307
Mishmetal Cobalt	7000	7000	1.05	50	.307

can be used to determine quantitatively the effect on the engine performance of each design parameter. Computer simulation utilizes a set of differential equations which express the principle of energy conservation, mass and momentum based on theories of thermodynamics, dynamics of compressible flow and heat transfer. These relationships are too complex for a general analytical solution, and hence, they are solved numerically by computer.

By specifying the necessary limitations and operating conditions and the three engine-alternator configurations selected for the parametric analysis, the computer seeks the combined optimum configuration so that the highest thermal efficiency is achieved. The computerized optimization was performed utilizing the free-piston Stirling engine optimization computer program developed by Sunpower Inc. This highly sophisticated computer program varies all the engine parameters, except those given limitations, until an optimum configuration is obtained as well as any detailed information in any segment of the cycle, if desired.

2.7.2 Constraints and Input to the Analysis

For this analysis, the input and constraints are as follows;

- Material Selection and Engineering Properties

  a. Inconel 713 L.C.

     A practical Stirling cycle has the same characteristics as a Carnot cycle, the higher the working gas temperature in the hot space or the lower the temperature in the cold space the higher the cycle efficiency. The temperature of the coolant is usually limited by the environmental conditions. To obtain the highest engine efficiency, it is desirable to operate the engine at the highest possible working gas temperature. A Stirling engine usually operates at high pressure and the heat transfer surface is limited to avoid unnecessary dead volume. In addition, to withstand the high temperature and pressure, it is also desirable to have a material with high thermal conductivity. Therefore, the material selection for Stirling design is a major consideration. To this end, metallurgists at MTI were consulted. After a careful study, Inconel 713 L.C. was selected not only for its high-creep stress but also for its good castability and the absence of cobalt, a strategic material.

Therefore, this material's properties were used for the selected engine configurations with the exception, of course, of the ceramic heater head design.

The engineering properties of Inconel 713 L.C. used as input to the computer program are as follows:

Temperature (°F)	1% Creep Stress for 50,000 hrs (psi)	Thermal Conductivity Btu/hr ft^2/in.
1200	45,000	166
1300	36,600	173
1400	23,800	179
1500	15,400	199
1600	11,200	218
1800	3,900	
2000	1,500	
Specific density	.286 lb/in.^{3}	
Specific heat	.14 Btu/lb °F	

b. Silicon Carbide

Ceramic materials were also investigated as possible candidates for heater head material. They differ considerably from most metals in their mechanical properties. Ceramics are considerably more brittle than most metals and consequently failure in ceramics occurs less predictably and more often catastrophically. Silicon carbide was chosen to be used in the engine optimization program for the selected concept at 1800 and 2000°F only. The ceramic heater head engine concept was included in the parametric analysis to indicate the performance potential of such an advanced engine design. However, before ceramic material could be used as a Stirling engine component, a considerable amount of fabrication, technology, development and experimental verification of reliability would have to be performed. The properties used in the program were as follows:

Tensile Strength	15,000 psi
Coefficient of Expansion	2.7×10^6
Specific Heat	.30 Btu/lb °F
Density	.10 $lb/in.^3$
Thermal Conductivity	15 Btu/hr ft °F

- General Engine constraints:

Frequency	60 Hz
Engine Output	5.5 kW (3 units for 3 phases) or 15 kW single unit
Piston Diameter	5" (determined by alternator design with stator coil inside piston)
Piston Amplitude	0.6" for kW engine 0.69" for 15kW engine (figures determined by alternator design)
Piston Weight	14.3 and 24 lb, respectively (weights determined by alternator design)
Displacer Diameter	5" (selected for design simplicity)
Displacer Weight	3.54 lb (determined by geometry)
Hot Space Metal Temperature	1200, 1400, 1600, 1800, and 2000°F
Coolant Temperature	70 and 150°F
Number of Cooler Tubes	100 (cooler tube length, diameter and number of tubes can be optimized by computer program. Optimized figures are theoretical not practical for manufacturing. Number of tubes was made as an input and computer seeks tube diameter and length for maximum thermal efficiency)

Number of Heater Tubes	46 same reason as cooler
Working Gas	Hydrogen for highest efficiency
Regenerator Porosity	.92 based on MTI experience

2.7.3 Analysis Results

The results for the 5 kW and 15 kW engine configuration and performance under different temperature combinations are listed in Table 2-5 through Table 2-7. The intended optimization of both engines at hot space material temperature of 2000°F was not performed because the thermal efficiency dropped at 1800°F. This is partially due to the drop of the metal creep strength that requires a heavier wall, and increased heat conduction loss from the hot to the cold end.

The highest overall thermal engine efficiency of 45.1 percent was obtained for the 5 kW engine operating at a 871°C heater head temperature and a 21°C cooler temperature. The engine thermal efficiency for the three engine configurations at the various heater head and cooler temperatures are presented in Figure 2-17.

The "best" engine configuration (i.e., highest efficiency) is run to check engine performance at off-design and partial-load requirements. The partial load results for the 5 kW engine are presented numerically in Table 2-8 and pictorially in Figure 2-18. It is noted that the efficiency of the engine does not diminish rapidly as a function of off-load. As shown in Subsection 2.4 the available energy below a partial load of 50 percent is less than 10 percent therefore a solar engine does not require partial load operation over the entire range of powers.

The result of the parametric analyses for the 15 kW engine at 1800°F heater head temperature indicated the optimum piston frontal area is 206.7 cm^2, equivalent to a 6.4 in. diameter piston instead of the given 5 in. diameter. Since it is desirable from an alternator packaging concern to have an inside out stator, the optimization program was again run for a 7 in. diameter piston (sufficient diameter to allow an inside out stator) to determine the design parameters (presented in Table 2-9) for the conceptual design shown in Figure 2-13 for the single 15 kW engine.

TABLE 2-5

5.5 KW ENGINE AT DIFFERENT OPERATING TEMPERATURES

HEATER TEMPERATURE		5 KW ENGINE 1200°F		1400°F	
COOLER TEMPERATURE		70°F	150°F	70°F	150°F
Total P-V Power	(watt)				
Piston NET POWER	(watt)	5470	5460	5390	5450
Frequency	(hz)	60	60	60	60
Charge Pressure	(bar)	70.3	86.1	47.88	55.9
Heat Input	(watt)	13675	15124	12682	13491
Piston Amplitude	(cm)	1.52	1.52	1.52	1.52
Displacer Amplitude	(cm)	.93	1.09	1.09	1.09
Displacer Phase Angle	(°)	36.1	30.4	36.09	36.1
Displacer Cyl. Diameter	(cm)	12.7	12.7	12.7	12.7
Displacer Rod Diameter	(cm)	2.8	3.52	3.45	3.44
Piston Frontal Area	(cm^2)	126.7	126.7	126.7	126.7
Regenerator Porosity		.92	.92	.92	.92
Regenerator Length	(cm)	12.9	12.9	11.2	11.2
Regenerator Void Volume	(cm^3)	967	967	728	728
Cooler Wall Temperature	(°C)	21	65	21	65
Cooler Tube Diameter	(mm)	2.95	2.62	2.29	2.95
Cooler Tube Length	(cm)	6.98	6.98	6.98	6.98
Cooler Tubes	(#)	100	100	100	100
Average Cooler ΔT	(°C)	10.5	11.5	10.8	13.3
Heater Wall Temperature	(°C)	649	649	760	760
Heater Tube Diameter	(mm)	3.45	3.45	3.43	3.45
Heater Tube Length	(cm)	11.2	11.2	11.2	13.0
Heater Tubes	(#)	46	46	46	46
Average Heater ΔT	(°C)	27.9	23.2	32.3	26.9
Efficiency	(%)	40	36.1	42.5	40.4

TABLE 2-5 (Cont'd)

5.5 KW ENGINE AT DIFFERENT OPERATING TEMPERATURES

		5 KW ENGINE			
HEATER TEMPERATURE		1600°F		1800°F	
COOLER TEMPERATURE		70°F	150°F	70°F	150°F
Total P-V Power	(watt)				
Piston NET POWER	(watt)	5430	5420	5390	5400
Frequency	(hz)	60	60	60	60
Charge Pressure	(bar)	49.5	48.9	34.2	41.8
Heat Input	(watt)	12040	13219	13051	13568
Piston Amplitude	(cm)	1.52	1.52	1.52	1.52
Displacer Amplitude	(cm)	1.09	1.09	1.09	1.25
Displacer Phase Angle	(°)	36.1	36.1	47.5	41.8
Displacer Cyl. Diameter	(cm)	12.7	12.7	12.7	12.7
Displacer Rod Diameter	(cm)	3.78	3.42	5.08	4.53
Piston Frontal Area	(cm^2)	126.7	126.7	126.7	126.7
Regenerator Porosity		.92	.92	.92	.92
Regenerator Length	(cm)	14.6	11.2	14.6	14.6
Regenerator Void Volume	(cm^3)	949	728	529	949
Cooler Wall Temperature	(°C)	21	65.5	21	65.5
Cooler Tube Diameter	(mm)	2.95	2.29	2.62	2.95
Cooler Tube Length	(cm)	6.98	6.98	6.98	6.98
Cooler Tubes	(#)	100	100	100	100
Average Cooler ΔT	(°C)	10.1	11.6	13.4	11.9
Heater Wall Temperature	(°C)	871	871	982	982
Heater Tube Diameter	(mm)	3.40	3.45	3.45	3.45
Heater Tube Length	(cm)	11.2	11.2	11.2	11.2
Heater Tubes	(#)	46	46	46	46
Average Heater ΔT	(°C)	30.6	34.5	45.1	34.0
Efficiency	(%)	45.1	41.0	41.3	39.8

TABLE 2-6

15 KW ENGINE AT DIFFERENT OPERATING TEMPERATURES

HEATER TEMPERATURE / COOLER TEMPERATURE		15 KW ENGINE 1200°F 70°F	1200°F 150°F	1400°F 70°F	1400°F 150°F
Total P-V Power	(watt)				
Piston NET POWER	(watt)	15090	15200	15500	15300
Frequency	(hz)	60	60	60	60
Charge Pressure	(bar)	168	188	143	163.1
Heat Input	(watt)	40240	44706	37621	40157
Piston Amplitude	(cm)	1.746	1.746	1.746	1.746
Displacer Amplitude	(cm)	1.25	1.24	1.25	1.25
Displacer Phase Angle	(°)	36.1	36.1	36.1	36.1
Displacer Cyl. Diameter	(cm)	12.7	12.7	12.7	12.7
Displacer Rod Diameter	(cm)	3.14	2.94	3.11	3.18
Piston Frontal Area	(cm^2)	126.7	126.7	126.7	126.7
Regenerator Porosity		.92	.92	.92	.92
Regenerator Length	(cm)	12.02	12.7	12.7	12.7
Regenerator Void Volume	(cm^3)	1022	825	952	952
Cooler Wall Temperature	(°C)	21	65.5	21	65.5
Cooler Tube Diameter	(mm)	2.50	3.16	2.83	2.83
Cooler Tube Length	(cm)	6.98	6.98	6.98	6.98
Cooler Tubes	(#)	100	100	100	100
Average Cooler ΔT	(°C)	14.4	20	15.5	16.8
Heater Wall Temperature	(°C)	649	649	760	760
Heater Tube Diameter	(mm)	3.5	4.1	3.57	3.50
Heater Tube Length	(cm)	10	10	10	10
Heater Tubes	(#)	46	46	46	46
Average Heater ΔT	(°C)	36.5	43.3	41.8	39.7
Efficiency	(%)	37.5	34	41.2	38.1

TABLE 2-6 (Cont'd)

15 KW ENGINE AT DIFFERENT OPERATING TEMPERATURES

HEATER TEMPERATURE		15 KW ENGINE 1600°F		1800°F	
COOLER TEMPERATURE		70°F	150°F	70°F	150°F
Total P-V Power	(watt)				
Piston NET POWER	(watt)	15300	15400	15500	15600
Frequency	(hz)	60	60	60	60
Charge Pressure	(bar)	132	151	51.6	59
Heat Input	(watt)	34931	38889	36730	40000
Piston Amplitude	(cm)	1.746	1.746	1.746	1.746
Displacer Amplitude	(cm)	1.25	1.25	1.25	1.25
Displacer Phase Angle	(°)	36.1	41.8	54.6	51.1
Displacer Cyl. Diameter	(cm)	12.7	12.7	16.2	16.22
Displacer Rod Diameter	(cm)	3.19	3.28	6.13	5.73
Piston Frontal Area	(cm^2)	126.7	126.7	206.7*	206.7
Regenerator Porosity		.92	.92	.92	.92
Regenerator Length	(cm)	14.4	16.1	16.1	16.1
Regenerator Void Volume	(cm^3)	1080	1368	1207	1207
Cooler Wall Temperature	(°C)	21	65.5	21	65.5
Cooler Tube Diameter	(mm)	3.16	2.83	3.49	3.49
Cooler Tube Length	(cm)	6.98	6.98	6.98	6.98
Cooler Tubes	(#)	100	100	100	100
Average Cooler ΔT	(°C)	14.5	13.9	18.9	21.1
Heater Wall Temperature	(°C)	871	871	982	982
Heater Tube Diameter	(mm)	3.5	3.5	4.1	4.7
Heater Tube Length	(cm)	10	10	8	8
Heater Tubes	(#)	46	46	46	46
Average Heater ΔT	(°C)	41.9	41.1	82	89.4
Efficiency	(%)	43.8	39.6	42.2	39

TABLE 2-7

15 KW CERAMIC ENGINE 7" PISTON DIAMETER ENGINE

HEATER TEMPERATURE		15 KW CERAMIC ENGINE			
COOLER TEMPERATURE		Cooler Temp. 70°F			
		1800°F	2000°F		
Total P-V Power	(watt)				
Piston NET POWER	(watt)	15200	15600		
Frequency	(hz)	60	60		
Charge Pressure	(bar)	117	71		
Heat Input	(watt)	33778	33766		
Piston Amplitude	(cm)	1.746	1.746		
Displacer Amplitude	(cm)	1.25	1.25		
Displacer Phase Angle	(°)	36.1	41.8		
Displacer Cyl. Diameter	(cm)	12.7	14.57		
Displacer Rod Diameter	(cm)	2.92	4.26		
Piston Frontal Area	(cm^2)	126.7	166.7		
Regenerator Porosity		.92	.92		
Regenerator Length	(cm)	12.7	16.1		
Regenerator Void Volume	(cm^3)	1097	1207		
Cooler Wall Temperature	(°C)	21	21		
Cooler Tube Diameter	(mm)	2.83	3.49		
Cooler Tube Length	(cm)	6.98	6.98		
Cooler Tubes	(#)	100	100		
Average Cooler ΔT	(°C)	13.5	15.3		
Heater Wall Temperature	(°C)	982	1093		
Heater Tube Diameter	(mm)	3.5	4.1		
Heater Tube Length	(cm)	10	8		
Heater Tubes	(#)	46	46		
Average Heater ΔT	(°C)	47.4	79.4		
Efficiency	(%)	45	46.2		

TABLE 2-8

PARTIAL LOAD PERFORMANCE OF THE 5.5 KW ENGINE

		5 KW ENGINE			
HEATER TEMPERATURE		1600°F Heater		70°F Cooler	
COOLER TEMPERATURE		Design Pt.	80% Stroke	60% Stroke	
Total P-V Power	(watt)				
Piston NET POWER	(watt)	5430	3840	2420	
Frequency	(hz)	60	60	60	
Charge Pressure	(bar)	49.5	49	48.8	
Heat Input	(watt)	12040	8658	5721	
Piston Amplitude	(cm)	1.52	1.22	.91	
Displacer Amplitude	(cm)	1.09	.93	.73	
Displacer Phase Angle	(°)	36.1	35.1	35.5	
Displacer Cyl. Diameter	(cm)	12.7	12.7	12.7	
Displacer Rod Diameter	(cm)	3.78	3.78	3.78	
Piston Frontal Area	(cm^2)	126.7	126.7	126.7	
Regenerator Porosity		.92	.92	.92	
Regenerator Length	(cm)	14.6	14.6	14.6	
Regenerator Void Volume	(cm^3)	949	949	949	
Cooler Wall Temperature	(°C)	21	21	21	
Cooler Tube Diameter	(mm)	2.95	2.95	2.95	
Cooler Tube Length	(cm)	6.98	6.98	6.98	
Cooler Tubes	(#)	100	100	100	
Average Cooler ΔT	(°C)	10.1	8.22	6.22	
Heater Wall Temperature	(°C)	871	871	871	
Heater Tube Diameter	(mm)	3.4	3.4	3.4	
Heater Tube Length	(cm)	11.2	11.2	11.2	
Heater Tubes	(#)	46	46	46	
Average Heater ΔT	(°C)	30.6	25.1	19.5	
Efficiency	(%)	45.1	44.3	42.3	

TABLE 2-9

15 KW CERAMIC ENGINE 7" PISTON DIAMETER ENGINE

		15 KW 7" PISTON			
		Inconel 713			
HEATER TEMPERATURE		Cooler 70°F			
COOLER TEMPERATURE		1600°F			
Total P-V Power	(watt)				
Piston NET POWER	(watt)	15500			
Frequency	(hz)	60			
Charge Pressure	(bar)	45.6			
Heat Input	(watt)	34831			
Piston Amplitude	(cm)	1.746			
Displacer Amplitude	(cm)	1.25			
Displacer Phase Angle	(°)	41.8			
Displacer Cyl. Diameter	(cm)	17.78			
Displacer Rod Diameter	(cm)	5.28			
Piston Frontal Area	(cm^2)	248.3			
Regenerator Porosity		.92			
Regenerator Length	(cm)	11			
Regenerator Void Volume	(cm^3)	1045			
Cooler Wall Temperature	(°C)	21			
Cooler Tube Diameter	(mm)	3.49			
Cooler Tube Length	(cm)	6.98			
Cooler Tubes	(#)	100			
Average Cooler ΔT	(°C)	20.5			
Heater Wall Temperature	(°C)	871			
Heater Tube Diameter	(mm)	4.7			
Heater Tube Length	(cm)	8			
Heater Tubes	(#)	46			
Average Heater ΔT	(°C)	84.5			
Efficiency	(%)	44.5			

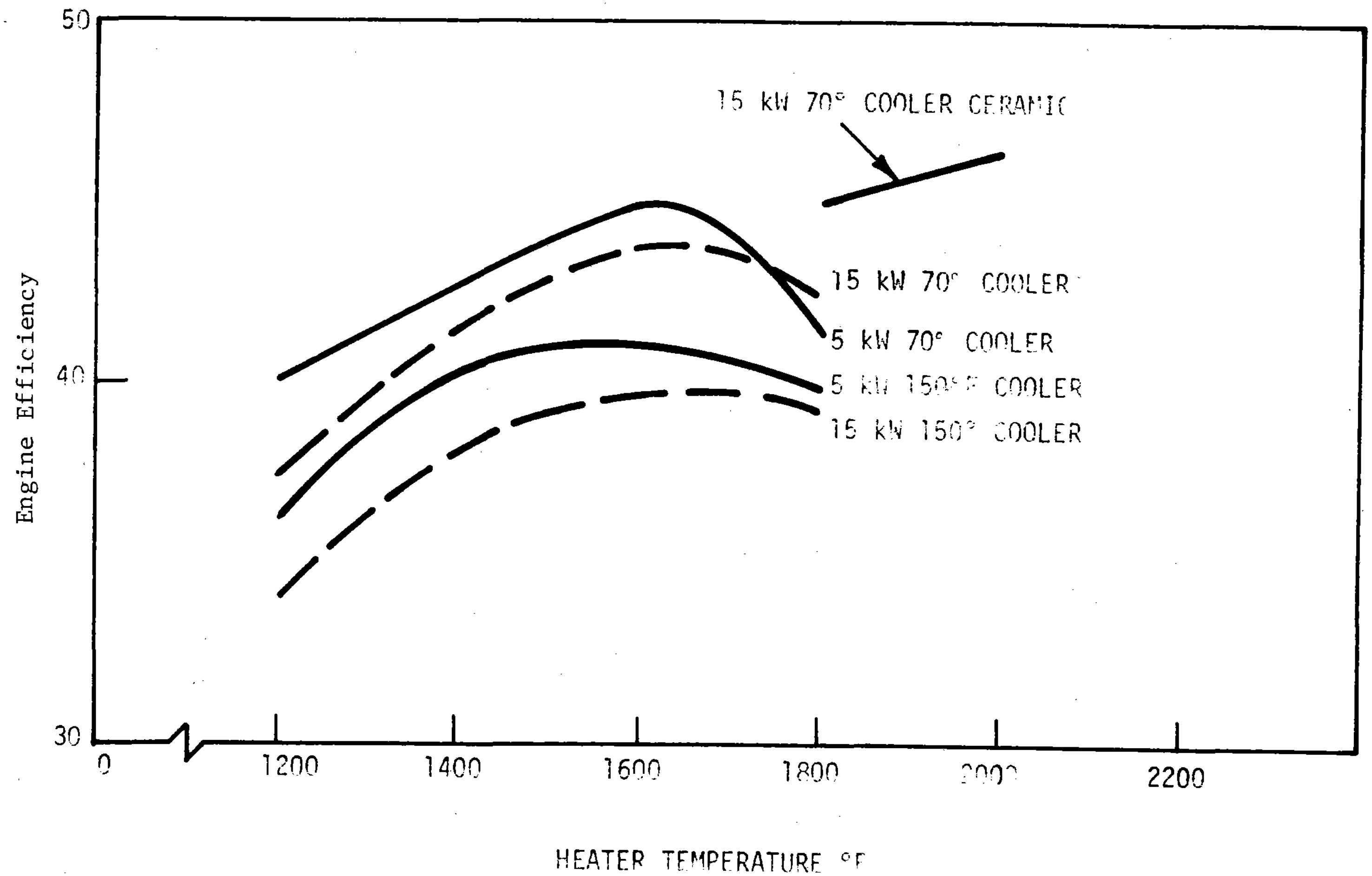

Fig. 2-17 Engine Overall Thermal Efficiency
Versus Heater Temperature

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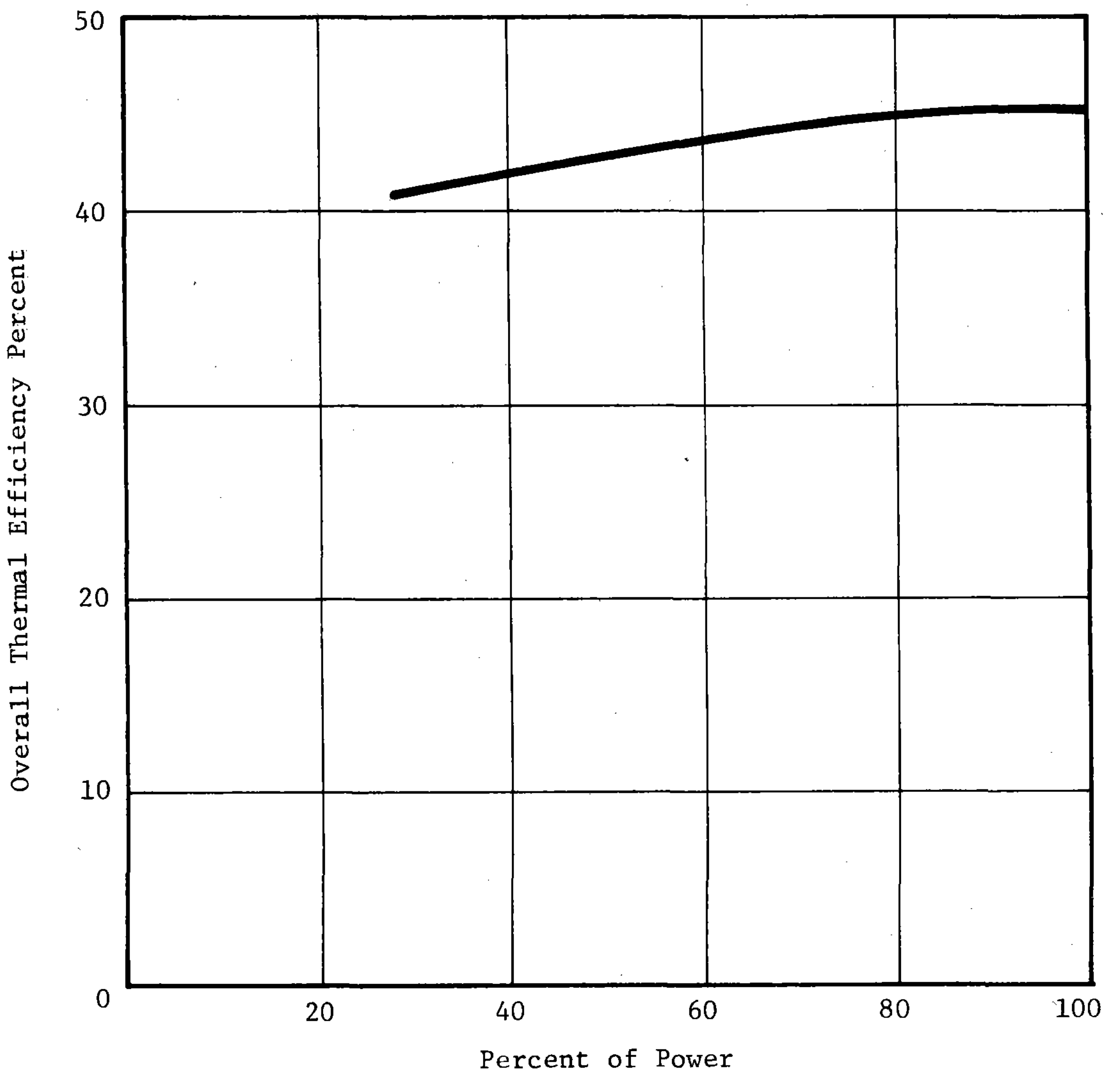

Fig. 2-18 5 kW Engine Partial Power Overall Thermal Efficiency

2.8 Ranking

2.8.1 Method

Ranking of attractive configurations was performed in two stages. Stage 1 is a preliminary screening prior to the parametric analysis, used to identify the three most attractive engine concepts and Stage 2 is a screening after parametric design and identification of attractive total system concepts. The ranking method used will be a decision analysis technique developed by Kepner-Tregoe.

Basically, this method identifies all desirable objectives as either musts or wants. In order to be a viable alternative, a configuration/concept must meet the must objectives. A preliminary screening of all configuration/concepts prior to input to the parametric analysis was performed to insure all engine concepts analyzed met the must conditions. These must conditions were:

- Capable of 15 kWe power output
- Capable of engine/alternator efficiency of 40 percent
- 3 phase - 60 Hertz output

At the completion of the parametric analysis, each engine configuration was ranked based on the Kepner-Tregoe method. Since all configurations/concepts passed the preliminary screening, then all met the must conditions. All other objectives are classified as wants and are given relative weighting factors from 1 to 10, with 10 being the most desirable objective. Each configuration then is reviewed with respect to the ability to comply with objectives relative to all other configurations; the best configuration for each objective is given a 10 and all other configurations a relatively low number.

After each configuration was rated relative to each other for each desirable objective, the relative rating is multiplied by the weighting factor for each objective and the total was then summed with the totals for each objective. This results in a numerical rating of all attractive Stirling engine/alternator system configurations with the highest number representing the most attractive configuration/concept.

If two or more configurations were numerically close, it was necessary to make a final choice based on an assessment of adverse consequences of each configuration and a review of how each configuration satisfies the most weighted objectives.

2.8.2 Criteria

The established requirements for the Task 1 study are:

power output = 15 kW - 3 phase
system efficiency $\geq$ 40 percent

Based on these requirements some initial engine/alternator configurations (i.e., a free-piston Stirling being used to drive a turbine) were not considered further. The wants (i.e., objectives) that were defined for this evaluation and their relative importance (weighting factor) are as follows:

Efficiency	10
Reliability	9
Control	8
Durability	6
Simplicity (cost)	5
Risk	4
Performance Potential	4
Power Potential	4
Installation	2
Weight	2

A preliminary screening based on the previous ranking system was made to determine the three best engine configurations to be parametrically analyzed. The concepts that were chosen for the parametric analyses were:

- 1 - 15 kW single acting Stirling engine/alternator
- 3 - 5 kW single acting Stirling engine/alternator
- 1 - 15 kW single acting Stirling engine/atlernator with a ceramic heater head

The results of the parametric analyses is presented in Subsection 2.7. Using the parametric analyses and engineering judgement, each of the 3 engine configurations at different operating temperatures were ranked according to overall system considerations. Hence, while a single 15 kW ceramic heater head engine has a higher engine efficiency than any other engine (see Figure 2-19), the overall system efficiency is slightly lower than the 3 - 5 kW engine/system because a 15 kW system would require a static converter to meet the required three-phase specification. Obviously if three phase was not a requirement, the 15 kW ceramic heater head configuration would obtain a 10 for efficiency and other configurations would be relatively lower.

Similarly for each want, the system (i.e., collector, selected receiver, engine/alternator/utility grid) is ranked relative to all other systems. The results of this ranking procedure are shown in Table 2-10.

2.8.3 Recommendation

Based on the original program guideline of achieving 15 kW-3 phase electrical output from a single disk, the 3-5 kW engine/alternator/system operating at 871°C was selected. The single 5 kW engine conceptual layout is shown in Figure 2-19. However prior to initiation of the conceptual design phase of this study results of the parametric analysis and configuration definition were reviewed with NASA/Lewis. It was determined at that time that the three phase output requirement did not need to be obtained with a single dish-engine system but could be obtained with a 3-15 kW dish-engine system. Because of this change of definition the concept selected for the conceptual design and implementation assessment was a single 15 kW engine-alternator system.

TABLE 2-10

RANKING OF ATTRACTIVE ENGINE/ALTERNATOR SYSTEMS

Musts: 15 KWE Output
30 - 60 HZ Output
η Eng/Alt $\geq$ 35%

Systems: Wants-Weight		15KW Ceramic 70°	1800°F	15KW Ceramic 70°	2000°	5 KW 70°	1400°	5 KW 70°	1600°	5 KW 70°	1800°	5 KW 150°	1400°	5 KW 150°	1600°
Efficiency	10	6	60	8	80	7	70	10	100	6	60	5	50	6	60
Reliability	9	9	81	9	81	7	63	7	63	7	63	7	63	7	63
Control	8	10	80	10	80	9	72	9	72	9	72	9	72	9	72
Durability/Main	6											No Difference			
Simplicity	5	8	40	8	40	8	40	8	40	8	40	8	40	8	40
Risk	4	4	16	4	16	10	40	10	40	10	40	10	40	10	40
Perf Potential	4	9	36	9	36	10	40	10	40	10	40	10	40	10	40
Power Potential	4											No Difference			
Installation	2	10	20	10	20	10	20	10	20	10	20	10	20	10	20
Weight	2	10	20	10	20	8	16	8	16	8	16	8	16	8	16
Total Score			353		373		361		391		351		341		351
Rank			7		3		5		1		8		12		9

Systems: Wants-Weight		5 KW 150°	1800°	15 KW 70°	1400°	15 KW 70°	1600°	15 KW 70°	1800°	15 KW 150°	1400°	15 KW 150°	1600°	15 KW 150°	1800
Efficiency	10	5	50	3	30	5	50	4	40	1	10	2	20	2	20
Reliability	9	7	63	10	90	10	90	10	90	10	90	10	90	10	90
Control	8	9	72	10	80	10	80	10	80	10	80	10	80	10	80
Durability/Main	6														
Simplicity	5	8	40	10	50	10	50	10	50	10	50	10	50	10	50
Risk	4	10	40	9	36	9	36	9	36	9	36	9	36	9	36
Perf Potential	4	10	40	9	36	9	36	9	36	9	36	9	36	9	36
Power Potential	4														
Installation	2	10	20	10	20	10	20	10	20	10	20	10	20	10	20
Weight	2	8	16	9	18	9	18	9	18	9	18	9	18	9	18
Total Score			341		360		380		370		340		350		350
Rank			13		6		2		4		14		10		11

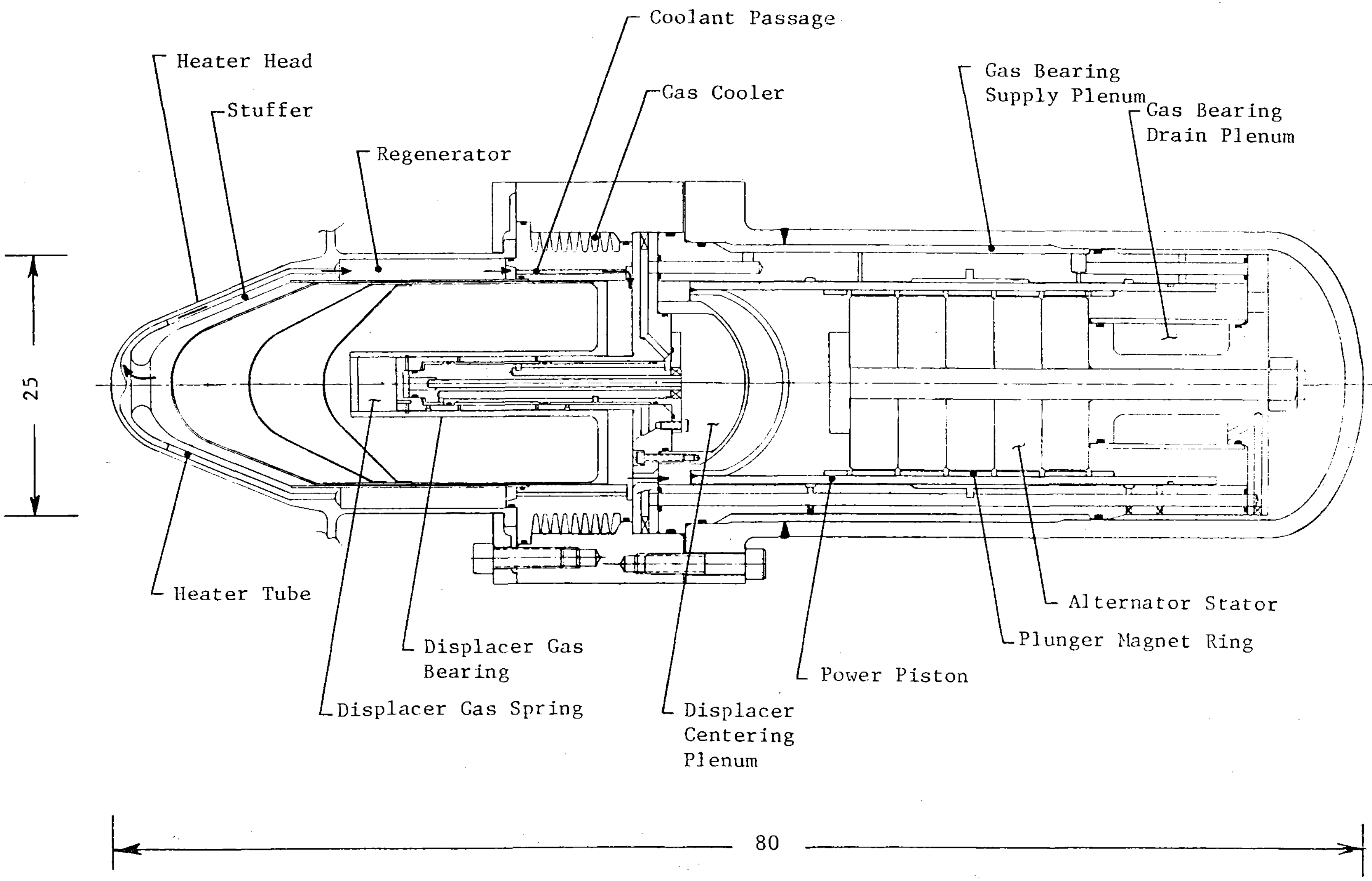

Fig. 2-19 Five kW Engine-Alternator
(Dimensions in cm)

3.0 IMPLEMENTATION ASSESSMENT

The purpose of the work completed under this task was to determine the potential for development of the 15 kW Solar Engine Concept Design to a production engine status and for eventual widespread implementation in solar thermal electric generating systems. The main objectives that were addressed are assessments of:

- Technology Status
- Producibility
- Durability
- Growth Potential

A continual cross-feed of information between this implementation assessment and the near term conceptual design activity was established in order to produce a positive impact upon the near term design.

3.1 Review of Technology Status

The objectives of the work completed under this subtask were to review the 15 kW Solar Engine concept design and:

1) Identify aspects of the design which are critical to its success.
2) Identify technologies which are key to the success or failure of a part.
3) Assess whether the design represents current technology, adaptations of current technology, or significant improvements.
4) Assess the system sensitivity to changes in component technology.

3.1.1 Free-Piston Stirling Engines/Linear Alternators

The basis for components technology for developing an efficient free-piston Stirling engine power conversion system is well established. Although the thermodynamic analysis used to predict engine performance is complex, the actual operation of the engine/alternator system is known for its simplicity. Nevertheless, the best overall efficiency is achieved when all of the critical components are well matched.

The potential for increasing the system efficiency beyond current performance levels has been verified with laboratory engine and alternator components.

Inherent design characteristics of gas-bearing supported free-piston Stirling engines show promise for developing a highly reliable system with a long-life potential. Although only a limited amount of operating time has been accumulated with the laboratory engines, inspection of the hardware confirms the potential for developing a power conversion system with long life and high reliability.

3.1.2 Component Technology Status

The 15 kW preliminary conceptual engine design developed as part of Task 1, shown in Figure 3-1 and outlined by part number in Table 3-1, was reviewed in terms of those operating and design parameters which could significantly impact engine life, cost, and performance. Parameters such as temperatures, stresses, clearances, complexity, materials, etc. were evaluated for each engine component. As the result of this evaluation, those components which are considered critical to the engine operation were identified. The key technologies upon which the success or failure of each component part relies were also evaluated. The following definitions were adopted for the purpose of categorizing the status of each of the identified technologies.

State-Of-The-Art. This term refers to the current level of sophistication of a developing technology. A significant amount of experience has been accumulated with the technology as configured. The design and operational characteristics of this engine are well known.

Adaptation of Current Technology. The technological concept has been proven in similar or analogous hardware systems. Adaptation to the solar Stirling design will require no basic research in materials or manufacturing.

Significant Improvement in Technology. The concept is new or has only been verified in a laboratory. Development is required in order to identify a proven design approach for adaptation to a Stirling engine system.

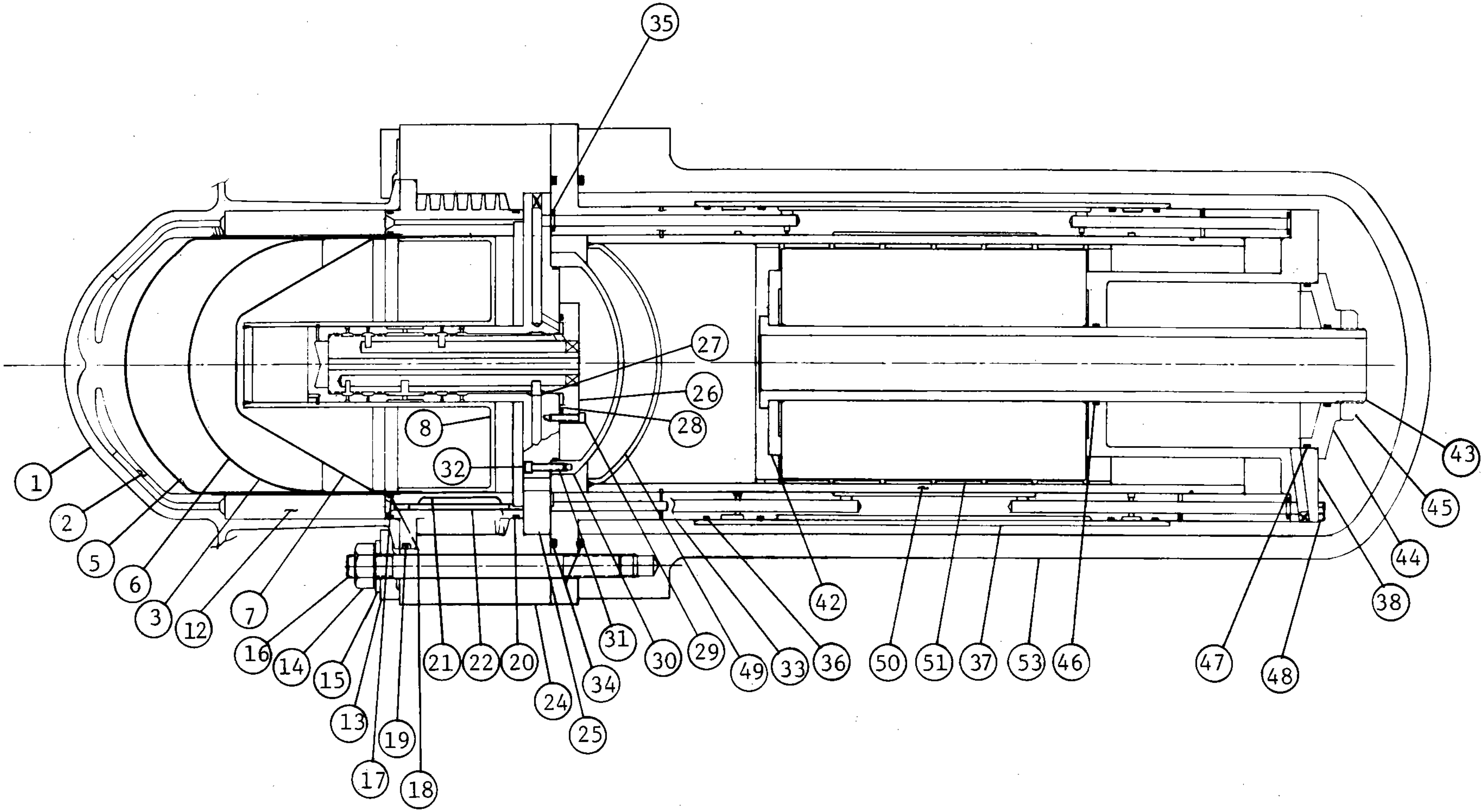

Fig. 3-1 15 kW Preliminary Conceptual Design

TABLE 3-1

15 kW CONCEPTUAL DESIGN PARTS LIST

Item	Name	Qty.	Remarks	Weight (lb)
1	Heater Head, Outer	1	Inco 713	18.8
2	Heater Head, Inner	1	321 SS	4.42
3	Regenerator Wall, Inner	1	321 SS Tube .035 Wall	1.10
4	Heater Head Assembly	1	Braze and Finish Machine 1-3	
5	Displacer Shell	1	321	1.83
6	Shield #1	1	321	.74
7	Shield #2	1	321	.72
8	Displacer Base	1	321	4.69
9	Spring Cavity Plug	1	32	.075
10	Bleed Orifice	1		
11	Displacer Assembly	1	Chrome Oxide Bore Weld, Finish Machine Items 5-10	
12	Regenerator Matrix		#5 Screen	1.57
13	Clamping Ring (3 Pc Split)		316	7.0
14	Clamping Nuts	12	$\bar{6}$/8-18 Hexnut	
15	Washers	12	STD	
16	Assembly Studs	12	$\bar{6}$/8 x 7.5 Steel	7.8 lb
17	"O" Ring, Fwd. Cooler Outer	2	See 20	
18	"O" Ring, Fwd. Cooler Inner	1		
19	"O" Ring, Fwd. Cooler Water	3	Same as 34	
20	"O" Ring, Aft. Cooler Water	-	Same as 17	
21	Cooler Housing	1	316	10.84
22	Cooler Tubes	100		.52
23	Cooler Assembly		Braze and Finish Machine Items 21 and 22	
24	Mounting Housing	1	Aluminum	22.3
25	Displacer Spider and Post	1		17.1
26	Displacer Post-Inner	1		4.39
27	"O" Ring - Displacer Post	6		
28	"O" Ring - Displacer Post Flange	1		

TABLE 3-1

15 kW CONCEPTUAL DESIGN PARTS LIST (cont'd)

Item	Name	Qty.	Remarks	Weight (lb)
29	Mounting Screws - Inner Post	3	#10-24 x .75 Sock HD	
30	Displacer Spring Plenum	1		2.78
31	"0" Ring - Displacer Plenum	1		
32	Mounting Screws - Displacer Plenum	8	#10-24 x .88 Sock HD	
33	Engine Cylinder	1		105
34	"0" Ring - Cylinder Flange	-	Same as 19 1 each side of flange	
35	"0" Ring - Port Seal	5		
36	"0" Ring - Cylinder OD	4		
37	Cover Sleeve - Cylinder	1		12.4
38	Alternator Stator Flange	1		17.75
39	Stator Laminations			.82
40	Stator Coils			8.74
41	Stator Assembly	1	Assembly of 39 and 40	
42	Spring Washer	1		1.45
43	Tie Bolt	1		6.54
44	Plenum Cap	1		2.18
45	Locknut	1		.56
46	"0" Ring - Tiebolt Seal	2		
47	"0" Ring - Plenum Cap Seal	1		
48	Mounting Bolt - Stator Flange	6	3/8-24 x 1.25 Sock HD	
49	Piston Head	1		1.74
50	Piston Body	1		26.6
51	Magnet Ring	8		
52	Piston Assembly	1	Items 49, 50, 51 Assemble Chrome Oxide	
53	Pressure Vessel	1		157.8

Table 3-2 Summarizes the critical components and the corresponding assessment of the technology status.

3.1.3 Technology Sensitivity Assessment

Variations in the component technologies were assessed in terms of their impact on overall system characteristics. The sensitivity of cost, reliability, performance, and maintenance requirements were evaluated for changee in the technology status for each critical component. The following outline highlights that assessment.

I. HEATER HEAD

A. Sensitive Parameters: Cost, life, reliability, efficiency

B. Independent Parameters: Design approach
- brazed-tube design
- cast-head design

C. Considerations: The heater head/regenerator housing assembly is one of the most crucial components in the entire engine/alternator system. Because of high engine operating temperatures, high-creep strength materials must be considered. Wall thickness must be kept to a minimum in order to limit thermal conduction losses, and an adequate heat transfer surface area must be maintained in order to effect the required energy input to the machine. An adequate heat transfer surface with a minimum void volume can readily be achieved via a brazed-tube heater design. On the other hand, single-cast head designs offer the potential for significantly reducing the machining and assembly time required for the more conventional heater-tube type designs. In addition, the cast design should insure a higher level of reliability and life. However, more complex shapes result and equally high manufacturing costs may be incurred.

As the power level of free-piston Stirling engines increases, the critical nature of the heater becomes more apparent, and optimization of the heater design positively impacts the overall cost, performance, and reliability to a greater extent. The payoffs for successful innovation in Stirling engine heater development are quite attractive:

TABLE 3-2

STATE-OF-THE-ART EVALUATION

Critical Component	Key Technology	Technology Status
1) Heater Head	• Condensing Liquid Metal Heat Transfer	• Significant Improvement
	• Cast Heater Head	• Significant Improvement For High Heat Transfer
2) Regenerator	• High-Volume Processing With Effective Material Utilization	• Adaptation of State-of-the-Art
3) Bearing System	• Internally Supplied Bearing Gas	• Adaptation of Current Technology
	• Surface Coatings Techniques	• State-of-the-Art
4) Seals	• Close Tolerance Seal	• Extension of State-of the-Art for Life
5) Displacer Drive	• Posted Displacer and Gas Spring	• State-of-the-Art
6) Alternator		
- Plunger	• Rare Earth Permanent Magnet Manufacturability	• Adaptation of Current Technology
- Stator	• Manufacturing Technique (Microlamination, etc.)	• Adaptation of Current Technology
7) Control	• Engine/Alternator Stability Matching	• Adaptation of Current Technology
	• Displacer Gas Spring Volume Control	• Adaptation of Current Technology
	• Engine/Receiver Interface Control	• Significant Improvement

- Increased cycle efficiency
- Higher reliability
- Lower engine costs
- Improved heat receiver interface compatibility

The integration of the engine heater head with the solar receiver is paramount to the successful operation of the total system. One of the main problems with state-of-the-art combustion heat source Stirling engine heaters is in achieving a uniform temperature distribution across the head. It is expected that by utilizing a condensing liquid metal heat source, this problem will be diminished. Nevertheless, depending on the design configuration, attaining a uniform liquid metal surface wicking effect may be an equally challenging problem.

Because the heater is one of the more critical engine components, the design trade-offs and alternatives are apparent, and the return on successful development is great. It is strongly recommended that a component development program in this area be established.

1. Cost: The conceptual heater head design was established on the basis of a single-piece investment casting which incorporates the internal gas flow passages as well as the relatively thin-walled regenerator housing. This approach ensures a high degree of engine reliability at metal temperatures that will result in a good overall efficiency. However, the investment casting process required for fabricating the heater shape is labor intensive; thereby contributing to a relatively high cost for this component.

   Alternative concepts which show potential for lower component cost have been considered. However, in each case some compromise in either efficiency, reliability, complexity, or technical risk will be required. These compromises should be thoroughly evaluated during the earliest possible phase of advanced engine design work.

2. Life and Reliability: Although a brazed-tube heater design offers additional design flexibility for obtaining adequate heat transfer surface area with a minimum void volume, it somewhat constrains the estimated engine system reliability. One report (Ref. 1) on vapor and liquid metal heat transfer technology points out a concern for the potential for electrolytic corrosion of the braze joints. A materials compatability must be established for the heater and interface medium in order to retain the high reliability that is characteristic of free-piston machinery.

II. REGENERATOR

A. Sensitive Parameters: Efficiency, cost

B. Independent Parameters: Regenerator material

C. Considerations: A highly effective regenerator is vital for the realization of the potentially high-thermal efficiency that can be achieved with a Stirling thermodynamic cycle. In practice, a design compromise must be established so as to provide for high effectiveness at a reasonable regenerator material cost. The General Motor Research Laboratories have verified that a strongly coupled relationship exists between regenerator cost and performance (Ref. 2). Recently, new candidate regenerator materials have been identified which show a potential for achieveing good performance at reasonably low costs. These estimates have been substantiated on a preliminary basis with a first order computer code. What is required is to verify the performance of the various new candidate materials in a regenerator test rig and a test engine so that the effect of the matrix porosity and the heat storage element-geometry can be evaluated with respect to the overall impact on engine performance.

III. BEARING SYSTEM

A. Sensitive Parameters: Life, cost, maintenance

B. Independent Parameters: Bearing surface clearance, surface coatings

C. Considerations: Linear component gas bearings have been verified as an effective means for maintaining a dynamic condition between sliding surfaces without surface material contact. In theory, such a bearing design can produce an extremely long machine life expectancy and ensure a high degree of reliability. Nevertheless, during transient start/stop cycles, surface contact may occur. Surface hardening finishes have been incorporated into the design in order to prevent surface wear during these short periods. However, surface coating processes can be time consuming and expensive.

Presently, wear characteristics and start/stop effects are not well-known for linear bearing systems designed for the long-life requirements of solar electric applications. Better understanding of these effects may lead to the identification of lower cost surface finishing techniques, and the extension of the life potential for free-piston Stirling engines.

The 15 kW preliminary design incorporates equal bore diameters for both the alternator plunger and the engine power piston in order to ensure concentricity between the dynamic components and the mating surfaces. This approach enhances the design potential for achieving both good efficiencies and a high-degree of reliability. However, it imposes an additional design constraint on the matching of the alternator with the engine. Because significant cost advantages have been identified for alternator design concepts incorporating different bore diameters than that of the engine piston, methods for maintaining close bearing surface clearances on shafts with multiple journal diameters should be developed.

1. Life and Maintenance: Internally supplied linear gas bearings depend on:

   - Sufficient off-design bearing performance to ensure gas bearing operation early in the engine start-up sequence.

- Bearing surface coating material with sufficient hardness to retard wear during the surface contact portion of the engine start/stop cycle.

Possible changes in surface clearance due to wear during start/stop cycles is a function of the coating hardness and the bearing start characteristics. This appears to be one of the main mechanisms effecting bearing life. At present, insufficient amounts of empirical data preclude making an assessment of the surface wear rate for free-piston engines operating over the predicted duty cycle for a solar application.

2. Cost: Close tolerance machining of mating components will require that selective fitting manufacturing controls be implemented as part of the assembly process.

   The process of chromium oxide surface coating is an expensive one in that the coating material contains the strategic element, cobalt. In addition, the process requires special tooling. Finally, a second grinding operation is required after the coating has been applied to the bearing surface. This process essentially doubles the manufacturing cost of the bearing surface component. Alternative processes and materials such as electrolyzing or nitriding should be thoroughly evaluated in an effort to identify the most cost-effective coating.

IV. SEALS

A. Sensitive Parameters: Efficiency, cost

B. Independent Parameters: Surface clearance

C. Considerations:

1. Efficiency: Close tolerance, leakage limiting seals have been proven to be an effective means for controlling displacer and piston leakage, thereby contributing to high engine cycle efficiencies.

2. Cost: Machining cost becomes the limiting factor in the design of seal clearance.

V. ALTERNATOR PLUNGER

A. Sensitive Parameters: Cost, size, weight

B. Independent Parameters: Permanent magnet material

C. Considerations: MTI has an extensive history of experience in the development of linear alternators and linear motors. To date, hardware has been demonstrated at power levels of up to 17 hp. Overall alternator efficiencies of 90 percent have been measured for 1 kW designs.

One of the key design parameters to be considered for matching linear alternators with free-piston Stirling engines is the alternator plunger weight. In order to ensure good system performance, the plunger mass must be minimized. The requirement for a low mass becomes more difficult to achieve as higher power system designs are addressed.

For a fixed operating frequency and for a given magnetic flux density, the output power of the alternator is proportional to the flux area. This relationship leads to an increase in the length and diameter for achieving an increased alternator power output. In the 15 kW conceptual design, this increased flux area is obtained by an externally mounted (with respect to the stator) plunger design. In theory, this approach should be electro-mechanically similar to the conventional internal plunger designs.

A samarium-cobalt permanent magnet plunger, with a high magnetic flux density, was incorporated into the conceptual design to minimize the mass of the dynamic components. This design should ensure a good match between the engine and alternator dynamic characteristics, resulting in a potential for achieving a high-system performance. Rare-earth permanent magnet rotary motors and generators have been built and demonstrate good performance with high specific power output. Laboratory experiments with permanent magnet materials confirm the potential for achieving high specific power levels with linear alternators as well. The following discussions outline the specific design concerns that need be considered for such a super-magnet linear alternator design.

1. Size and Weight: Rare-earth permanent magnets allow for low weight, small size armature designs due to the high magnetic flux density associated with such material. Because of this, rare-earth (cobalt containing) magnets lead to relatively small alternators for a given power level. The more common ferrite-type magnets result in relatively larger design envelopes. The larger size complicates the structural design problems and adds complexity to the piston gas spring design as well.

2. Cost: Samarium-cobalt permanent magnets provide extremely high magnetic flux density. However, the material is quite expensive ($50/lb). Alternative mishmetal magnets offer compromise alternatives for lower cost designs at the expense of increased weight.

   As the world price of cobalt continues to increase, electromagnetic or ferrite magnet plunger designs may offer attractive alternatives to the higher cost, rare-earth magnets. The design alternatives should be evaluated now, and a continuous assessment of the impact on system costs should parallel the hardware development phases.

VI. ALTERNATOR STATOR

A. Sensitive Parameters: Manufacturing costs, efficiency

B. Independent Parameters: Lamination design

C. Considerations:

1. Cost: Conventional linear alternator stator designs employ single structure tapered laminations in order to keep the performance losses to a minimum. This approach appears to have potential for achieving reasonable manufacturing costs for the conventional internal plunger - external stator design. However, either rolling or machining tapered laminations for the internal stator concept incorporated in the 15 kW design may be a difficult and expensive process. The fabrication difficulty stems from the large degree of taper that must be achieved over relatively long lamination section lengths.

VII. CONTROLS

A. Discussion: Apart from the more obvious start/stop and safety controls required for a normal system operation, the solar engine control system must be designed for effecting an operational integration with the solar receiver energy flux. The approach that has been identified as being the most attractive method for optimizing the engine/alternator efficiency at off-design conditions is that of displacer gas-spring-rate control. This approach shows promise for maintaining the heater-head temperature at a constant level regardless of the energy input. With this scheme, the displacer/piston phase angle is continually adjusted so that the Carnot efficiency is maintained and the cycle efficiency is maximized.

3.2 Producibility

The objective of the effort applied to this subtask was to make an assessment of the manufacturability of the 15 kW conceptual engine design through the following activities:

- Identify materials, manufacturing processes, and assembly techniques
- Identify components that will require development of new mass production techniques
- Identify potential solutions for manufacturing problems
- Establish estimates of unit manufacturing costs
- Define a production learning curve

3.2.1 Production Rate

Since the problems associated with the producibility assessment are strongly dependent upon the annual level of production, an assessment of the most likely rate-of-production was completed. The techniques described in Reference 3 were used for this purpose. This method yields an estimate of the market replacement rate for technically innovative products as a function of market segment receptivity, investment requirements, investor's perceived rate-of-return on the investment, and the initial product exposure in the marketplace.

The relative measure of market receptivity is referred to as an Innovation Index. The Innovation Index used in this analysis was 0.60. This value is representative of the investment philosophy which might be characteristic of a high product throughput industry, such as the chemical industry. The chemical industry is a high-technology industry which is quite sensitive to product reliability and operational efficiency. Product reliability and operational efficiency are concerns which probably will be seriously considered by potential customers for dispersed solar power systems; therefore, the assumed Innovation Index appears to be justified.

The Profitability Index expresses the ratio of the rate-of-return associated with the innovation to the minimum rate-of-return for investments required by the customer. A value of 2.0 was assumed for the analysis. The investment ratio is an expression of the capital investment required as compared to the total assets of an average (electric utility) firm. A scan of the financial data reported for U.S. electric utility companies showed that $4 billion is a representative number for the total assets of an average size utility. A minimum investment of $1.5 million for 15 Mw of installed power was also assumed. These values yield a required investment which is less than 0.5 percent of a utility company's total assets. Market penetration estimates were calculated with the above assumptions for initial market exposures of 100, 500, and 1000 demonstration units, each with an output of 15 kW. The results are illustrated in Figure 3-2.

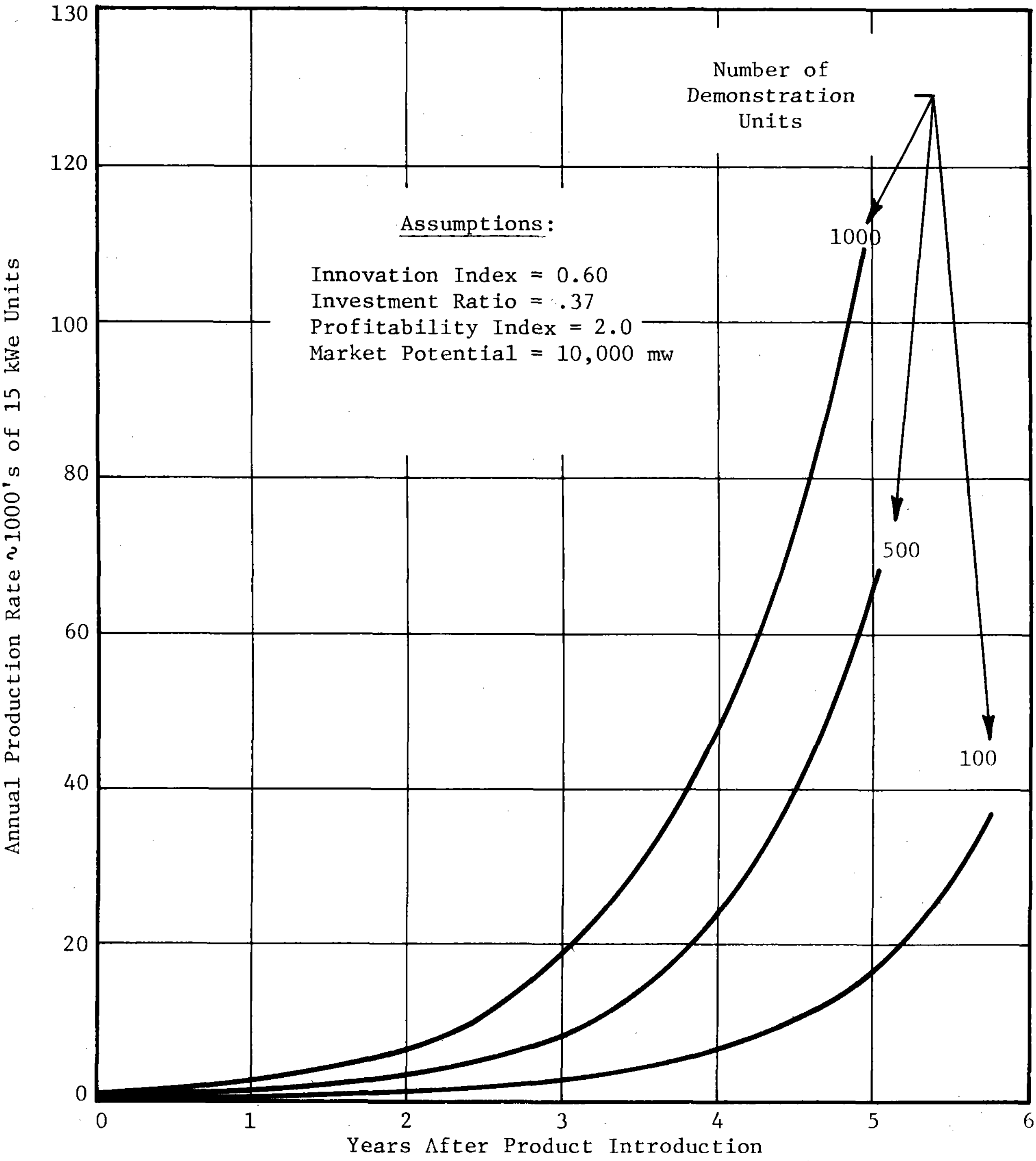

Fig. 3-2 Estimated Market Penetration for 15 kW Dispersed Solar Electric Power Units

It was further estimated that less than 100 demonstration units appear likely. In addition, it was determined that the initial production equipment planning should include a sufficient capacity to satisfy the forecasted sales requirements for at least the first five years. Therefore, based on the results presented in Figure 3-2, a production level of 25,000 units per year was adopted as the value for estimating production unit costs.

3.2.2 Unit Production Cost Estimate

Assuming an annual production level of 25,000, the following approach was implemented for assessing the material requirements and manufacturing techniques. A review of the 15 kW conceptual design (Figure 3-1 and Table 3-1) was made, and the component material requirements were identified by alloy and the weight was calculated. Suppliers were contacted in order to establish the various material availability and price. Based on the annual requirements and economical buy quantities, the material costs were established on a dollars-per-pound basis. Those materials that were determined to have availability difficulties were reviewed and alternatives were suggested. As a result, several design alterations were identified and incorporated into the conceptual design. A knitted wire was substituted for metal cloth in the regenerator matrix and cold-roll steel lamination material was substituted for hyperco in the alternator stator design. Additionally, various alternative permanent magnet materials were identified as potentially lower-cost material substitutes for samarium-cobalt. Due to a potentially negative impact on engine performance and the extensive design trade-offs that need be considered, samarium-cobalt was retained in the design for the producibility evaluation.

Preliminary make-versus-buy decisions were established on the basis of discussions with various material suppliers and machine shops that were contacted. The make items that were included are defined as follows:

- The raw material form such as casting, forging, bar stock, tubing, etc.
- The major machining operations such as turning, milling, broaching, grinding, heat treating, and the appropriate sequence.
- The special process requirements such as chrome oxide coating, anodizing, plating, nitriding, and the like.

The direct material costs are based on current vendor quotations for raw material and/or piece parts, but do not include incoming freight. Direct labor costs are based on a direct labor hour which was estimated by using a motion time method (MTM) approach. The raw times have been adjusted to the allowed times by dividing by 80 percent for machining labor and by 60 percent for assembly. This accounts for such nonproductive times as wash-up, break, and relief time. The direct labor rate used is $5.80 per hour and is an average for the operation.

A summary of the prime costs (direct material and labor) for most of the engine/alternator major subsystems are shown in Figures 3-3 and 3-4. Figure 3-3 indicates that the engine piston and displacer are largely labor intensive components. This is a result of the machining techniques that are required as part of the chrome oxide coating process. Alternative coating material might be identified which can result in attractive wear characteristics with substantially lower machining costs. In this area, component development should be aimed at gathering empirical evidence of wear rates for free-piston Stirling engines operating under solar duty cycles.

Figure 3-4 shows the total engine/alternator system prime cost (direct material and labor) and dramatizes the fact that in the conceptual design the samarium-cobalt permanent magnet material accounts for more than half of the total cost than the remaining components combined. Furthermore, the indicated heater head prime cost is nearly equivalent to all of the other engine alternator components, excluding the permanent magnets.

Obviously, these two areas (light-weight linear alternators and brazeless heater heads) present the greatest challenge to both design and production engineers. Likewise, these areas offer the greatest potential for realizing significant cost reductions. Because of the strong coupling between cost and performance in this area, it is beyond the scope of this effort to identify the most cost-effective design concept from an overall solar thermal system perspective.

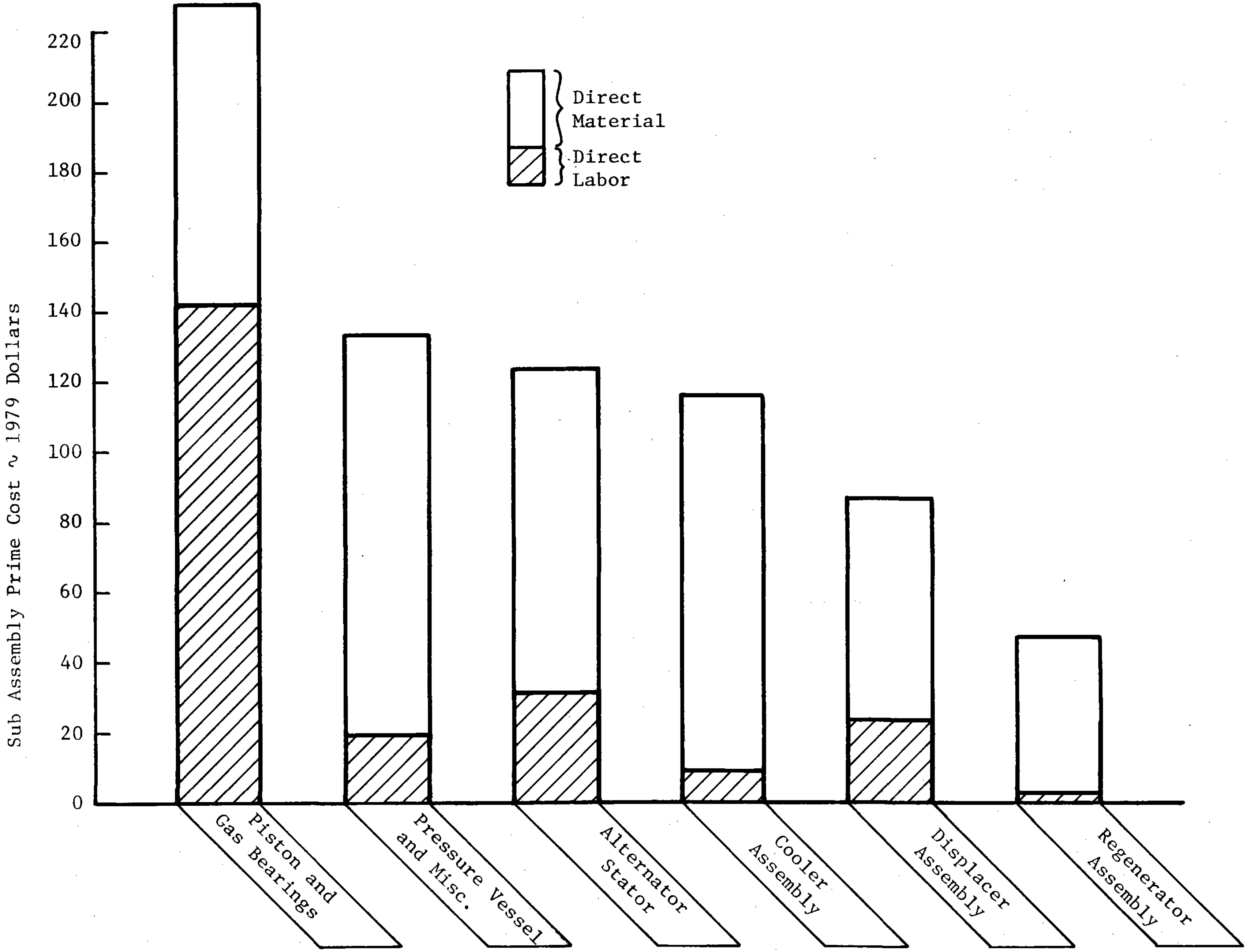

Fig. 3-3 Prime Cost Estimates for the 15 kW Solar Free-Piston Stirling Engine/Alternator Component Conceptual Design

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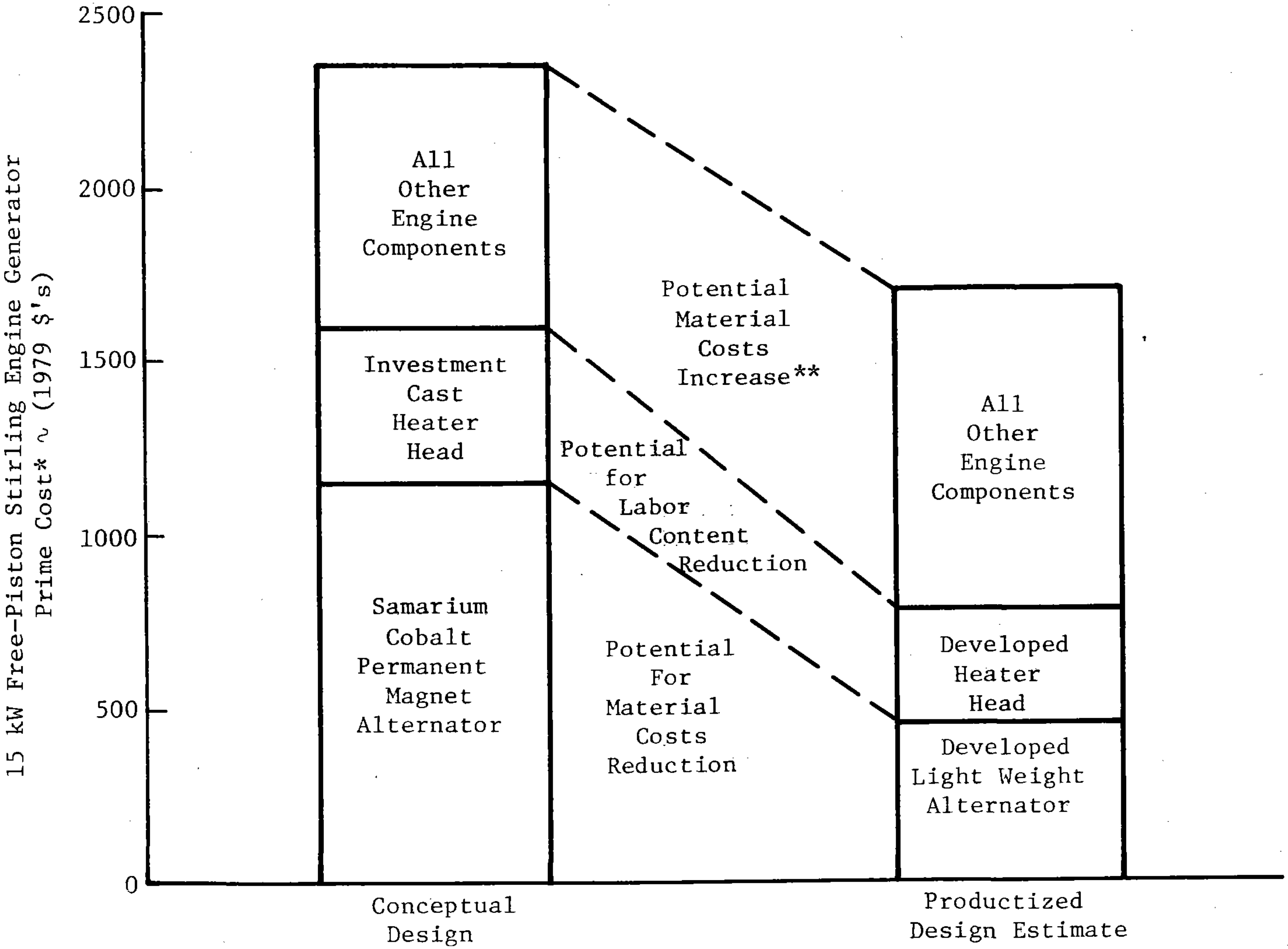

*Prime Cost Includes Direct Labor and Materials Only

**Possible Compromise to Accommodate Potentially Heavier Alternator Plunger Design

Fig. 3-4 Prime Costs Estimates for the Conceptual Design and a Developed Engine/Alternator System

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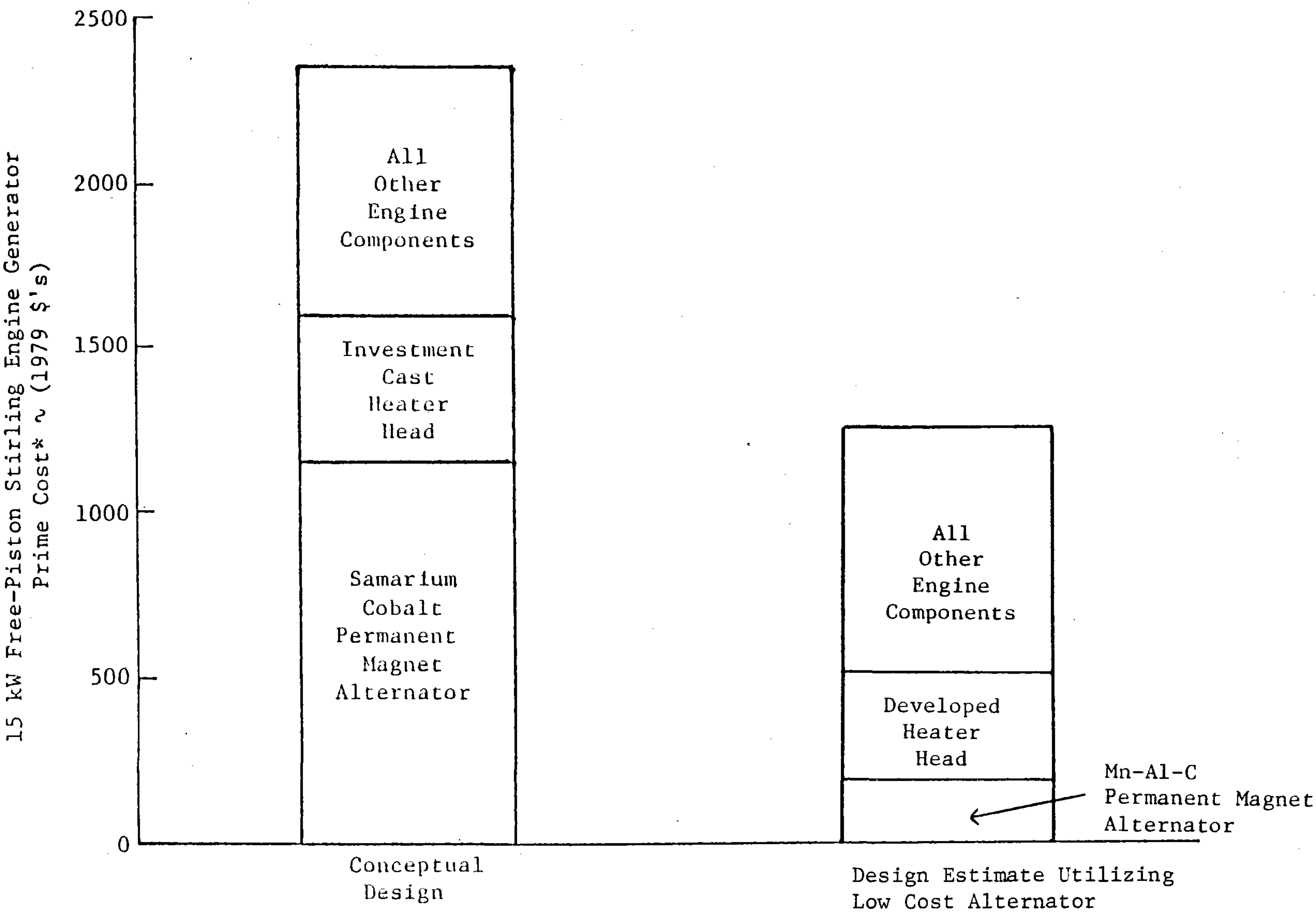
During the course of this study, several alternative low-cost, light-weight barium ferrite alternator configurations were conceptualized. Some of these concepts are reasonably innovative in nature and are deserving of a more detailed parametric evaluation. Nevertheless, the potential for identifying a low-cost, light-weight, high-efficiency plunger design appears to be within the grasp of near term technology. A high magnetic flux permanent magnet material (Mn-Al-C) has recently been identified as a good substitute material for the high cost samarium cobalt material. A cost reduction of two orders of magnitude may be realizable. A sizable reduction in the cost of alternator material can have a profound effect on overall engine-alternator costs. The possible cost reduction is shown in Figure 3-4a.

3.2.3 Predicted Learning Curve Effects

This section deals with the application of a learning curve to estimate the cost of 1, 5, and 100 engine/alternator sets.

The basic theory of the learning curve technique used in this study is that a worker learns as he works; and that the more often he repeats an operation, the more efficient he becomes. The result is that the direct labor input per unit declines. It has been shown (M. A. Requero, et. al), that this pattern can be quantified and a learning rate can be identified. For example, an 80 percent learning curve means that the fourth unit will require only 80 percent as much direct labor as the second unit; the twentieth only 80 percent as much as the tenth; and so on. Furthermore, the rate (slope of the learning curve) is a function of the ratio of direct labor machining to direct labor assembly (see Figure 3-5). This occurs because the assembly effort is more people oriented, and therefore, has more room for the learning experience. Machining, as the term implies, is machine dependent. In this situation, the important point is that learning applies only to the direct labor. Therefore, to utilize this technique for the 15 kW Stirling engine/alternator design concept, the unit labor content must first be evaluated. In addition, the percentage of labor which is assembly related must also be determined.



*Prime Cost Includes Direct Labor and Materials Only

Fig. 3-4a Prime Costs Estimates for the Conceptual Design and a Developed Engine/Alternator System for Production Quantities of 25,000 Per Year

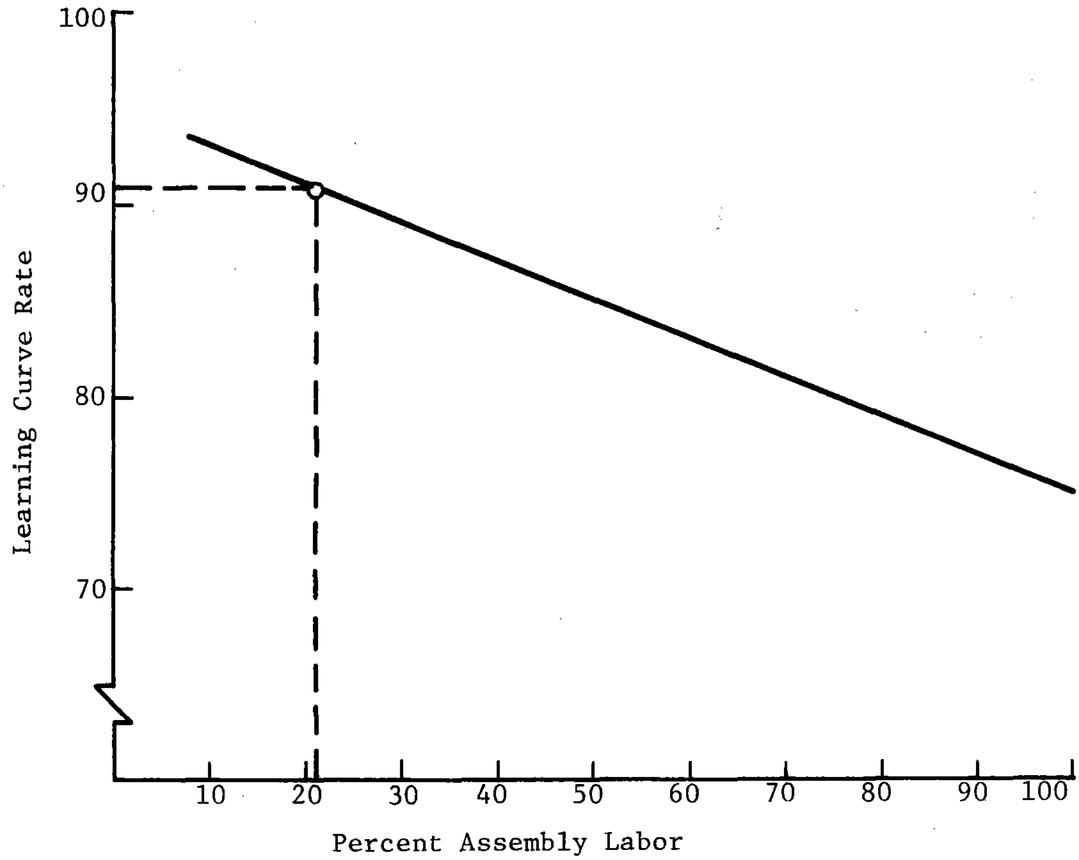


Fig. 3-5 Learning Curve Rate As A Function of Assembly Labor

The approach used here is to determine the labor content for existing MTI Stirling engine/alternator prototypes on a component basis, and to apply appropriate scaling factors to estimate corresponding values for a 15 kW design concept. The two primary assumptions involved are:

1) The learning rate determined for the existing engines can be defined using the direct labor ratio.

2) This learning rate is applicable to all Stirling engine/alternators of this type.

The second is not a bad assumption, considering that these engine/alternators are all single-cylinder, free-piston, gas-bearing, linear alternator designs. The prototype used to define these parameters is a 1 kW unit now in the process of final assembly. The cost breakdown is shown in Table 3-3.

Although the cost data presented in this table reflects prototype hardware, it has been adjusted to reflect costs that could be expected if MTI were to build the 1 kW machine again in the previously mentioned quantities. In other words, it is assumed that the laboratory type of instrumentation packages will not be included in the deliverable units.

This is an important point since the following analyses assume that MTI has designed, manufactured, assembled, and tested a 15 kW prototype unit and that the costs for 1, 5, and 100 units is proceeding from this "tried and true" design.

From the values listed in Table 3-3, a labor assembly to labor machining ratio of 21 percent can be calculated. Figure 3-5 shows that this is indicative of a 91 percent learning rate. The next step is to determine the single unit 15 kW cost and then apply this learning rate to establish the various multiple unit costs.

The single unit 15 kW costs are detailed in Table 3-4 and, as with Table 3-3, reflects the "tried and true" design.

TABLE 3-3

1 kW SINGLE UNIT COSTS

Labor Machining		Dollars	Hours[1]
Bearing/Pressure Vessel		37,000	1,850
Engine (less regenerator)[2]		27,000	1,350
Alternator		3,000	150
	TOTAL	67,000	3,350

Labor Assembly		Dollars	Hours[1]
Bearing/Pressure Vessel		6,000	300
Engine		10,000	500
Alternator		2,000	100
	TOTAL	18,000	900

Material		Dollars
Bearing/Pressure Vessel		7,000
Engine		5,000
Alternator		2,000
Instrumentation		5,000
	TOTAL	19,000

1 Based on $20.00 per hour

2 Regenerator used in the prototypes is wire screen at very high cost and will not be the production approach

TABLE 3-4

15 kW SINGLE UNIT COSTS

Labor Machining		Dollars
Bearing/Pressure Vessel		42,000
Engine		30,000
Alternator		6,000
	TOTAL	$78,000

Labor Assembly		
Bearing/Pressure Vessel		6,000
Engine		10,000
Alternator		4,000
	TOTAL	$20,000

Material		
Bearing/Pressure Vessel		9,000
Engine		7,000
Alternator		8,000
Instrumentation		5,000
	TOTAL	$29,000

The learning curve can be mathematically expressed as follows:

$$Y = K_x^{\eta}$$

where: Y = cumulative average hours for any number of units

K = number of hours to build the first unit

x = number of completed units

η = log (% of learning curve) ÷ log (2)

Applying this equation to the \$98,000 single unit labor costs calculated for the first 15 kW assembly, results in the curve are shown on Figure 3-6. Utilizing the appropriate labor dollars from this curve and adding in the \$29,000 constant material dollars gives the 1, 5, and 100 one-time only average unit costs of:

Quantity	Labor \$'s	Material \$'s	Total/Unit
1	98,000	29,000	\$127,000
5	78,700	29,000	107,700
100	52,400	29,000	81,400

It is assumed that, in these small quantities, the unit material costs will not change significantly as a function of production volume.

3.2.4 Cost Sensitivity to Production Volume

The following discussion summarizes the work that was completed in the identification of engine cost as a function of production volume.

The relationship between unit cost and production level can be characterized by a decrease in unit cost as the production quantity increases. This effect is primarily a result of applying more sophisticated manufacturing techniques, which is justifiable from increased sales. This should result in a reduced component labor content, and thus, a reduced unit cost. This applies to purchased materials such as castings, forgings, bar stock, nuts, bolts, and packing boxes as well. (The volume cost reductions may occur outside the actual manufacturing site but will still be reflected in the final cost.)

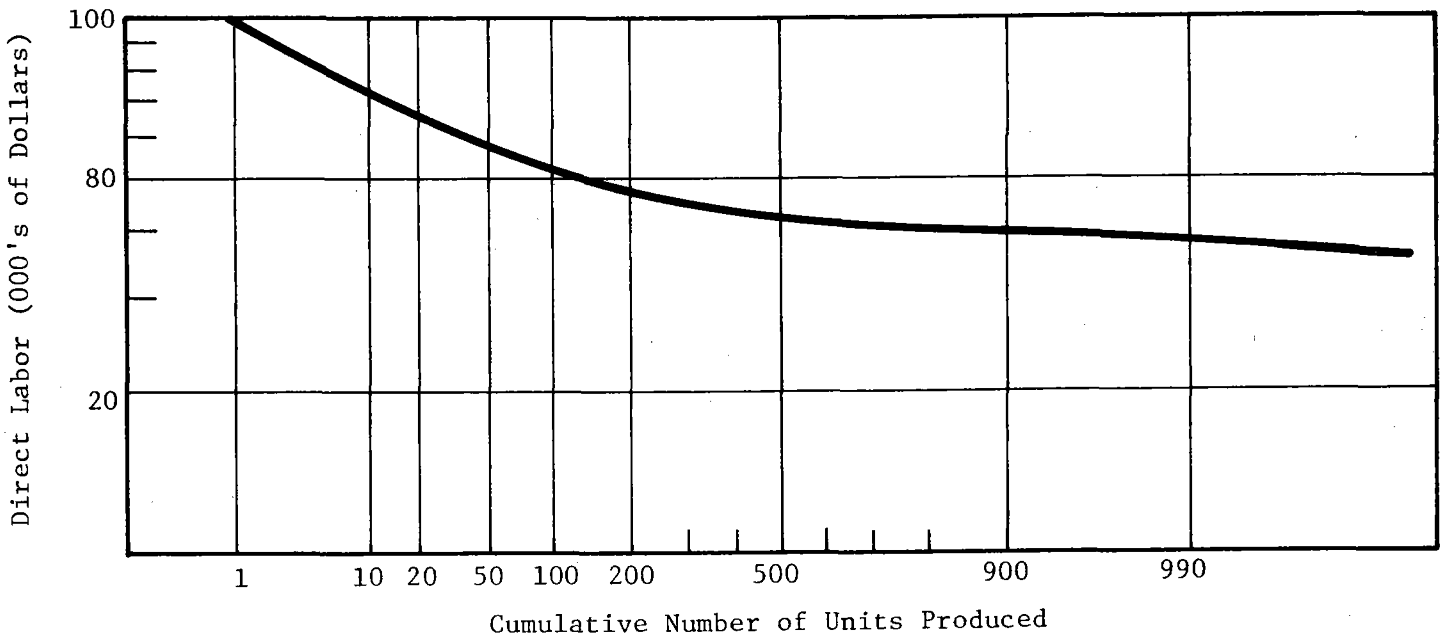


Fig. 3-6 15 kW Stirling Engine
91 Percent Learning Curve

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This cost-to-volume relationship has been established for the 15 kW conceptual design, using a 25,000 annual production level unit cost and the single unit cost as presented in the previous section. Material costs were evaluated at lower (5,000 units per year) and higher (100,000 units per year) quantities (see Table 3-5) while the labor content has been adjusted via the basic learning curve equation:

$$Y = K_x^{\eta}$$

where the symbols have the same definition as before. By inputing the single unit labor cost and the 25,000 unit labor costs, the exponent η can be determined. The curve shown in Figure 3-7 shows the labor cost as a function of volume. The material versus volume curve is superimposed and, at any production level, the prime cost is the sum of these two values. Figure 3-7a shows the same information as presented in Figure 3-7 with the assumption that a low cost alternator material such as Mn-Al-C be utilized in place of the samarium cobalt material.

3.2.5 Commercialization Potential

The potential for commercializing any technical innovation is most likely to be dependent upon the predicted rate-of-return on investment as perceived by the prospective investor, as well as the financial exposure or risk associated with the investment. The rate-of-return can be calculated as a function of the market potential, the penetration rate, the profit margin, and required capital investment. One of the key market parameters, the penetration rate, is, to a large extent, product feature dependent. In other words, it can be estimated in terms of product features such as efficiency, maintenance requirements, reliability, price, and other input to the product life cycle and first cost calculations.

The level of risk or financial exposure is another indication of commercialization potential. The financial exposure is strongly dependent upon the required investment for plant, equipment, and tooling. These investment parameters are evaluated with respect to the total assets required to implement the business. The following discussion addresses this aspect of commercialization potential for free-piston solar Stirling engine generators.

TABLE 3-5

MATERIAL COSTS FOR CHANGE IN PRODUCTION VOLUME*

	Annual Production Quantity		
Component	5,000	25,000	100,000
Heater Head	1.2	1.0	.8
Alternator	1.24	1.0	.83
All other Engine Components	1.05	1.0	.95

*All costs shown as a ratio of costs for 25,000 units per year production rate.

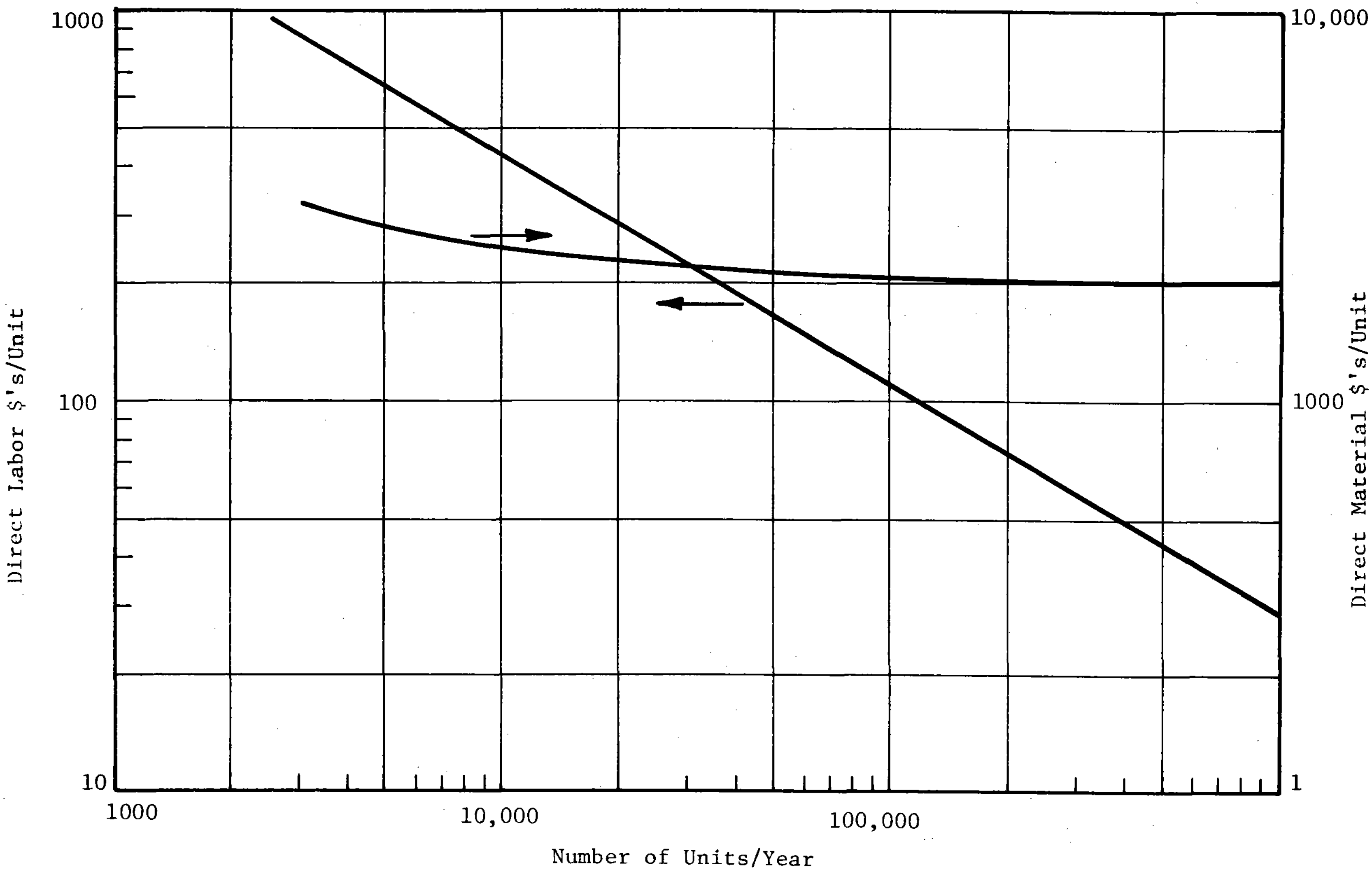

Fig. 3-7 Material/Labor Dollars as a Function of Volume

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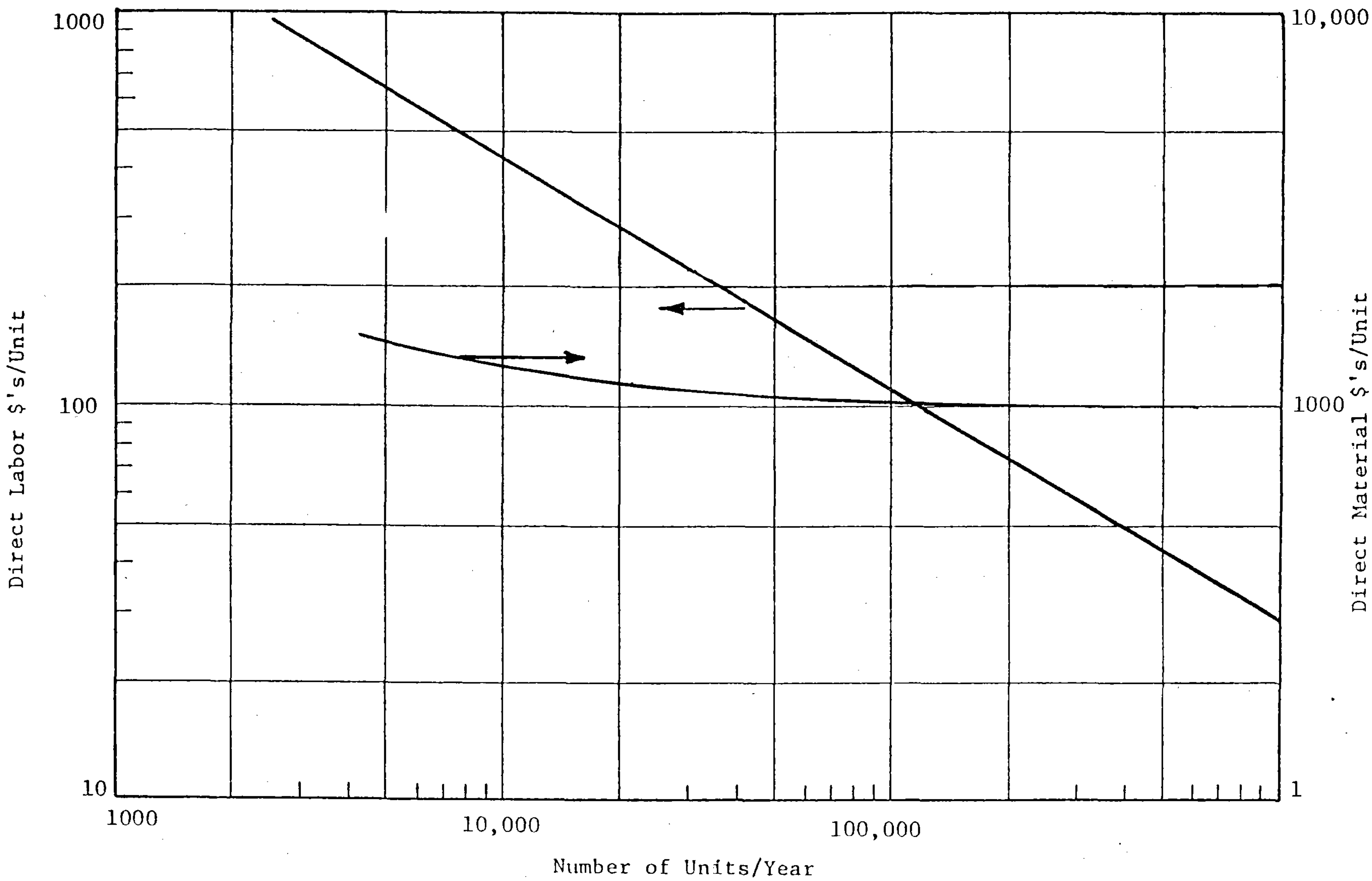

Fig. 3-7a Material/Labor Dollars as a Function of Volume

3.2.5.1 Plant and Equipment. The machinery and machining tools identified for fabricating the components of the conceptual design are, for the most part, standard pieces of equipment. The bulk of the operations involve commonly used metal shop techniques such as drilling, boring, milling, etc. Large, horizontal lathes will be required for finishing the tubular sections such as the piston cylinder and alternator plunger structure. In general there appears to be a good availability of horizontal lathe capacity in the U.S. Estimated delivery time for most of the machine tool parts is between eight and fourteen months.

For annual production rates of 25,000 units, special automatic welding machines will be required. These machines feed electrode wire to a multiple gun fixture; parts are automatically stacked and assemblies rotated. Fully fabricated pieces are produced at a single station. It was determined that about three or four such automatic welders would be required to meet capacity requirements. These types of machines are used with regularity in the automotive and related industries.

All total, about thirty metal processing machines will be needed to complete all of the fabrication and assembly operations identified. The average cost of each machine is in the neighborhood of $250,000. An estimated three to four years will be required to prepare a fixed site for manufacturing the power conversion units at a single location. Obviously, these figures represent a sizable investment in both capital and time. Because of this, and because of the long lead-time required for factory construction, it might be desirable to initially subcontract all of the component fabrication. Based on the vendor contacts that were made, it is possible to have all of the parts made at existing machine shops with existing tooling. This approach requires little investment for machines or tooling and presents an attractive commercialization scenario for prospective firms evaluating the small Stirling engine manufacturing business.

The alternative manufacturing approaches have little impact on the prime costs listed in Table 3-2, since these costs reflect direct labor and material charges only. However, the factory selling price is expected to be higher in the case of totally subcontracted fabrication since profit margins would be realized by the subcontracted machine shop, as well as manufacturer. In essence, the central manufacturing facility in this scenario would simply consist of an assembly and set up plant.

Because of the large diameters, long lengths, and close tolerance surface clearances which characterize the 15 kW conceptual design, unique assembly requirements can be anticipated. An assembly control plan would necessarily be implemented for selective fitting of parts. In addition, a high quality clean room would be required for assembly. This will help to ensure that the design's inherently high reliability is maintained in production engines.

Generally, the potential for a firm to enter into the manufacturing of free-piston Stirling engines with a minimal investment in plant and equipment leads to a high commercialization potential in terms of the initially moderate financial risk involved. The calculated rate-of-return on the investment depends on the market potential, sales forecast, and profit margin that can be established for the potential product as seen in the context of the total end-use system. This aspect of product potential has not been addressed here. Nevertheless, it appears that the attractive features associated with free-piston Stirling engines such as high efficiency, long life, and high reliability will enhance the market acceptance of the entire solar thermal power conversion system. These features, coupled with the potential for an initially low investment in plant and equipment, indicate that a reasonably high commercialization potential can be justified for free-piston Stirling engine generators in solar thermal power applications.

3.3 Durability

3.3.1 Reliability

The approach taken to assess the durability of the solar Stirling engine/ alternator was to examine the engine alternator drawing (Figure 3-1) and associated parts list (Table 3-1) to determine the function, failure mode, possible failure cause, and failure effect on the system. In order to assess the impact of the failure, a failure probability (failure rate) was assigned to each failure mode. Failure probabilities represent average values for mechanical failure modes from industry. Based on this preliminary assignment of failure probabilities by design item, an overall engine/alternator failure rate of 20.859 failures per million hours was estimated. This corresponds to a reliability in terms of the mean time between failure (MTBF) of 47,940 hours.

A list of those design items that could cause a system shutdown as the result of its failure is presented in Table 3-6. These items account for a cumulative "shutdown" failure rate of 4.05 failures per million hours. In effect, less than one-fifth of all possible engine alternator failures would result in a total disablement of the system.

The reliability established as the result of this preliminary assessment is encouragingly high. As a comparative reference, a typical MTBF for auxiliary power units on jet aircraft is in the neighborhood of 15,000 hours. In order to obtain more accurate reliability estimates for free-piston Stirling engine/ alternator systems, additional empirical data must be obtained. These data should include such information as bearing surface wear rate, regenerator deformation rate, statistical debris particle size, and other physical evidence which can impact the life expectancy for the gas bearings, close tolerance seals, and other critical engine components.

TABLE 3-6

FAILURES RESULTING IN POTENTIAL SHUTDOWN

Item No.	Distribution	Failure Rate (failures per 10^6 hours)
2	Displacer Orifice Bleed	1.00
4	Clamping Ring	.20
11	Cooler Assy	.20
12	Cooler Tubes	.12
15	Displacer Spider & Post	.10
16	Displacer Post Inner	.10
17	Displacer Post "O" Ring (6 ea)	.375
22	Engine Cylinder	.10
23	Cylinder Flange "O" Ring	.375
24	Port Seal "O" Ring	.375
25	Cylinder OD "O" Ring	.375
28	Alternator Stator Coils	.137
29	Spring Washer	.200
30	Tie Bolt	.02
31	Plenum Cap	.27
32	Locknut	.10
	CUMULATIVE TOTAL	4.047

3.3.2 Recommended Maintenance Plan

Based upon a review of the FMEA and recommendations from Design Engineering, a maintenance concept has been formulated to provide assurance that the Stirling engine/alternator will have an extended operational life.

Following is an itemization of the recommended preventative maintenance action: At six month intervals:

- Monitor working gas consumption
- Check overall engine/alternator for gas leakage
- Check cooling systems for any water leakage

If engine teardown is necessary for a repair, accomplish the following:

- Displacer gas bearing parts pair (displacer and post)
  - Examine for wear or surface coating deterioration. Feed bearing with high-pressure gas and measure gas flow. Flow value to be determined (TBD)
- Piston gas bearing parts pair (cylinder and piston)
  - Examine for wear or surface coating deterioration
  - Feed bearing with high pressure gas, measure gas flow. Flow value to be determined (TBD)

When disassembly is required, disassemble at item 24 on assembly drawing. Remove nuts and studs, split item 24 and 23. Visually check displacer and piston gas ports and working surface for contamination and wear. In the event of complete teardown, replace the "O" Rings.

Maintenance Care: when an engine/alternator is disassembled for periodic maintenance or repair, extensive caution must be exercised to prevent foreign material (contamination) from entering the machine. Maintenance must be performed in a clean area, free of dust, metal particles, etc.

The maintenance concept is designed for avoiding any disassembly in the machine unless either a verified failure has occurred or a substantial maintenance task has been identified.

3.4 Growth Potential

An assessment of the free-piston Stirling engine generator growth potential was completed. This study addressed both the near-term growth potential of a near-term-type design and the long-term growth potential of large power free-piston concepts. In each case the objective was to identify the approaches by which increased power output might be realized, to assess the limitation of the approach, and to evaluate the impact of alternative scaling methods on system performance and design feasibility.

3.4.1 Near-Term Growth Potential

The analysis addresses a fixed, near-term-type engine/alternator design. Methods by which power output can be increased are evaluated for a given design. The baseline design is the same 15 kW engine/alternator conceptual design that was evaluated in the producibility assessment completed in Subsection 3.2. The system operating parameters which have a significant impact on power output were identified. Those which offer the most promising opportunity for increasing the power output include:

- Increase in charge pressure
- Increase in engine piston spring rate
- Increase in engine stroke
- Increase in heater-head temperature

A first-order model computer code was utilized to determine the sensitivity of the generated engine power to variations in each of the operating parameters. The results of this analysis are shown in Figures 3-8 through 3-11. In each of the figures only the parameters shown were allowed to vary. All other operating and design parameters remained constant.

One of the most direct methods for increasing the output power of a given free-piston Stirling engine is to raise the engine charge pressure. This effect is illustrated in the results plotted in Figure 3-8. However, due

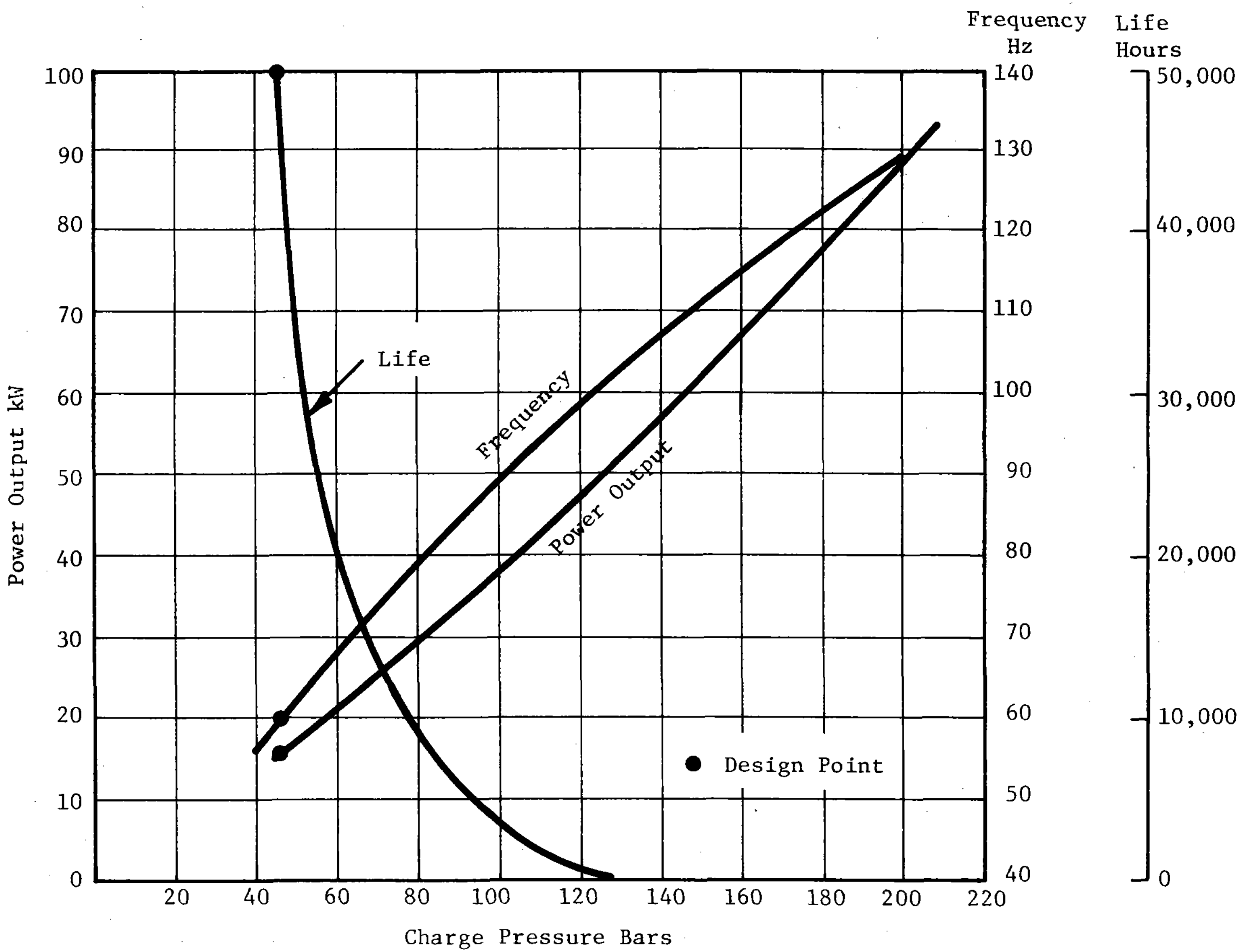

Fig. 3-8 Power Output versus Charge Pressure for the 15 kW Conceptual Design

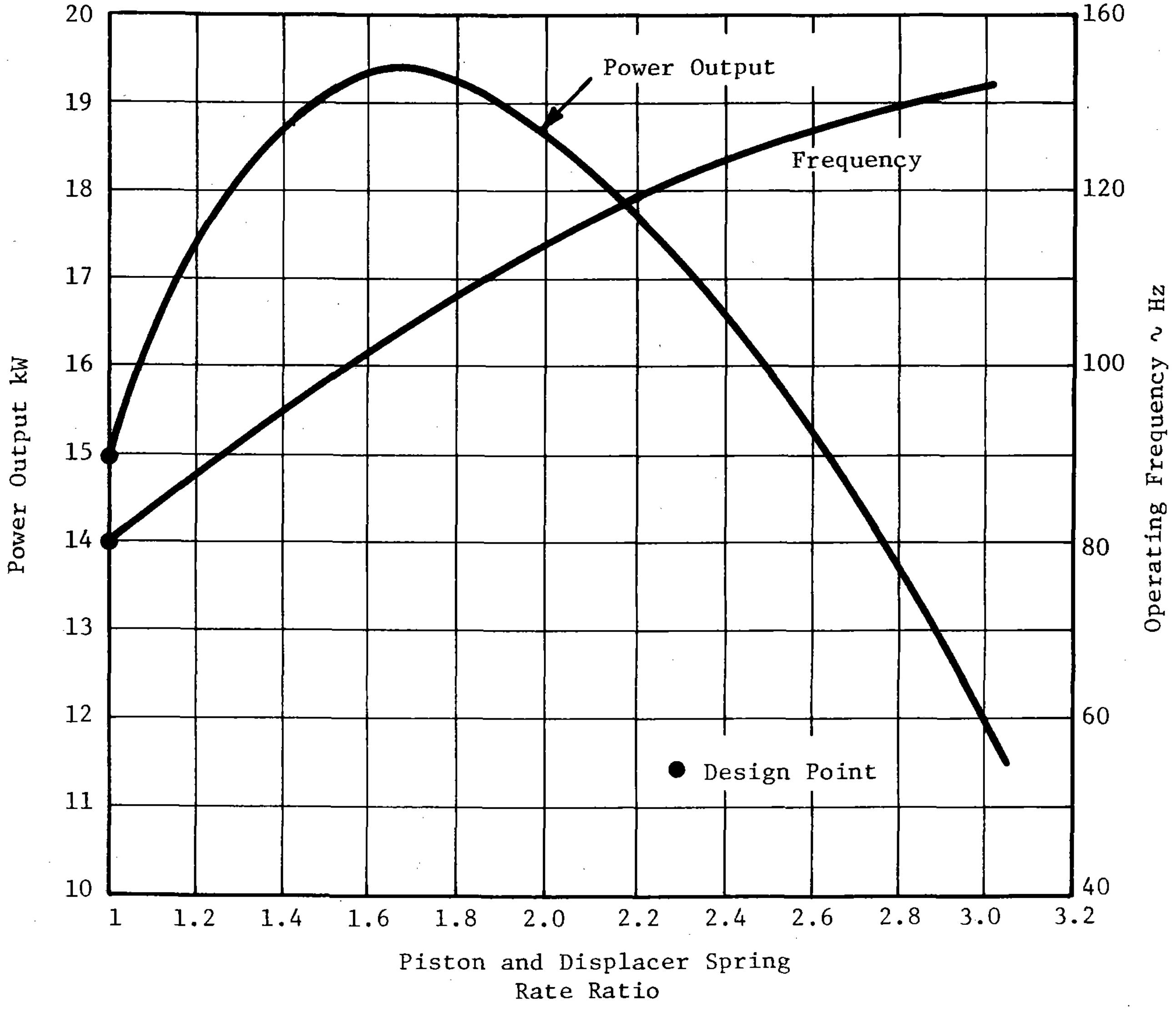

Fig. 3-9 Power Output versus Gas Spring Rate Ratio

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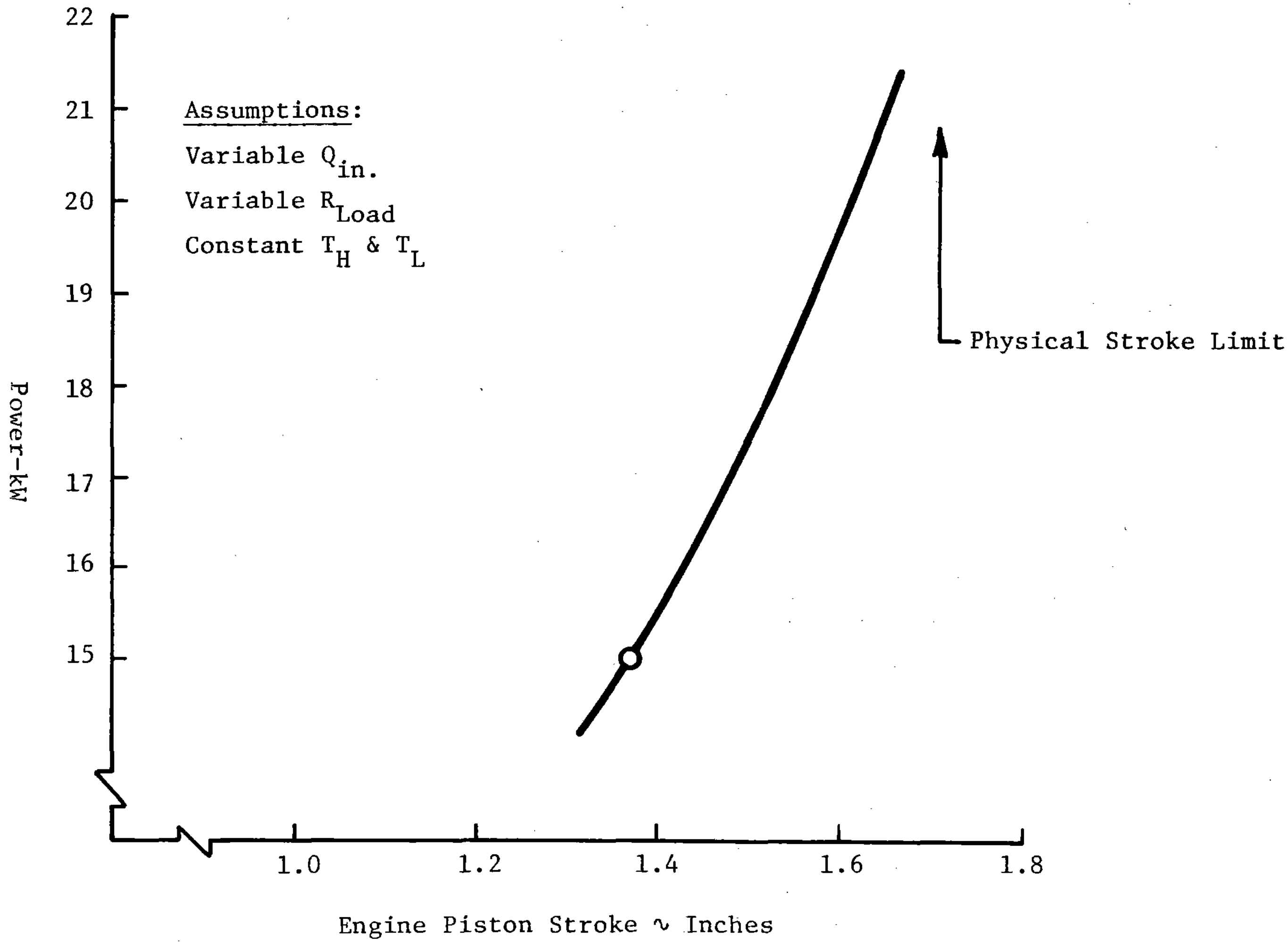

Fig. 3-10 Variation of Engine Power as a Function of Piston Stroke for the 15 kW Solar Conceptual Design

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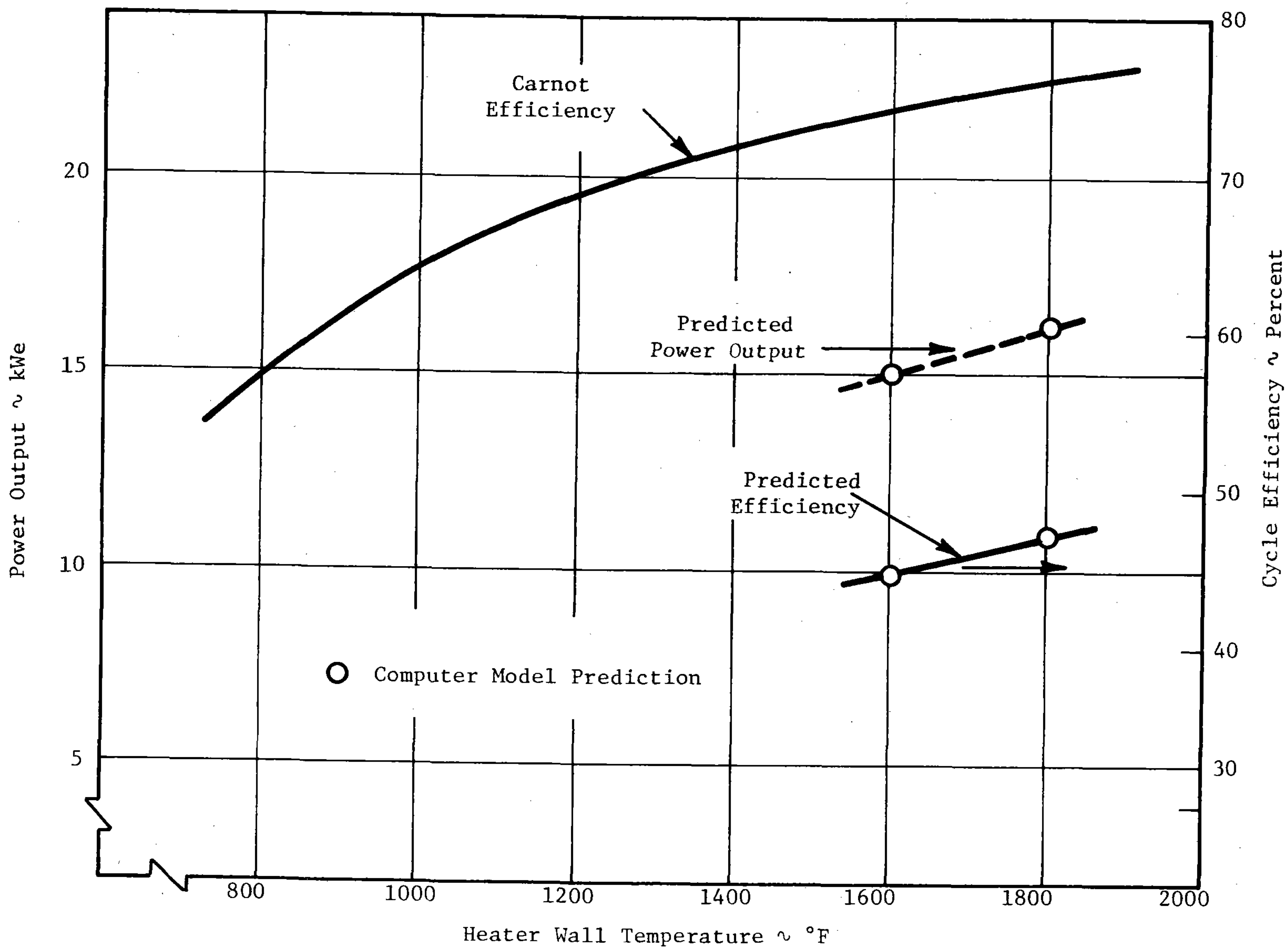

Fig. 3-11 Power Output and Efficiency versus Heater Wall Temperature for the 15 kW Conceptual Design

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to high mechanical stresses and large pressure drops, the direct power increase is realized at the expense of a sharply reudced life expectancy and a decrease in cycle efficiency. The efficiency penalty is a direct result of the increase in operating frequency that follows an increase in engine pressure. The increase in both frequency and gas density increases the flow losses in the heat exchangers at a rate which is greater than the increase in developed engine power, thereby causing a degradation in engine efficiency at higher charge pressures.

MTI has designed and built externally controlled piston and displacer gas springs. The gas spring rate in a fixed engine design can also be increased by filling the bounce space with a filler material, thereby decreasing its effective spring volume. Figure 3-9 illustrates how the power output responds to an increase in the dynamic components' spring rates. The power output will actually increase to a certain point at which the dynamic relationship between the displacer and piston deteriorates resulting in significant decrease in power output. Therefore, the maximum power output that can be realized by increasing the gas spring rate (other than by increasing the gas pressure) is limited.

Figure 3-10 indicates that an increase in the engine power output will result as the piston stroke increases. This might be accomplished by reducing the load resistance to the alternator. Engines are designed to accommodate a 25 percent increase in overstroke before a physical stroke limit is encountered.

With a design point expansion space temperature of 1600°F, the potential for realizing an increase in power by raising the hot side temperature is quite limited. There is little to gain in power between 1600°F and 2000°F however efficiency will increase from 45 percent to 47.5 percent. Figure 3-11 shows the results predicted by an analytical model as compared with the increase in Carnot efficiency and operating temperature.

In general, it is expected that the near-term power output of a test-bed type 15 kW free-piston Stirling engine generator can be increased to about 20 kW through one or more combinations of the operating parameter variations previously described. Beyond 20 kW, significant life and efficiency penalties would be incurred.

3.4.2 Long-Term Growth Potential

Methods for growing the conceptual design to large power output levels (approaching 500 kW) were reviewed. The present task required that the fundamental concern of at what power level the single cylinder free-piston Stirling engine design concept becomes unfeasible be addressed. Initially, an approach was adopted in which all of the design parameters (e.g., operating frequency, stroke-to-bore ratio, alternator type, etc.) were unconstrained. Various combinations of design parameter changes were established, and computerized performance prediction models of the designs were generated. The design changes as compared with the 15 kW solar engine included combinations of:

- Increased operating frequency
- Increased charge pressure
- Increased stroke
- Increased engine diameter
- Increased alternator diameter and length
- Decreased void volume

The two conclusions that were drawn from this exercise are: 1) the number of design parameter combinations resulting in an increased power output is very large; and 2) all of the combinations examined required significant size scaling. Based on these axioms, a more restrictive approach to the problem was adopted. The methodology consisted of the following items:

- Apply a common scaling factor (Z) to all dimensions
- Simplify all relationships to a first order form to achieve a closed-form solution
- Assume that the 15 kW conceptual design is an optimum for that level of power output (this is not necessarily true)
- Utilize the 15 kW conceptual design as a baseline from which all geometries are scaled linearly

- Scale the heat transfer loop (heater, regenerator, and cooler), the engine, and the alternator by the same dimensional scaling factor (Z)
- Identify and assess the specific concerns relative to dimensionally scaling, separately for each system component

  e.g.

Engine:	Heater:	Alternator:
- gas bearings	- manufacturability	- side forces
- thermal conductance	- thermal conduction	- magnetic saturation
- vibration		

- Address integration problems between each of the three components separately

In addition, the following assumptions were adopted:

- frequency = constant
- (void volume/working space volume) = constant
- design temperatures = constant

The analysis indicated that the operating frequency of the machine will tend to slow down as the power level and size increases. In order to keep the frequency constant, an additional scaling factor (j) was applied to the engine charge pressure. A constant design frequency demands that the following relationship be satisfied:

$$j = Z^2.$$

Based on this equation, additional scaling relationships were developed to account for the increase in charge pressure as a function of dimensional scaling.

The scaling relationship for the power output and the heat input are shown in Figure 3-12. If the displayed trends are valid, then the engine efficiency will eventually exceed Carnot efficiency, thereby making the results of the simplified scaling approach unacceptable. This means that if the dimensions of a given engine design were scaled directly, then the heater head, operating at a fixed wall temperature could not transfer a sufficient amount of energy into the working fluid to support the potential growth in engine power. In

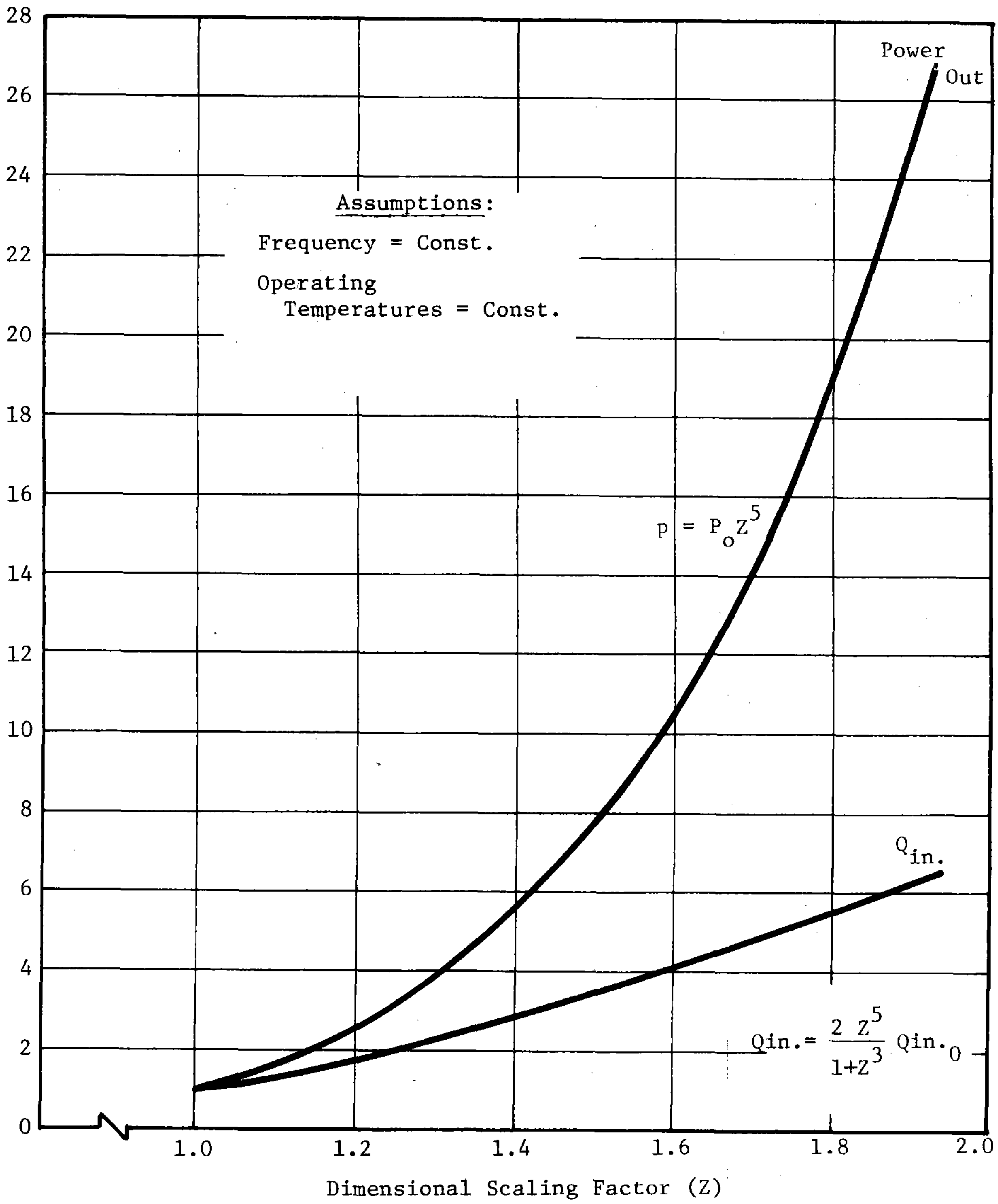

Fig. 3-12 Engine Power and Heat Transfer Ratios As a Function of Scaling Factor

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an actual case, the increased engine design stroke and corresponding power output would never be achieved without increasing the hot side temperature.

In order to correct for the apparent mismatch between the developed engine power and the available heat transfer, an additional constraint was introduced into the analysis. The heat transfer components were scaled independently of the engine's nonheat transfer components. With this approach, the scaling of heat transfer to the working gas can follow the increase in developed engine cycle power output as required.

Following are the scaling factors that were employed in the analysis:

Z = engine and alternator dimensional scaling factor

θ = heat transfer loop scaling factor (diameter and length)

G = heat transfer loop scaling factor (number of gas passages)

A similar type of scaling analysis was completed for the linear alternator. The results indicate that the output power capability of the alternator grows with the cube (Z^3) of the scaling factor for an assumed constant operating frequency. Since the engine power grows with Z^5, additional alternator design changes would be required to obtain a match between the alternator and engine. This match might be obtained by:

- Scaling the alternator diameter by Z^3.
- Providing for additional alternator segments at the rate of Z^2
- Substituting a high field intensity electromagnetic plunger for the permanent magnet design.

Any of these design alternatives leads to a plunger weight that scales at a proportion greater than Z^3. However, a dynamic component weight scaling proportional to the cube of Z was assumed in the simplified analysis. Therefore, the engine gas spring design would require appropriate modification to maintain a constant 60 Hz operating frequency at higher power output. This could be accomplished by increasing the engine piston effective gas spring surface area for high output power designs.

Besides alternator power matching, other practical problems associated with increasing the design power of a free-piston Stirling engine are high charge pressures and large numbers of heater tubes. Although the operating pressures shown are not excessive, there might be some safety concerns associated with maintaining large diameter, high-pressure engine casings.

This study comprises a preliminary assessment of the design problems that might be encountered as the free-piston Stirling engine concepts address larger levels of power output. This study employed some very simplified physical relationships and qualifying assumptions. In a more detailed design study, some of the simplifying assumptions used here may be proven to be invalid for wholesale extrapolations. Certainly, innovative design techniques could be employed to obtain extended heater head and alternator power scaling. Nevertheless, the design concerns identified in this study are real; they typify the generic problems that need be addressed in the design analysis of large power free-piston Stirling engines.

Based on this analysis outline, the following conclusions can be supported:

- Limitations of the heat transfer surface preclude the direct linear scaling of free-piston Stirling engine designs for increased power.
  - Both the heat transfer loop and the alternator must be scaled separately in order to obtain a match with the developed engine power output.
- For a constant design frequency, the engine charge pressure scales with the square of the scaling factor (Z^2) and the engine power scales with the fifth power (Z^5).
- The alternator power scales with the cube of the scaling factor (Z^3); therefore, an additional alternator structure (e.g., more segments) is required to obtain a match with the engine power.
- A stiffening of the engine piston gas spring is required to compensate for the additional alternator weight and, thereby, maintain a constant design frequency.

- The maximum design power level for a single cylinder free-piston Stirling engine appears to be limited by the heater design and the piston dynamics constraints.

The following are based on the results of a simplified, first-order analysis.

- Conventional, first-order design approaches appear to breakdown for system power levels of 50 kW or greater.
- Beyond 50 kW, second-order effects and innovative design techniques should be incorporated into the analysis.

3.4.2.1 High Power Engine Conceptual Layout. A conceptual layout for a large power engine system was completed based on the results of the first order scaling analysis. The concept design was established with the following assumptions:

- The scaling relationships discussed in subsection 3.4.3 were adopted as the basis for power scaling.
- The design incorporates an appropriate match between the heater head, engine, and alternator subsystems.
- The engine system was designed for an optimum efficiency according to scaling relationships.
- Additional piston gas springs were incorporated as required for maintaining a constant 60 Hz operation.
- Multiple engine cylinders were configured so that the coolant fluid plumbing interface results in single external connections for the inlet and discharge lines to the total system heat exchangers; a unitary receiver configuration could be incorporated into the design.
- Multiple engine cylinders would be dynamically phased so that a three-phase electric signal characteristic results.
- The arrangement of multiple cylinders would be such that a vibration-free cluster results.

These design efforts resulted in the six-clustered cylinder arrangement shown in Figure 3-13. The net forces and moments for the engine system is equal to zero and the system should be vibration free. The engine is designed for a total output power level of 250 kWe with an overall system efficiency of 43.2 percent. Innovative heater concept alternatives which might extend the output power were discussed during the design exercise. However, a considerable analytical effort is required in order to substantiate these approaches. It was determined that this type of effort is beyond the scope of this study.

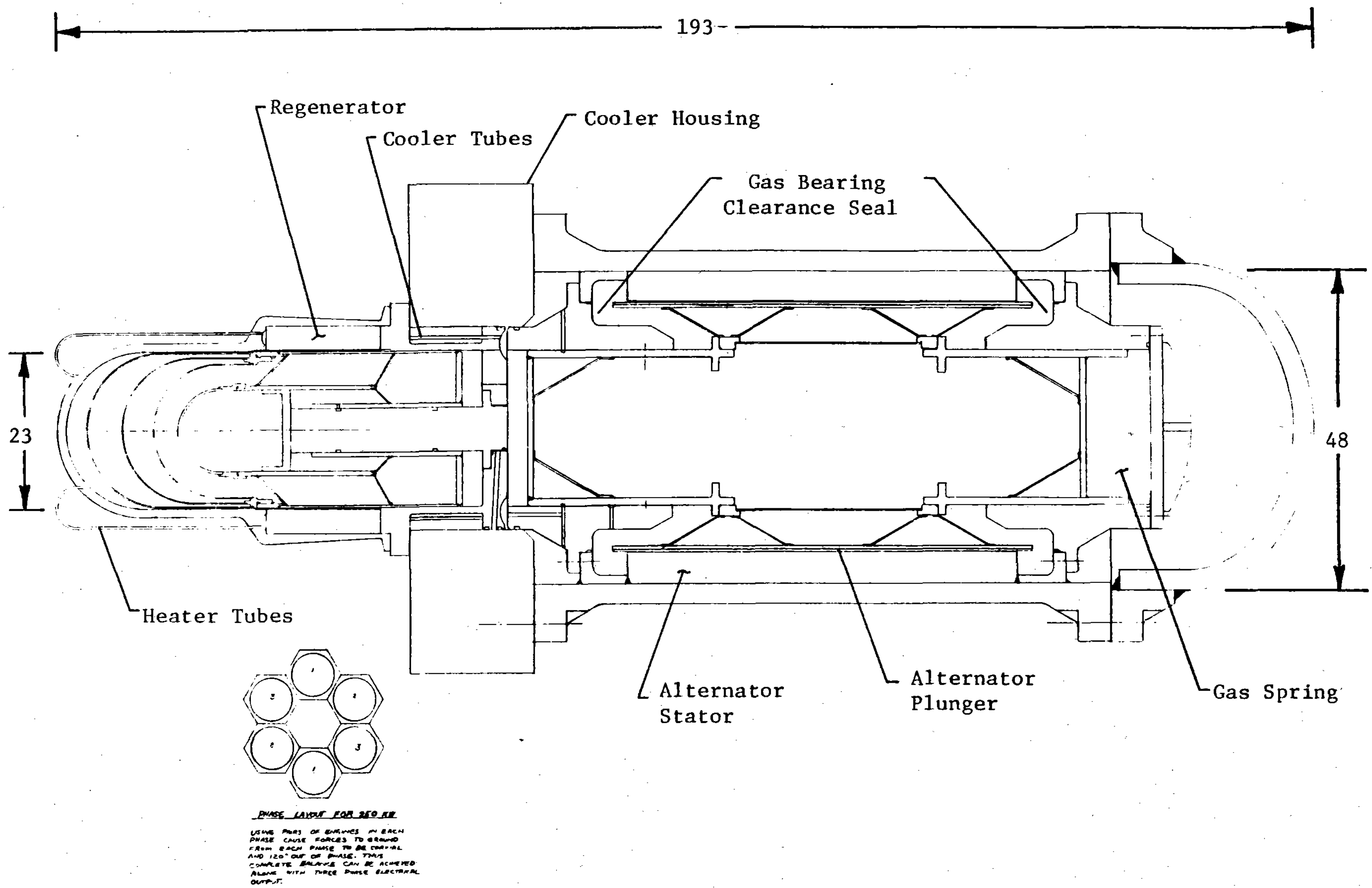

Fig. 3-13 Large-Power Free-Piston Stirling Engine Conceptual Design (dimensions in cm)

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4.0 CONCEPTUAL DESIGN OF NEAR-TERM SOLAR STIRLING ENGINE

4.1 Purpose

The purpose of this task was the development of a conceptual design of a near-term engine suitable for fabrication and operation in a solar Stirling power system in the early 1980 time period. The objective was to generate design layouts, select materials, perform required heat transfer and stress calculations, specify critical clearances and processes, and prepare control system schematics.

The design includes the following components:

- Stirling engine
- Alternator
- Auxiliary heat source
- Any auxiliary equipment required to operate engine

4.2 Requirements

Upon completion of the definition, parametric analysis and evaluation of all attractive Stirling solar engine concepts, MTI/NASA selected the single 15 kW solar engine as the concept to be developed in the conceptual design phase of this study. The three-phase requirement can be obtained from a combination of three collector-receiver-engine system, each with single phase output. However a detail analysis needs to be performed to determine the best control method for matching the phases of the engine output with those of the grid.

The selected engine-alternator base preliminary design developed in Task 1 is presented in Figure 4-1. The following requirements for the conceptual design were specified:

Engine Concept	- 15 kW single engine with alternator stator inside
Thermal Input	- Condensing sodium
Heater Head Design Temperature	- 816°C
Maximum Metal Temperature	- 871°C

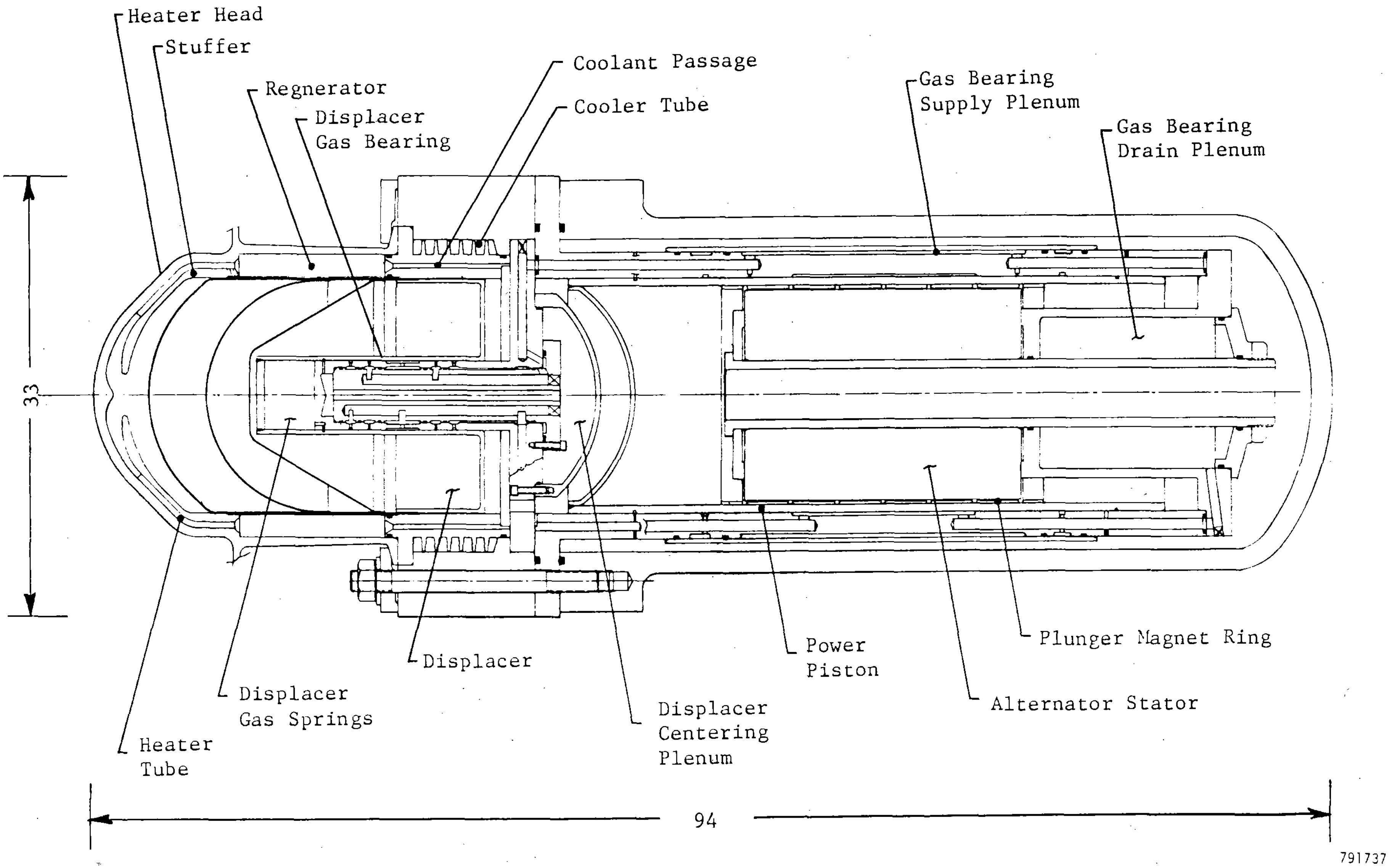

Fig. 4-1 15 kW Preliminary Conceptual Design
(Dimensions in cm)

Cooler Water Temperature	- 43.3°C
Worked Fluid	- Helium
Design Heat Input	- To be determined as part of design
Maximum Heat Input	- 10 percent above design
Heat Storage	- 70 minutes at full power with receiver
Operating Mode	- Constant power; no modulation
Power Output	- 15 kW plus parasitic power for coolant pump
Output Specifications	- 120-240 V single phase 60 Hz
Life	- 50,000 hours
Interfaces	
a) Heater Head	GE Heat Receiver
b) Electrical	Utility Grid

The conceptual design was developed based on the specified requirements.

4.3 Technical Approach

The preliminary conceptual design of a single 15 kW solar engine completed as part of Task 1 will provide the base for the development of the final conceptual design.

An assessment of the system design as identified was performed prior to initiation of the conceptual design. Results of the implementation being performed in parallel with the conceptual design provided input to establish risk and/or critical areas of design. Design alternatives were also provided that could be incorporated with cost savings with little or no performance penalty. Modifications made to the preliminary design include:

- Changes in component concepts
- Design changes necessary to reduce risks based on initial design assessment
- Refined system interface parameters and effect of close coupled heat receiver

- NASA design point and operating profile definition
- Changes in engine components to affect cost savings

The engine thermodynamic and heat transfer analysis was performed using the Sunpower second-order code in conjunction with the Sunpower system optimization code. The choice of engine dynamics was made on the basis of previous experience, using the second-order analysis and the engine stability and control requirements as a guide. The size and structure of the main heat exchanger components (heater, regenerator, and cooler) was defined with the aid of the optimization code so as to yield engine efficiency optimized at the design point.

Structural and thermal stress analyses were performed as required to evaluate life, strength and rigidity of the engine's components. Particular attention was given to the design of the bearings and seals with respect to manufacturability, life, reliability, and system losses (friction, pumping, and leakage).

The alternator design was established as part of the initial configuration definition studies. The alternator design so selected was reviewed and tuned to the engine developed in this conceptual design phase.

Once the engine itself is designed and the alternator is determined, then the system (i.e., engine/alternator) was evaluated for system transient and control evaluation. A thermal/mechanical/electrical transient model of the system was developed and used to verify stability and response to system transients.

The conceptual design of a near-term engine required a design of an auxiliary heat source to permit engine testing and checkout to be carried out independently of the solar energy input. The heat source and its controls are capable of simulating the solar energy schedule.

NASA also requested that a conceptual design of a direct heated receiver concept be developed to interface with the engine contemplated for the conceptual design. The auxiliary heat source and direct heated receiver design proceeded in parallel with the conceptual design.

4.4 Implementation Assessment Impact

The implementation assessment (part of Task 2) was conducted in parallel with the conceptual design effort in order to ensure that the results of the assessment study positively impact the design philosophy incorporated into the design layout. With this approach, design concepts evolved in such a way that:

- design aspects which appeared to be beyond a 1980 state-of-the-art status were identified along with alternatives.
- design aspects which result in a disproportionate system cost were identified and alternative approaches outlined
- design aspects which require especially complex or unproven manufacturing techniques were evaluated in terms of potential solutions or innovative alternative approaches.

The following discussion delineates the resulting evolutionary process and the effective impact upon the final conceptual design layout.

The results of the implementation assessment detailed in Section 3.0 lead to the final conceptual design layout depicted in Figure 4-2. This figure outlines an engine/alternator concept that evolved through the perturbating process of the implementation assessment study and detailed analyses. Initially, the 15 kW engine design concept included some component specifications which were evaluated as having a negative impact upon the producibility of the engine/alternator system. As a result, design changes were implemented in those areas where either the costs or reliability could be enhanced without leading to any negative impact upon the system performance. The changes that were adopted reflect more of a "fine tuning" of the conceptual design rather than radical or innovative changes. The significant design alterations impacted the regenerator and alternator specifications. The following discussion outlines those alterations.

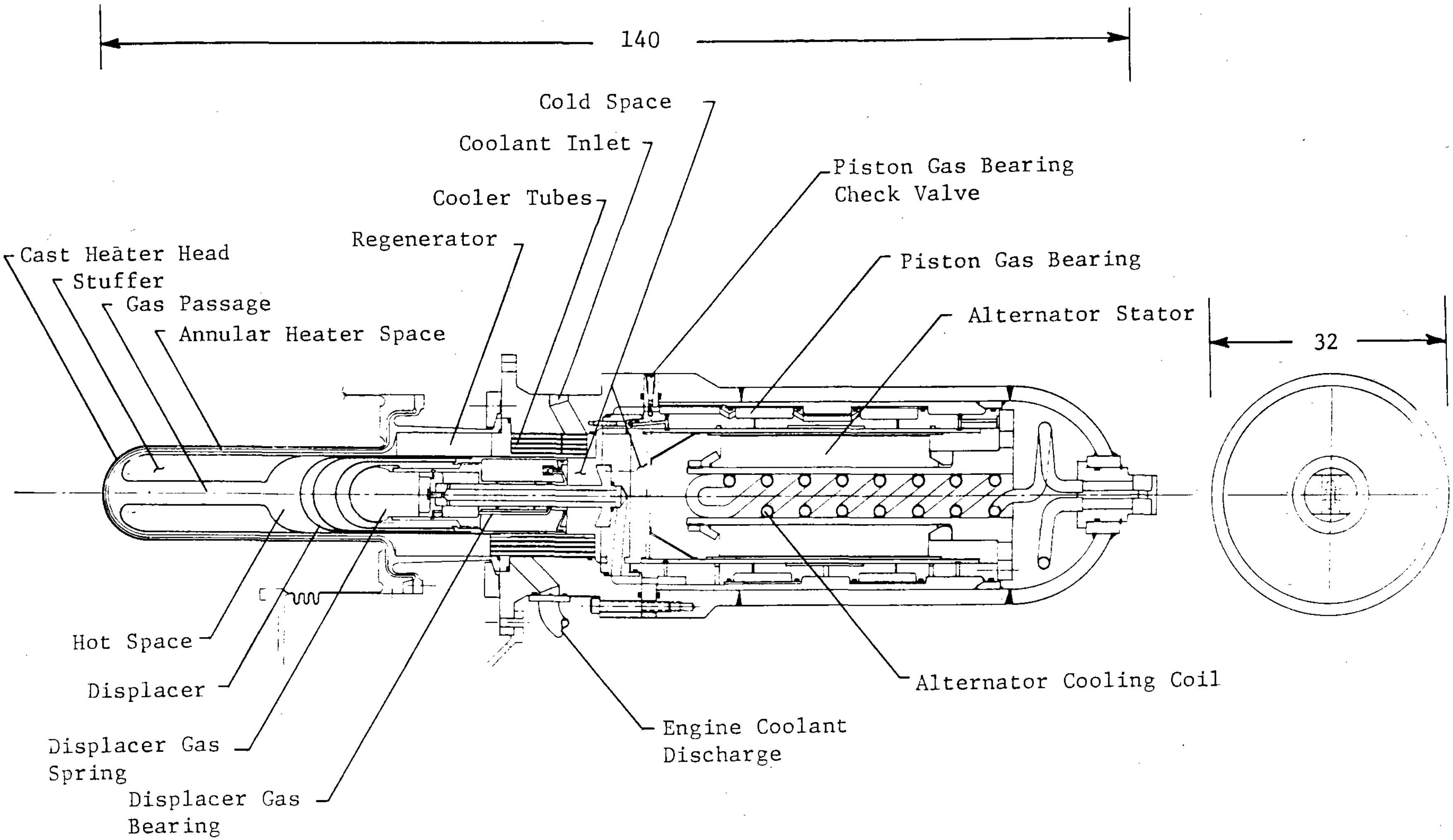

Fig. 4-2 15 kW Free-Piston Stirling Engine - Alternator Conceptual Design (Dimensions in cm)

4.4.1 Regenerator

The initial regenerator matrix design was of a fine mesh screen construction. The vendor contacts that were made established that not only was the cost of .001 in. screen expensive, but the availability of the material in large quantities is questionable. A computer model analysis showed that, with proper design optimization, a nearly equivalent regenerator performance could be achieved with a much less expensive .003 in. knitted wire matrix material. This material is included in the costing study and is reflected in the final conceptual design layout.

4.4.2 Alternator Plunger

The first approach for the permanent magnet fabrication incorporated a single-ring section concept. A preliminary producibility assessment concluded that a polygon ring consisting of permanent magnet segments would result in an equally high-performance potential with substantially reduced fabrication difficulty. The segmented magnet concept is included in the design layout and is reflected in the costing exercise.

4.4.3 Alternator Stator

Hyperco laminations were initially specified as part of the stator design. A preliminary materials evaluation showed that by substituting cold-rolled steel for the Hyperco laminations, a substantial cost savings would result, with a minimal negative impact on the design. The material substitution was incorporated at the expense of additional alternator sections.

4.4.4 Other Considerations

As a result of the implementation assessment, other cost intensive items were identified, such as: cast heater head, rare-earth permanent magnets, and material costings. The cost of the system utilizing a Mn-Al-C magnetic material is presented in Section 3.2.2.

It was determined that the alternatives identified in these areas might significantly impact the system efficiency and that further, extensive analytical efforts are required to properly identify the tradeoffs. In addition, attractive design innovations were identified in these areas. However, in some respects these innovations reflect advancements in the state-of-the-art. Therefore, they were not considered in the near term conceptual design, but rather because of the potentially high payoffs in both cost and efficiency, these concepts should be evaluated during advanced solar engine component development efforts.

4.5 System Design

4.5.1 15 kW Free-Piston Stirling Engine Conceptual Design and Description

The engine conceptual design is based on the conceptual design developed for the parametric analysis, with modification resulting from the more detailed analysis performed during the design phase of the program and the implementation assessment discussed above. The 15 kW engine shown in Figure 4-2 was selected. The one-piece heater head would be an investment cast from Inconel 713 LC, a well developed colbalt-free casting alloy with creep properties which are two to three times better than most wrought materials available. The head would be cast with a series of pins projecting from the inside surface. When finishing the casting, it will be necessary to grind only the tips of the pins, thereby reducing machining time and also wheel wear. The reduced wheel wear will make it easier to maintain size and straightness. The head shape is somewhat different than that shown in the drawings representing the configuration used for parametric analysis. It was found that although the area of the head surfaces was sufficient on both the vapor side and the engine side, the combination of wall thickness and low material conductivity in the head caused unacceptably high temperature gradients. This in turn led to unacceptable thermal stresses and an excessive temperature drop across the wall. To improve this situation, it was decided to reduce the diameter and increase the length of the head. In this manner it was possible to decrease wall thickness (due to the reduction in diameter) and increase area, while maintaining the charge pressure at 844 psi.

The hot end of the head is partially filled with a stuffer body. There is an axial passage for the working gas, 1.3 inches in diameter, from above the displacer which runs to the crown of the head and connects to an annular heater space, .120 in. wide, between the stuffer and the head casting itself. The stuffer is located by pins from the head surface allowing the working gas to flow through the space and back to the regenerator. The annular regenerator is intended to be packed with a knitted wire material using a wire diameter of .003 in. The regenerator surrounds a portion of the displacer and is followed by a fabricated tubular cooler. The cooler was switched from a machined to a fabricated structure in order to reduce the wall conduction loss. There are 200 tubes of .138 in. inside diameter with a wall thickness of .025 in., 4.00 in. long, arranged in radial rows of four. The tubes are straight and parallel to the engine axis, exiting into a gallery which connects to both the cold space below the displacer and the compression space above the power piston.

The displacer is 4.00 inches in diameter, and has a stroke of 3.0 inches. It is of the "posted" type, carried on gas bearings for high reliability and low wear. The gas bearings are fed from the engine pressure wave (164 psi amplitude) which charges a plenum through one or more check valves. The bearing drains discharge into a plenum in the pressure vessel surrounding the cylinder. This plenum is also center ported to the working space, the piston gas spring and the displacer gas spring, allowing all spaces to operate at the same average pressure.

The cylinder for the power piston is 7.00 inches inside diameter, with an overall length of approximately 19.5 inches and is made from low carbon steel. The inside surface forms a two-plane gas bearing which supports the power piston. The bearing is supplied by the engine pressure wave charging a plenum through one or more check valves which can be replaced without dismantling the engine-alternator.

The bearing surfaces are coated with chrome oxide and ground to a highly accurate finish diameter. The relatively large diameter in relation to length will permit both bearing surfaces to be finished at the same time, eliminating all alignment problems. The low carbon steel piston is also coated with chrome oxide and ground in one setup guaranteeing concentricity and alignment. Because the alternator stator is inside the piston, the finish grind of the exterior may be done either before or after the magnets are assembled to the piston.

The stator for the alternator is made of tapered cold rolled steel laminations clamped to a center tie bolt. Because the coil slots open to the outside, it should be possible to wind the coils and achieve a very good fill factor. Although the alternator is very efficient, the energy to be dissipated exceeds 1 kW due to the high power level. Consequently, a cooling coil was brazed to the interior of the tie bolt, providing a means of stabilizing the temperature. A diverter fitting routes part of the engine cooler discharge water through the alternator. The cooling lines enter and exit through a plug in the end of the pressure vessel. The same plug would contain feed throughs for the alternator electrical output.

The stator assembly is cantilevered from the end of the cylinder, and would be subassembled using shims or a centering ring for alignment. The piston would then form an excellent guide during the insertion of the piston preventing the permanent magnet rings on the piston from contacting the stator. The altenator and cylinder would then be installed in the pressure vessel, and then installed on the engine as a unit. The pressure vessel would be manufactured from carbon steel, designed to satisfy ASME codes as an unfired pressure vessel.

4.6 Component System Design

4.6.1 Engine

Upon completion of Task 1 and review and evaluation of parametric analyses, the engine/alternator/system concept selected to be carried into the conceptual design phase of the study was a single 15 kW engine. The requirements for this engine are presented in Subsection 4.3. The conceptual design was developed from the initial layout developed in Task 1 and presented in Section 2.0 and the parallel implementation study presented in Section 3.0. The following major component areas of the engine/alternator were reviewed and analyzed to develop the conceptual design presented in Subsection 4.5. The analyses to support the conceptual design are presented in this section.

4.6.1.1 Heater Head Design. As previously stated, low system cost is probably the key to wide acceptance of a solar power generating system. Therefore in the program's early stage efforts concentrated on eliminating the costly heater tube construction and replacing it with a heater head manufactured by casting. Heat transfer analysis indicated that the limited heat transfer area available with casting heater head geometries, such as a hemisphere above the displacer cylinder, results in a large heat flux which creates a large temperature difference across the heater head wall. This large temperature difference not only causes unacceptable thermal stresses but also decreases the working gas temperature for the same allowable metal temperature and thus decreases the engine cycle efficiency.

To reduce the temperature difference of a cast heater head, the design was changed to a smaller diameter displacer and heater head cylinder which results in a thinner wall thickness for the same engine charge pressure and hence less less temperature difference across the wall. Therefore, a relationship was derived of the heat transfer capability of a cylinder as a function of its physical dimensions. The results show that the necessary heat transfer surface area and desired heater head temperature drop are only dependent on the length and diameter of the heater head. In order to maintain no greater than 44.5°C heater head temperature drop it was determined that with a reasonable displacer cylinder diameter of 10.16 cm, the length must be 37.64 cm.

4.6.1.2 Cooler Design. The engine configuration was optimized by the computer program. The computer optimization was based on the gas side thermodynamic cycle and does not take into consideration the heat transfer characteristics of the outer surface of the heater head or cooler. To ensure that adequate heat is removed, the number of tubes, tube length and tube wall temperature of the cooler tubes were specified as inputs to the program and the computer selects the optimum tube ID. In the process of designing the cooler, gas side information was also provided by the computer as follows:

Heat to be removed = 23,260 watts
Tube ID = .349 cm

Average gas temperature in the cooler for the complete cycle: 74°C

Temperature difference between the cooler tube wall at the ID and the main gas stream: 28°C

Given the heat to be removed, heater tube heat transfer area, and the temperature gradient the gas heat transfer coefficient was calculated. The mass flow at the interface between compression space and cooler was calculated by the computer as .00275 kg/half-cycle. Assuming this mass flow, one specific heat at constant pressure for Helium, and heat to be removed, the inlet temperature and outlet temperature of the gas at the cooler were determined as 182.4°F and 147.6°F, respectively.

Three-Tube Pass Split Flow Heat Exchanger:

A three-tube pass split flow heat exchanger was chosen as pictorially shown in the conceptual design in Figure 4-2.

The summary of the cooler design established by calculations is as follows:

Cooling water inlet temperature:	43°C
Cooling water outlet temperature:	49°C
Cooling water flow:	56.8 liters/min
Pumping power:	.179 kW

4.6.1.3 Bearing Design. Ideally, the bearings used in the Stirling engines would be operated without any lubricant. This is because any foreign material, such as lubricating oil or Rulon particles, deposited on the heater, regenerator or cooler heat transfer surface will reduce engine performance. A gas bearing using gas as a lubricant is free from any potential failure mechanism. It consumes less power due to its low mechanical friction as compared with any other type of bearings and results in no wear of mating parts. It does require, however, a pressure source to support the gas flow and this pressure has to start before the engine is turned on to avoid rubbing during start-up and/or shutdown or have material compatability to eliminate wear.

It also requires close tolerance in manufacturing. Comparing the few disadvantages and requirements of an efficient, reliable and durable engine for the solar system, it was decided to use gas bearings for the engine. MTI has successfully designed several free-piston Stirling engines that utilize gas bearings. To facilitate the gas bearing design, a manual published by MTI was used to select the dimensions and operating conditions for a gas bearing which meets a given set of specific requirements. The data and charts in the manual were obtained by computerized analysis. Experience indicates general validity of this analysis. All the gas bearings of the 15 kW engine were designed, based on this manual.

The results of the gas bearing designs are listed as follows:

Piston Gas Bearing Design

Given Conditions

Diameter	7 in.
Overall bearing length available	11 in.
Load:	
Piston and stator weight	34 lb
Alternator side pull	160 lb/mil
Initial offset	.001 in.

Result of Analysis

Supply pressure	928 psi (10 percent over charge pressure)
Radial clearance	.00075 in.
Number of admission plane	2
Bearing length for each admission plane	3 1/2 in.
Number of feeder holes	22/admission plane
Orifice diameter	.037 in. (#63 drill)
Gas flow	.0059166 lb/sec total
Radial stiffness	571700 lb/in.

Displacer Rod Gas Bearing Design

Given Conditions:

Displacer rod diameter	1.114 in.
Overall bearing length available	3.6 in.
Displacer dead weight	3.26 lb
CG offset from CL Bearing (longitudinal)	1.8 in.

Results of Analysis:

Supply pressure	928 psi (10% over charge pres)
Radial clearance	.00075 in.
Number of admission plane	2
Bearing length of each admission plane	1.114 in.
Number of feeder holes	10/admission plane
Orifice diameter	.040 in.(#60 drill)
Gas flow	.002955 lb/sec
Radial stiffness	37064 lb/in.

Gas Supply Pump Elimination

As previously stated, a gas bearing requires a pressure source to supply the gas which flows through the bearing at a pressure higher than the mean engine pressure. This can be accomplished by a separate pump powered by the engine outside the sealed engine system. This pump adds cost and complexity to the system. To simplify the design and to reduce the system cost, MTI has developed a gas bearing system in which the pump is eliminated. It takes advantage of the pressure fluctuation of the engine pressure or its gas-spring pressure during the cycle. A separate gas reservoir is charged by the engine or gas spring pressure during part of the cycle when its pressure is higher than the designed gas-spring supply pressure. During the remainder of the cycle, the gas in the reservoir is discharged to the bearings and the pressure drop is not allowed to fall below the design level by properly selecting the reservoir volume. The flow back of gas at the charging area is prevented by using a check valve or appropriate porting.

4.6.1.4 Engine Optimization Analysis. The optimization of the engine configuration was accomplished by utilizing the free-piston Stirling engine optimization computer program developed by Sunpower Incorporated. The inputs to the program were as follows:

Engine design point power output:	16.5 kW
Metal temperature at design point:*	1500°F
Cooling water temperature:**	110°F
Material for the heater head:	Inconel 713 LC
Material properties (see Subsection 2.7)	
Piston diameter: (determined by alternator design)	7 in.
Piston amplitude: (determined by alternate design)	.6689 in.
Piston and alternator stator mass: (determined by alternator design)	34 lb
Engine operating frequency: (fixed by alternator)	60 Hz
Displacer diameter: (determined by heat transfer analysis)	4 in.
Displacer weight (determined by displacer geometry)	3.25 lb

Cooler: 200 tubes four inches in length were selected based on waterside heat dissipation and the computer optimization of the tube ID.

Heater: The computer program of Sunpower Incorporated for engine optimization was developed to handle the heating of tubular configuration. The simulation of an annular heater flow area by this program requires modification of the flow equations. Lacking time and funds to do

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* Heat wall temperature at gas side: 1420°F
(80°F temperature drop as shown in Subsection 4.6.1.1)

**Cooler wall temperature at gas side: 117.5°F
(based on a primary cooler design analysis)

this, it was approximated by using the hydraulic diameter of the annular section as a substitute for the tube diameter. The hydraulic diameter of an annular section of 0.1 in. gap on a 4 in. diameter, ID is 0.2 in.

Working gas: Helium

Results of the optimization analysis for the conceptual design are presented in Table 4-1.

4.6.1.4.1 Engine Efficiency. The overall thermal efficiency of this stepped engine (displacer diameter 4 in.; piston diameter 7 in.) is 41.5 percent (engine only). The Carnot cycle efficiency is 64.44 percent.

4.6.1.5 Engine Heat Balance. The heat balance is shown in Figure 4-3.

4.6.2 Permanent Magnet Linear Alternator

The decision to proceed with a permanent magnet linear alternator with the inside stator was discussed in Section 2.6. The preliminary conceptual design of the alternator was also discussed in Section 2.6. This section discusses the selection of the final design parameters for the final conceptual design of the linear alternator to match with the free-piston Stirling engine design.

4.6.2.1 Determination of Number of Sections and the Thickness of the Magnet.

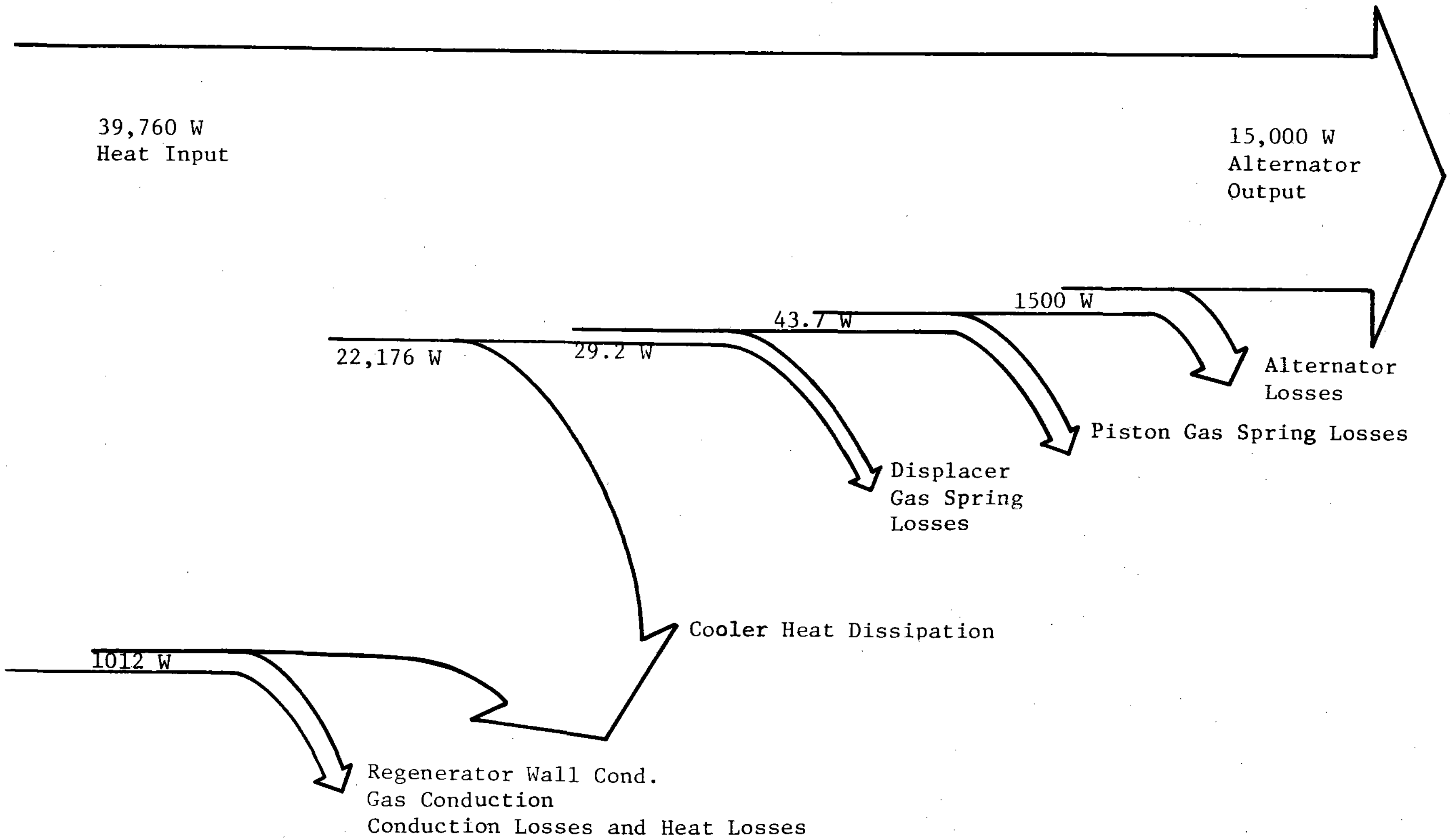
The amplitude of maximum stroke, ℓ", and the diameter, D", has been determined by the Stirling engine considerations. Thus the required power output can be obtained by a suitable combination of the number of sections and the thickness of the magnet. Increasing the thickness of the magnet results in an almost linear increase in power of each section up to a certain point. This increase results from the reduction of the airgap reactance as the magnet permeability is close to that of air.

From the power equation presented in Section 2.6.4 it can be seen that the magnet weight per unit power is more or less constant. Thus when a thicker magnet is used, fewer sections are needed for a given power. However, thicker magnet results in poorer efficiencies. This is clearly shown in Figure 4-4.

TABLE 4-1

15 kW FREE-PISTON SOLAR STIRLING ENGINE CONCEPTUAL DESIGN PARAMETERS

Engine power output	16,500 watts
Charge pressure	58.2 bars
Displacer amplitude	3.828 cm
Displacer phase angle	36.1°
Displacer rod diameter	2.83 cm
Cooler tube ID	3.49 mm
Cooler pumping loss	247 watts
Average cooler delta T	27.8°K
Regenerator length	12.7 cm
Regenerator void volume	1308 cm^3
Regenerator effectiveness	.99
Regenerator wire diameter	1.17×10^{-3} cm (.003)"
Heater tube number	42
Average heater delta T	64°C
Heat from heater to gas	39,760 watts
Overall engine (before alternator) thermal efficiency	41.5
Piston external spring	3.1×10^5 N/m^2
Displacer external spring	21×10^5 N/m^2
Cooler NTU	.52
Heater NTU	.76
Conduction and heat losses	1012 watt



Engine Heat Balance

Fig. 4-3 15 kW Solar Stirling Engine

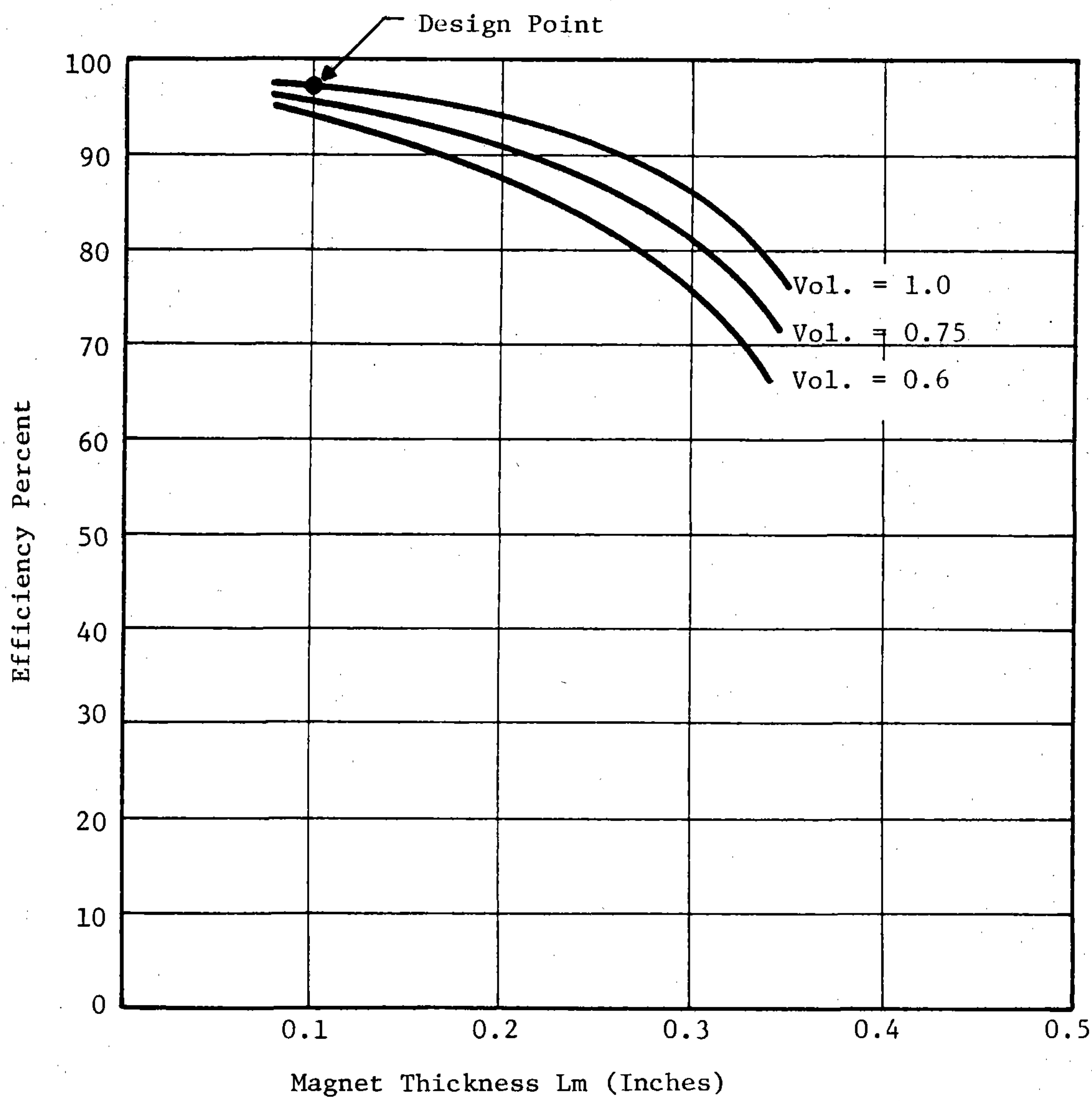


Fig. 4-4. Effect of Magnet Thickness on Efficiency at Constant Magnet Volumes

4.6.2.3 Design Procedure

Various parameters of design affect the design in different ways. By a suitable choice of these parameters, an optimum alternator could be designed. However, it is important to define what is meant by "optimum". For example, if efficiency is to be maximized without regard to any other constraint, the alternator will be prohibitively massive. Alternatively, by increasing the thickness of the magnet and the ratio V/E, very high specific power output levels could be obtained. As previously indicated, such a design would result in very poor efficiency. Again, it is possible to improve the efficiency of such a high-power density machine by operating it at a derated level. It may appear at first glance that there may be nothing to gain by such a choice, whatever weight reduction is achieved by increasing the magnet thickness is being lost completely due to operating at derated levels. In order to get a feel for such variables, several designs were made by treating the thickness of the magnet (lm"), the ratio of slot opening to full stroke length (S_o/ℓ), and the ratio of applied voltage (V/E) as independent design parameters. The range of these variables were as follows:

ℓm"	0.1 to 0.3 in steps of 0.05
S_o/ℓ	0.15 to 0.3 in steps of 0.05
V/E	1.05 to 1.25 in steps of 0.05

The power output per section was computed by using the power equation given in Section 2.6. From the power output per section, a number of sections were determined to give the required 15 kW output. As the number of sections can only be an integral value, actual total power developed in all the sections will be a certain percentage higher than the desired 15 kW, depending on the number of sections. When there are large numbers of sections, the design can be better matched with the required output. Such is the case when the design calls for thin magnet and low values of V/E which result in low power output per section.

Figure 4-5 defines various dimensions of stator that have to be designed. The airgap diameter D" and the length of each section (two times ℓ") is determined by the engine considerations. The inside bore D1" is determined by the structural considerations. This can be seen from a look at the overall system

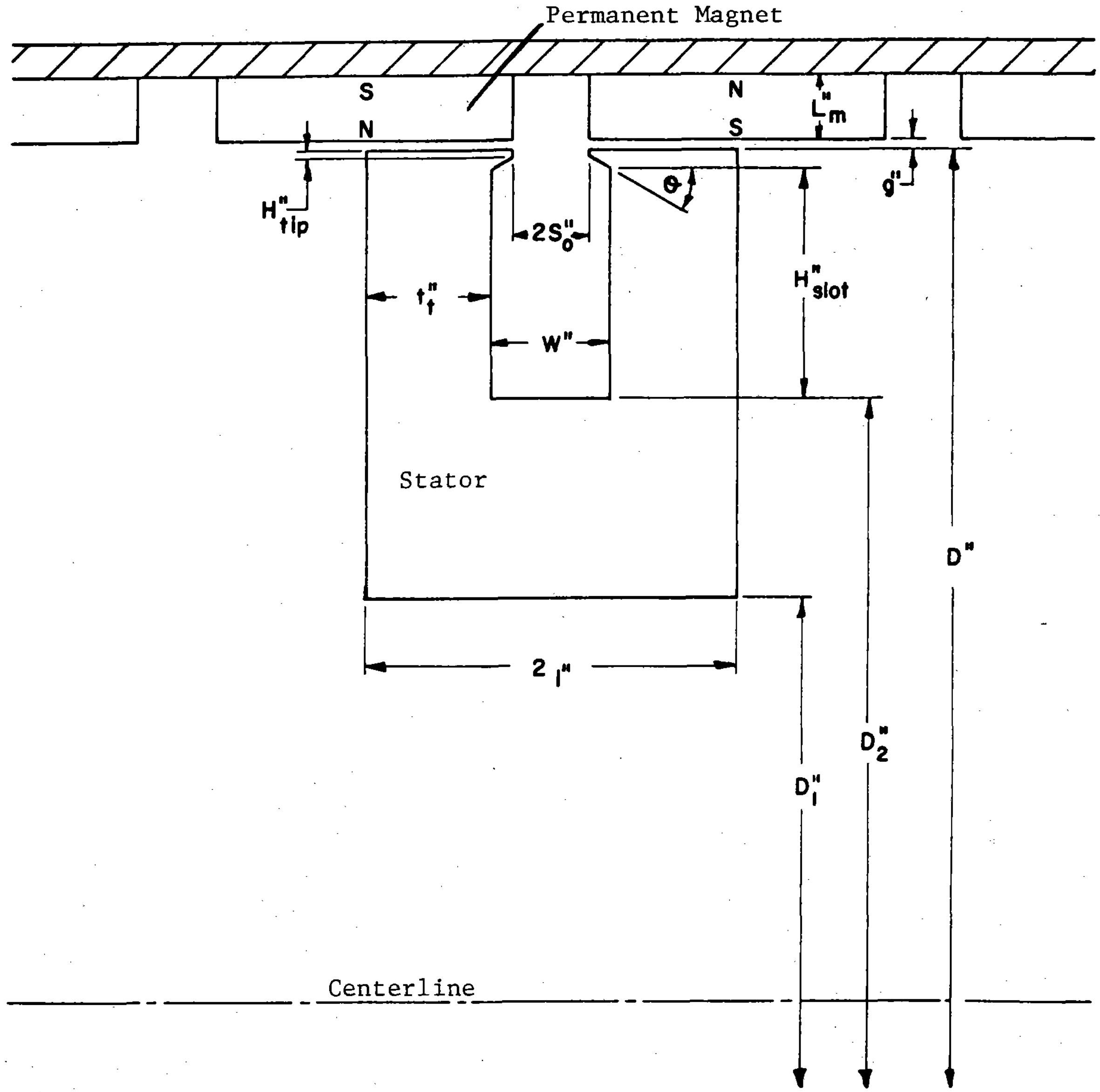


Fig. 4-5 Nomenclature for Permanent Magnet Linear Alternator

drawing (Figure 4-2) in which the stator is shown to be cantilevered from one end. As the stator is to be made up of laminations, structural strength is provided by a sleeve running through the center of the stator. For limiting the deflection then, the sleeve has to have certain section moduli which determine the minimum value of D1". The magnetic flux goes around the back of the slot to keep the flux density within limits; this path should have a certain area. Thus setting a minimum value for D1" results in a certain maximum value for $H_{s/t}$" the depth of the slot. The leg thickness t_t" is determined by the total flux going through it and the choking area which occurs at the radius which corresponds to the back of the slot. The total flux in the leg is composed of the airgap flux and the slot leakage flux. The airgap flux is determined by the magnet residual flux density Br, the ratio S_o/ℓ, and the ratio V/E. The slot leakage flux can be calculated from techniques similar to those used in conventional rotary machines with suitable modification to account for the cylindrical nature of the slot, which results in a smaller area for the flux at the slot back then at the tip.

The slot tip height h" is set at 0.02" to facilitate manufacturing of the laminations. The slot tip angle θ is determined from the following considerations:

To reduce the stator leakage reactance, the slot width should be as large as possible over the entire slot height. As a matter of fact, it is more important at the tip than at the back due to the fact that all the ampere turns in the slot appear across the slot tip and a continually diminishing fraction appears across the slot width towards a deeper portion of the slot. However, it is not possible to keep the same slot width all the way to the airgap since this would result in excessive S_o/ℓ, causing poor specific power. Thus the slot width should be made as quickly as possible from its value at the slot opening to the full width W" as one moves away from the airgap. Ideally this would imply $\theta=0$; but this would result in an excessive flux density in the tooth tip. The value of θ is then determined to be the smallest possible angle which would prevent the flux densities in the tooth tip from reaching excessive values.

Having thus determined the dimensions of the magnetic circuit completely, the coil is designed to its number of turns and the gauge.

The number of turns for any given gauge wire can be easily computed from the geometrical considerations as the slot geometry is already defined. Adequate allowance is made for the slot linear insulation and the insulation on each wire. The wires are then assumed to lie next to each other and the top of each other. The fact that the number of turns per layer and the number of layers that can be laid in the slot can only be integers is properly considered. The value of the induced voltage per section is then computed. Using this procedure, a table of wire gauge number and the induced voltage is computed. The choice of gauge number will be that which corresponds most closely to the system terminal voltage by a suitable combination of series/parallel connections of various sections.

The resistance, reactance, and copper loss at rated current, etc. are then computed for the selected combination of connections and the wire gauge.

4.6.2.4 A Sample Design for 6.3 Inch Diameter Inside Stator Arrangement

Using the previously mentioned procedure, a design has been selected which approximately represents engineering judgements on the various tradeoffs of efficiency, weight and overall system economics. The details of this design are as follows: (Nomenclature is presented in Figure 4-5 and design represented in Figure 4-6).

Stator

D"	= 6.3
D1"	= 3.0
D2"	= 4.459
1"	= 1.375/2
H"	= 0.02
θ	= 15°
W	= 0.425
t_t"	= 0.950
Hslt"	= 0.875

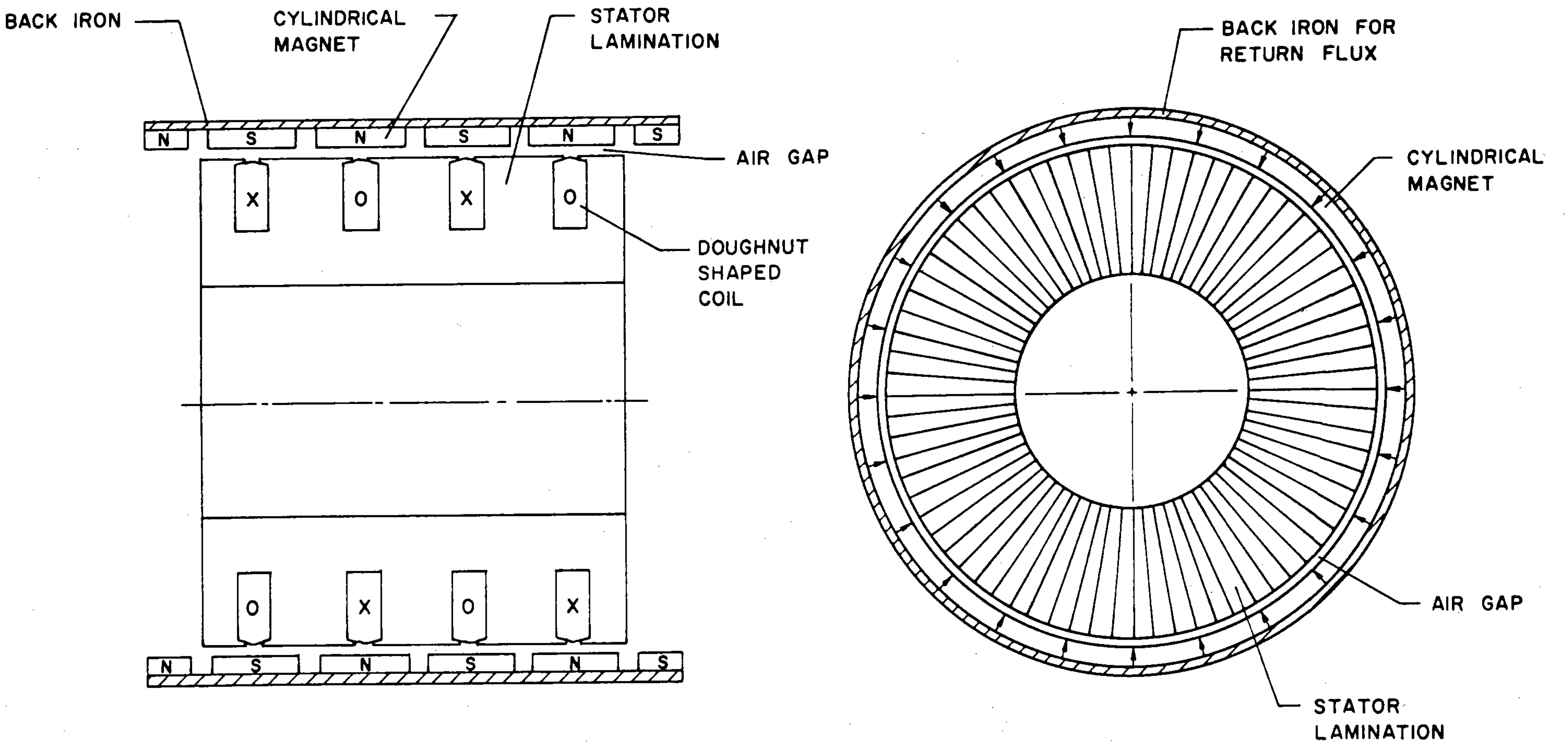


Fig. 4-6 Permanent Magnet Linear Alternator Design

Voltage	= 240
S_o/ℓ	= 0.2
Number of turns in each slot	= 20
Wire gauge number	= 9
Number of sections	= 8
Weight of copper	= 8.74 lb
Weight of stator iron	= 82 lb
Resistance @ 20°C/section	= .02165Ω
Reactance/section	= .385Ω
Total copper loss	= 654 watts
Total iron loss	= 654 watts (assumed)
Total power output	= 14.7 kW
Efficiency	= 91.8 percent

Plunger

Radial airgap	= 0.010"
Magnet material	= $SmCO_5$
Magnet thickness	= 0.1"

4.7 System Control and Stability

4.7.1 System Stability

A critical question associated with the proposed engine/alternator design concerns the stability of the system in the grid connected mode. There are two main aspects to this question: first, the basic stability of the system connected to a steady harmonic voltage source; and second, the stability and transient response of the system connected to a finite grid with many other similar systems. The former aspect, referred to as simple grid stability, has been analyzed as a subtask of the present design effort. The results of this analysis are encouraging. These results indicate that the proposed engine alternator can indicate stabilized behavior by partially tuning the alternator output with a series capacitor. A description of the mathematical model used in the stability analysis, along with a description of the analytic solution technique and a summary of the analytical results are presented in the following subsections.

4.7.1.1 Mathematical Stability Model. The engine, alternator and grid form a complex nonlinear mechanical electrical system. Experience to date with similar systems operating in an isolated mode, i.e., feeding a resistive load bank, indicate that the predominant nonlinearities tend to stabilize the system. As a consequence, in the present analysis a simpler linear lumped parametric system was used to represent the engine alternator. This system is shown schematically in Figure 4-7. The system model has three degrees of freedom: two mechanical (the engine displacer and alternator plunger) and one electrical (the alternator current). The lumped parametric model of the mechanical dynamics is represented by two-spring constants, K_D and K_P, and two damping coefficients, C_D and C_P. The plunger damping coefficient contains the gas spring leakage and hysteretic loss, as well as the gas bearing pumping loss. The thermodynamic aspects of the engine are contained in two linear lumped parameter models denoted as P_e and C_e. The first model accounts for the forces exerted on the displacer and plunger due to the main engine pressure wave and the second model accounts for the forces due to the heat exchanger pressure drop. The alternator model consists of a voltage source E and a series connected inductance, resistance, and capacitance. The voltage source models the voltage induced in the alternator coil by the motion of the plunger. This voltage is proportional to the instantaneous plunger velocity. The resistance element models both the copper (I^2R) and iron (hysteresis and eddy current) losses of the alternator. The inductor models the alternator coil reactance, and the condenser models the external tuning capacitor. The reaction force on the plunger is assumed to be proportional to the instantaneous alternator current. The grid is represented by an ideal 60 Hertz voltage source whose voltage and frequency are independent of the alternator current.

4.7.1.2 Characteristics of the Stability Model. The system shown in Figure 4-7 is a lumped parameter linear system with a harmonic forcing function. The overall system response is the sum of a transient response and a steady-state response. Steady-state response is determined by the forcing function amplitude and frequency. Transient response is independent of the forcing function, but dependent on the system's initial condition. If the transient term for all allowable initial conditions decays with time, then the system is stable. If the transient term grows with time, then the system is unstable.

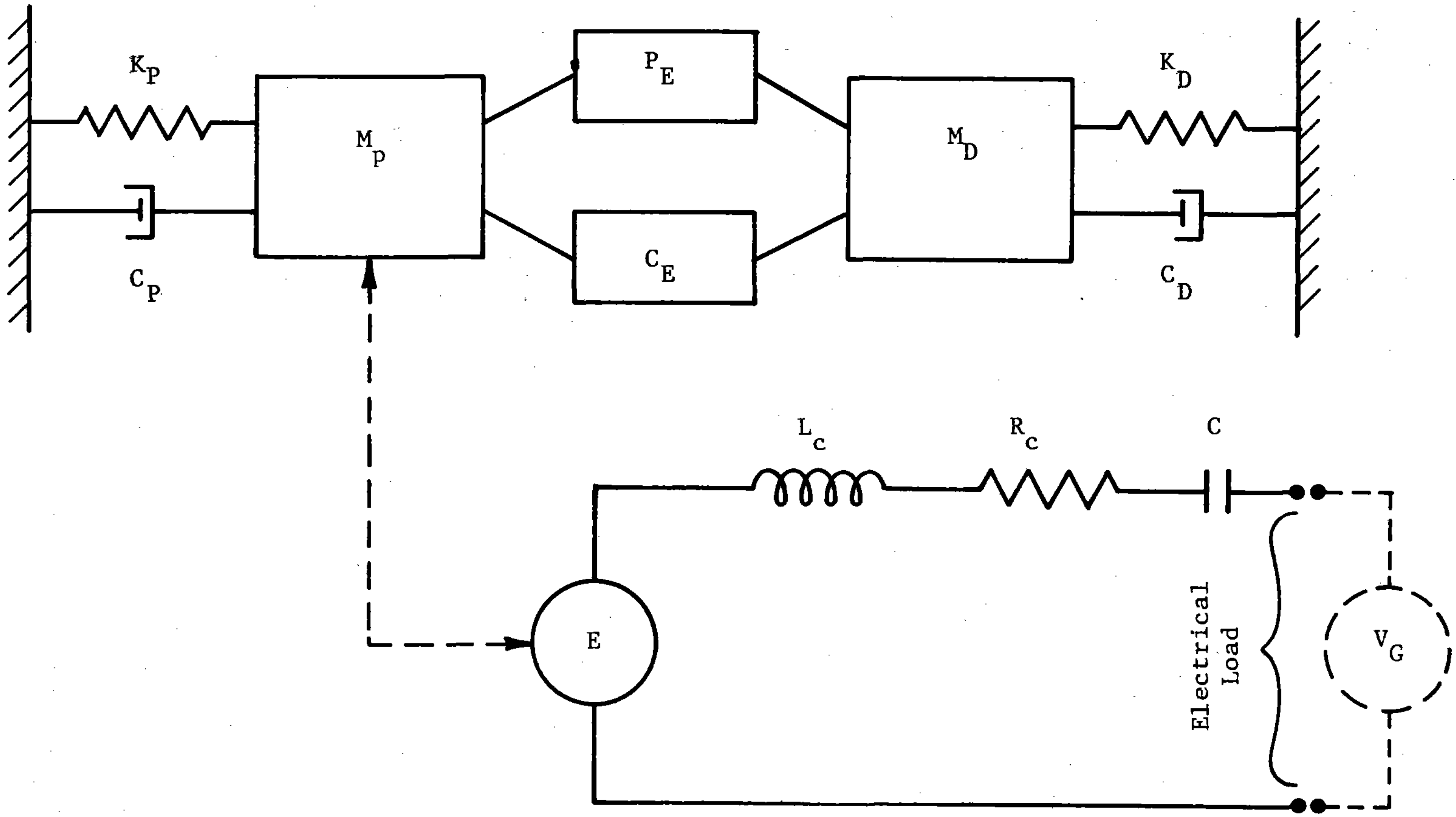


Fig. 4-7 Lumped Parameter Stability and Control Model

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Transient term stability is determined by the eigenvalues of the corresponding homogeneous system, (the system in which the voltage source V_{grid} is replaced by a short circuit). If the homogeneous system is stable, i.e., the real part of all three eigenvalues is negative, then the forced system will be stable. Rephrased, neglecting nonlinearities, the stability of a single engine alternator connected to an ideal grid is determined by the stability of the system when it is isolated from the grid and the alternator output terminals are shorted.

From a stability viewpoint, the only dissipative, or power absorbing element, in the alternator circuit of the homogeneous system is the coal resistance R_c. (A homogeneous system is a system with the grid voltage source replaced by a short circuit.) As a consequence it can be anticipated that the system stability will be a function of the value of R_c. In particular, if R_c is zero (no dissipative element), or infinity (open circuited alternator), then the system is unstable. Consequently, the general stability dependence on coal resistance is illustrated in Figure 4-8. The lowest coil resistance for which the system is stable is denoted on R_{crit_1}. The highest coil resistance is devoted as R_{crit_2}. The coil resistance corresponding to maximum stability is denoted as R_{max}. If the coil resistance is less than R_{crit_1}, or greater than R_{crit_2}, the system is unstable. If the coil resistance is between R_{crit_1} and R_{crit_2} then the system is stable. One potential method for improving the stability of the engine alternator system is to add a resistance in series with the alternator and the grid. As long as the overall resistance is less than R_{max}, the larger the external resistance the greater the stability. Increasing the overall resistance beyond R_{max} reduces the stability of the system. This method for achieving stability is obviously undesirable since it lowers overall system efficiency. Fortunately it has not proved necessary to utilize dissipative stabilization in the present design.

4.7.1.3 Analysis Technique. As previously mentioned, the stability of the harmonically forced system is insured if the real part of the eigenvalues of the corresponding homogenous system are negative. In the present stability analysis, the real part of the dominant stability eigenvalue, i.e., the one with the most positive real part, was evaluated using a time-stepping transient analysis code.

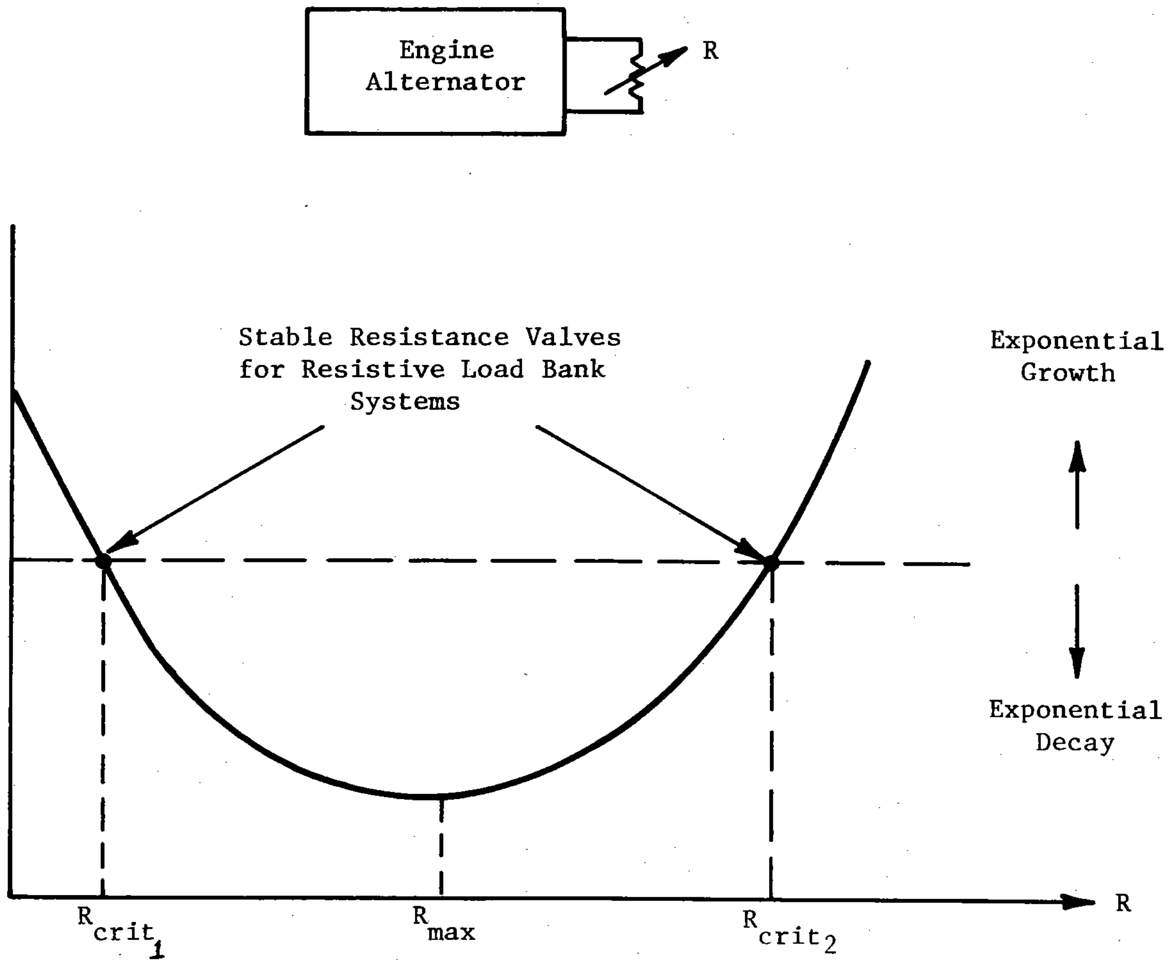

Fig. 4-8 Effect of Overall Altenator Resistance on Stability

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In this approach, the system is given an arbitrary initial deflection and the corresponding system transient allowed to develop until the transient contributions of the two subdominant eigenvalues have become negligible. For the present system, this takes between 30 and 120 cycles (1 to 2 seconds actual transient time) depending on the system parameters. The dominant eigenvalues are determined from the frequency of oscillation and the rate of growth or decay of the remaining transient. This is not the most elegant method of evaluating system stability but it was the most convenient since, with a few modifications, a fast, easy- to-use time stepping code was available. In order to facilitate the evaluation of the numerical data, only the piston amplitude at the end of each cycle and the corresponding frequency of oscillation was utilized from the printout.

4.7.1.4 Results. The initial stability analysis was performed with a perfectly tuned alternator output (C = 546 μf). The effective coil resistance was .243 ohms corresponding to a design point alternator efficiency of 92 percent. The system was unstable with a dominant eigenvalue real time constant of 0.17 seconds and an oscillatory frequency of 51.8 Hertz.

Given that the system is unstable at the tuned operating point, the simplest means of improving the system stability is to add resistive dissipation or change the value of the tuning capacitor. The former is undesirable since it reduces the performance of the system. The effect of the latter is that the system is stable for external capacitor values between 470 and 280 μf, and maximum stability is achieved at approximately 400 μf. Above 400 μf, the long-term transient is dominated by an eigenvector having a frequency of approximately 50 Hertz. Below 400 μf, the transient response is dominated by an eigenvector with a frequency of approximately 65 Hertz. At 400 μf, the decay of both eigenvectors is the same and the decay transient has a beat frequency of 15 Hertz.

4.7.1.5 Conclusions. The present engine alternator design can be stabilized for simple grid operation by proper choice of the tuning capacitance value. The optimum capacitor value is 400 μf. The value of capacitance can be varied between 280 μf and 470 μf before the system becomes unstable.

4.7.2 Engine/Alternator Control

4.7.2.1 Operating Modes. There are two basic operating modes for the solar alternator system as shown in Figure 4-9 . Mode I corresponds to normal operation with the system connected to the grid. Mode II corresponds to the shutdown configuration with a damping resistance across the alternator output terminals. The basic configurational differences between the modes are listed below:

Mode I Normal Operation

- Thermal value 'on' to allow engine to draw energy from the solar receiver
- Control system activated to maintain system operating point
- Heat rejection system fully activated
- Alternator output connected to grid

Mode II Shutdown

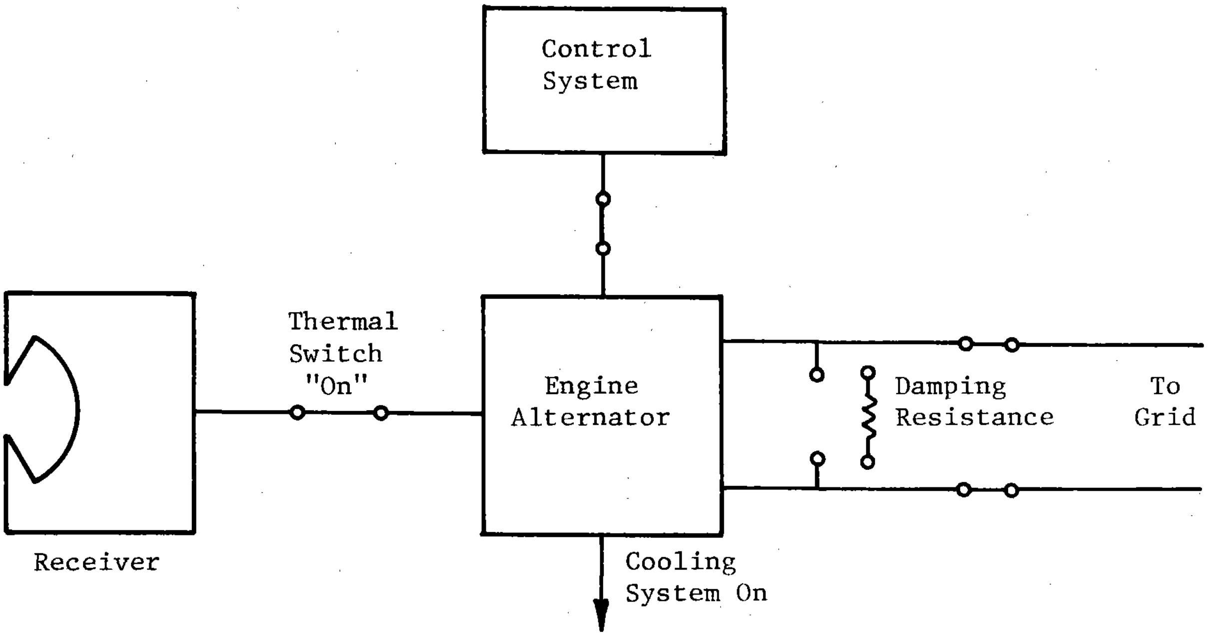
- Thermal value "off" to prevent excessive heat loss from solar receiver to engine coolant
- Control system deactivated
- Heat rejection system on standby with sufficient cooling to prevent overheating of engine cold side components
- Alternator output connected to damping resistance to prevent operation of the engine/alternator system.

Mode I Controls

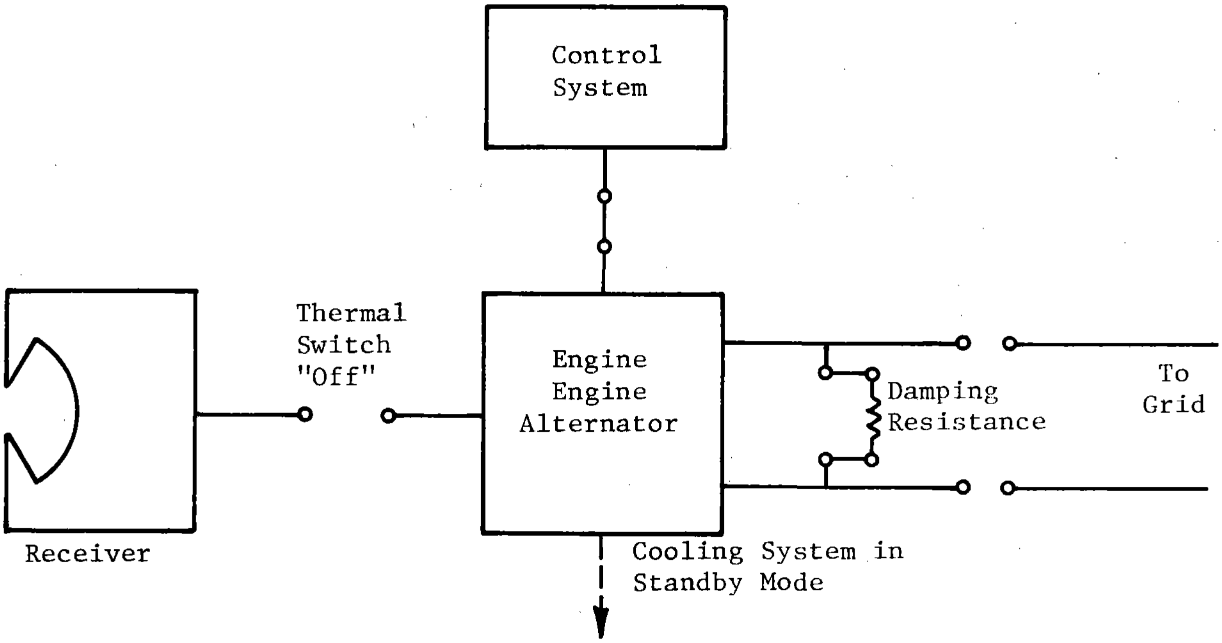
The basic requirement of the Mode I control system is to maintain an essentially constant system power output independent of changes in the system's environment. The "environment" relative to the engine alternator system consists of four main parameters as follows:

- Heater Head (Receiver) Temperature
- Cooling Water Temperature
- Grid Voltage
- Grid Frequency

In the following analysis, changes in only the first and third have been considered, since, to a first order approximation, a 1°C increase in cooling



Mode I Normal Operation



Mode II Standby

Fig. 4-9 Solar System Operational Modes

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water temperature is equivalent to a 4°C decrease in heater head temperature; and, in general, grid frequency values are held fairly tight.

The conveniently available control parameters are:

- Engine Pressure
- Piston Gas Spring Rate
- Displacer Gas Spring Rate
- Tuning Capacitance (within the range consistant with engine/ alternator stability)

4.7.2.2 System Transient Response. The two major system operating transients occur in response to conversion from Mode II to Mode I (shutdown to grid connection) or Mode I to Mode II (grid connection to shutdown). The response of the system to the first response is shown in Figure 4-10 and 4-11. The response of the system to the second transient is shown in Figure 4-12. As can be seen the transients decay and stable operation is achieved in approximately 90 cycles (1.5 seconds). The initial region of the transient in Figure 4-10 exceeds the overstroke limits of the engine (2.1 cm) and as a consequence some mechanism must be provided to prevent engine pounding. In the present design this is done by providing end-of-stroke dampers on both the piston and displacer. The transient response of this nonlinear system has not been evaluated. An alternative method of controlling the overstroke transient is to bring the system on-line in a number of steps. This can be achieved by making the initial connection to the grid through a series resistor. The value of this resistor should be large enough to control the transient and small enough to ensure system stability. The value of series resistance can then be reduced in steps after the decay of each successive transient until the system is directly connected to the grid.

The shutdown transient, Figure 4-12, does not violate the engine geometric constraints and can be made in a straightforward fashion using standard switching techniques.

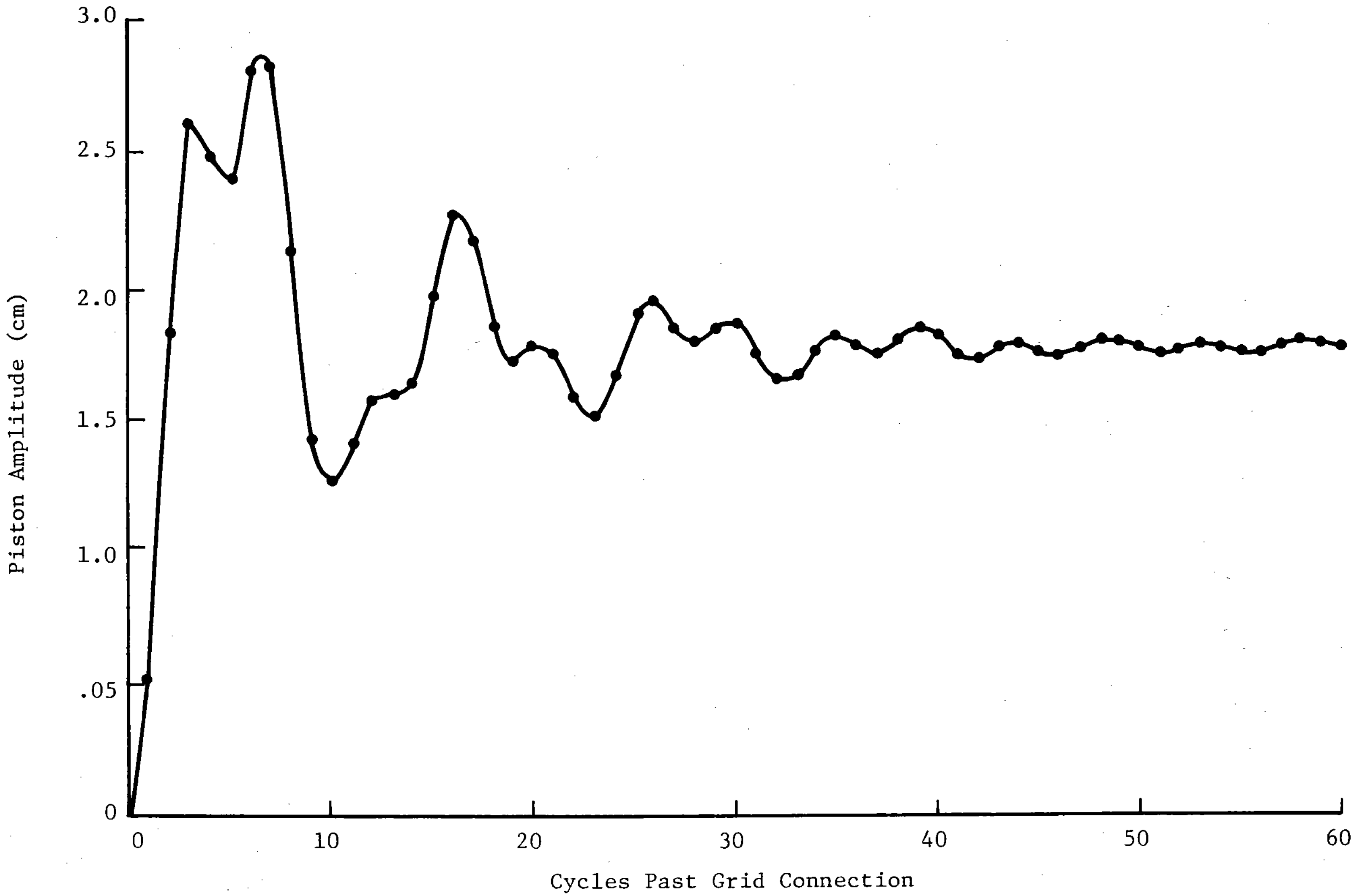

Fig. 4-10 Transient Engine Response
Mode II to Mode I Conversion

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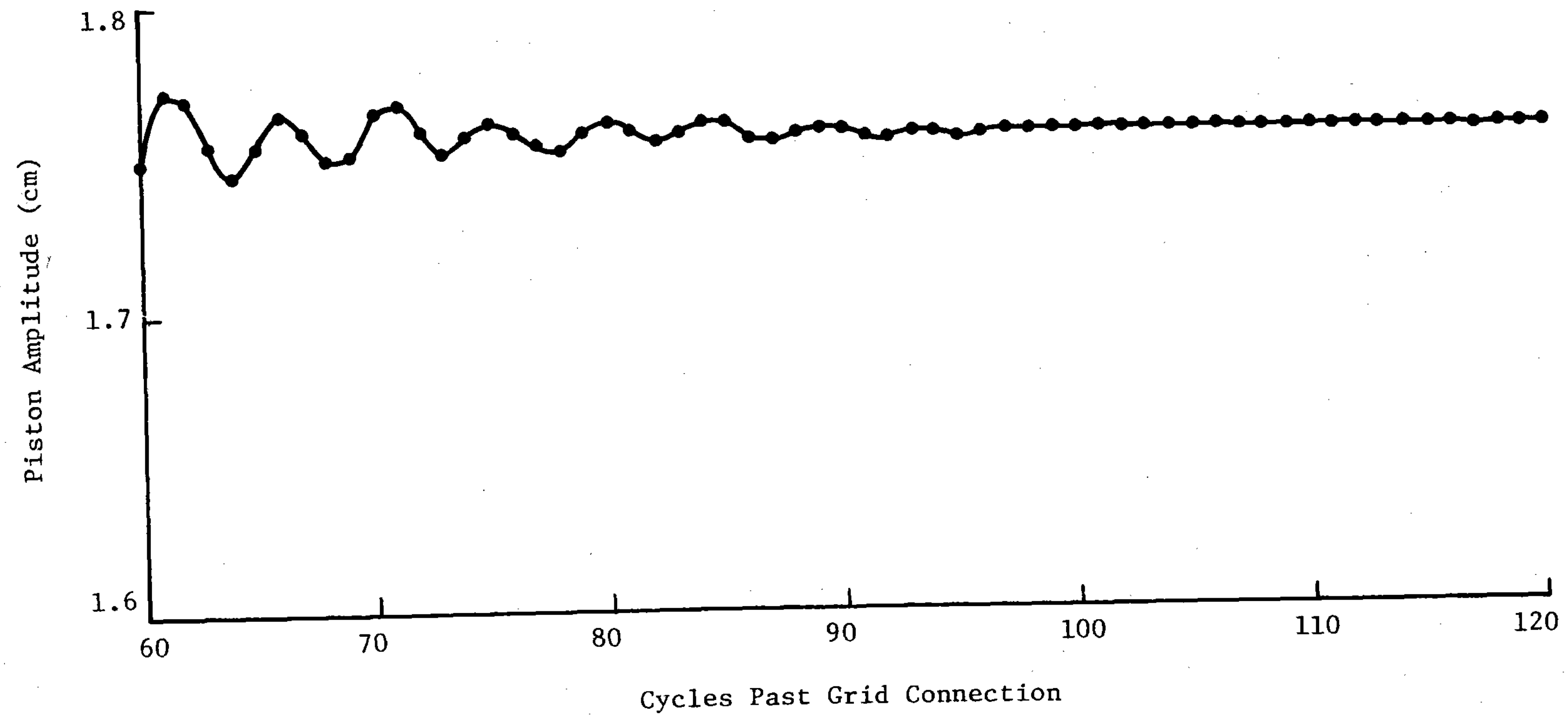

Fig. 4-11 Transient Engine Response
Mode II to Mode I Conversion (Cont'd)

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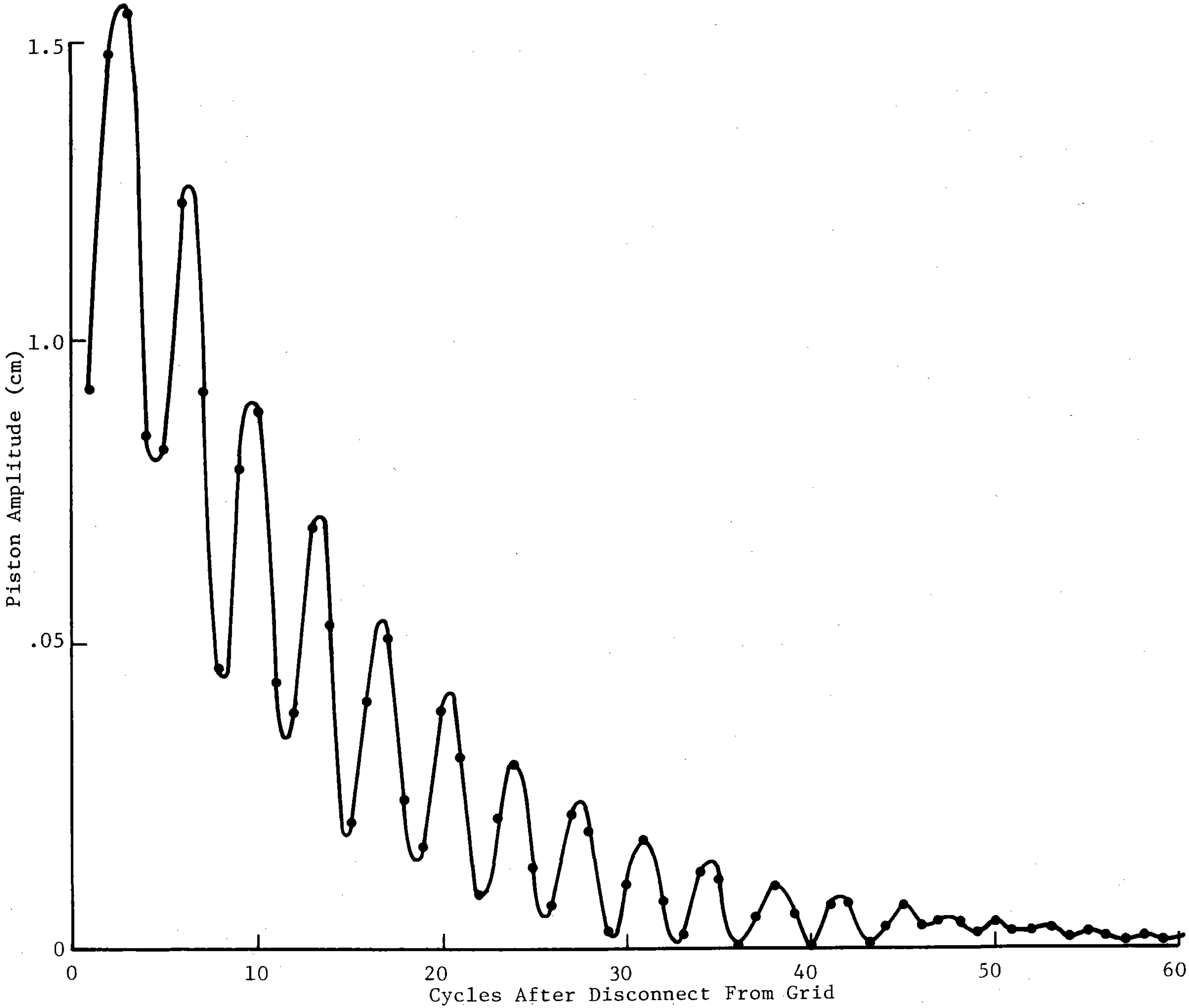

Fig. 4-12 Transient Engine Response

4.7.3 Power Modulation by Engine Pressure Control

The strong dependence of output power on engine mean-charge pressure requires that some form of pressure control device be used to correct for temperature induced mean-pressure variations and long-term system leakage. This control device provides a convenient means for controlling engine power output. A potential pressure control and power modulation system is shown in Figure 4-13.

The system consists of two long-life reed valves and two electrically activated solenoid valves connecting the engine working spare and a helium storage tank. A reduction of engine mean pressure is accomplished by activating solenoid A, and an increase in pressure is accomplished by activating solenoid B. Activation of solenoid A results in a pressure reduction by allowing the engine working fluid to be transferred from the engine to the storage tank through the outgoing check valve during that portion of the engine cycle in which the working space pressure is higher than the storage tank pressure.

Proper operation of the system in both directions requires that the tank pressure be between the maximum and minimum value of working space pressure. When the tank pressure equals the maximum working space pressure, activation of solenoid A no longer effects system operation. Similarly, solenoid B is ineffective when the tank pressure equals the minimum working space pressure.

Since the tank pressure tends to increase as engine pressure decreases, the effective pressure control range is dependent on both the engine pressure amplitude and the size of the storage tank. The maximum change in engine charge pressure as a fraction of the nominal charge pressure is approximately given by:

$$\frac{\Delta P}{P} \cong \frac{\pm\ \alpha X}{1 \mp \alpha X + \beta}$$

where:

α is the normalized pressure amplitude,

X is the ratio of final to nominal engine stroke,

β is the ratio of tank pressure rise to engine pressure drop corresponding to a small fluid transfer.

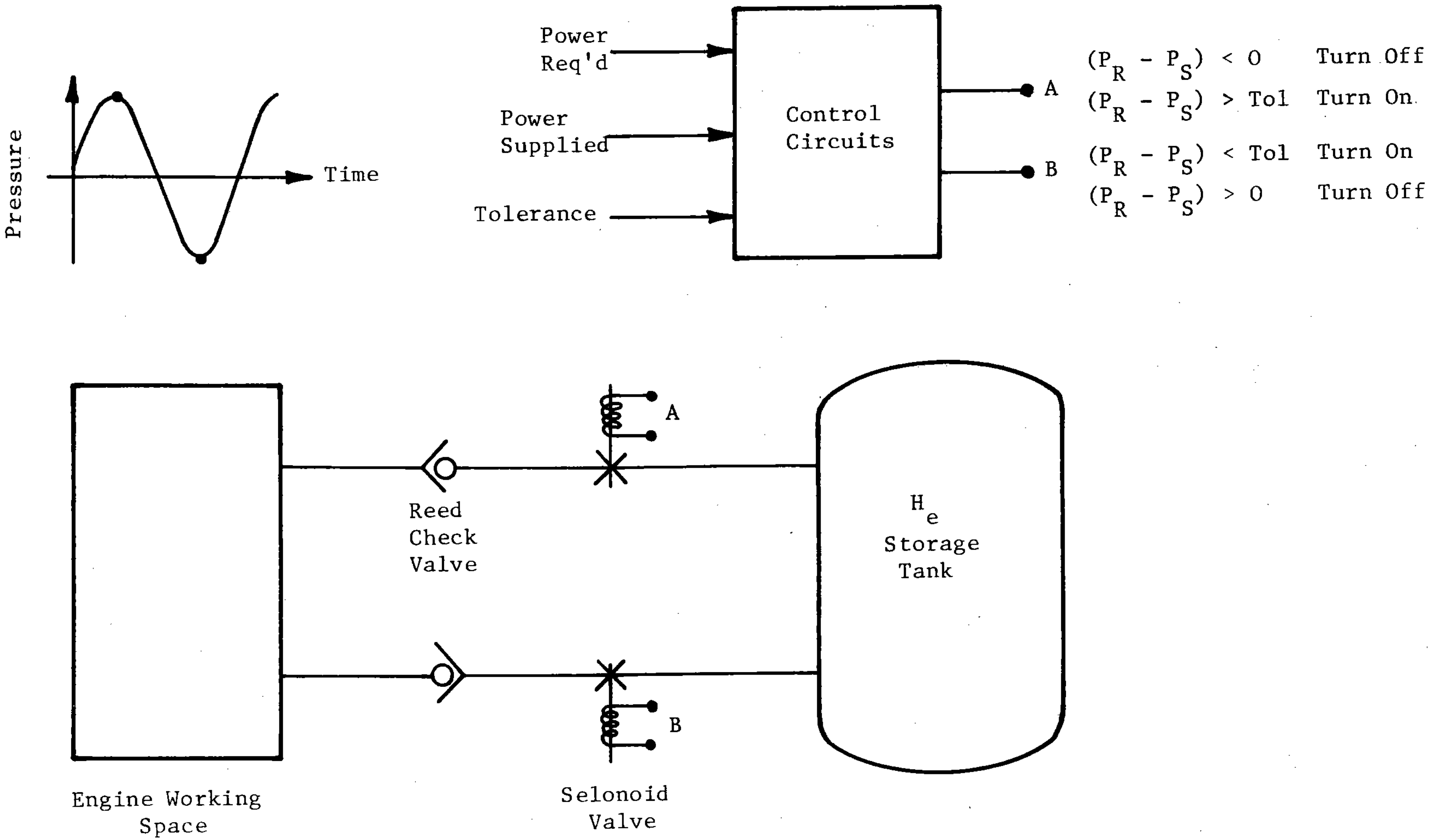

Fig. 4-13 Pressure Control System

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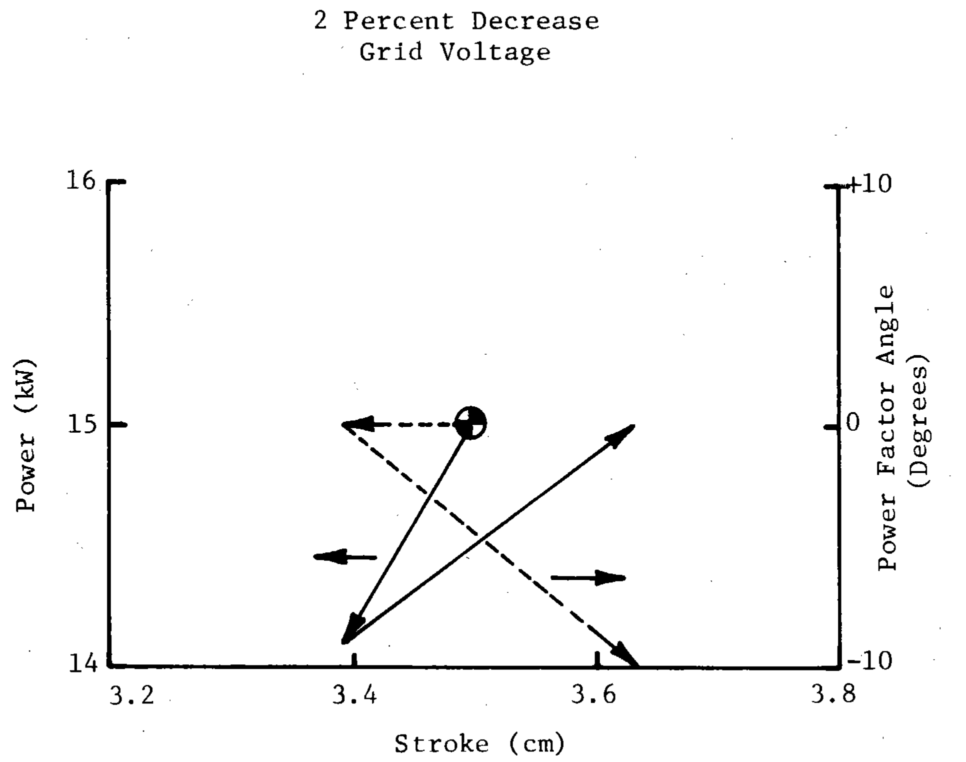
A positive value of ΔP corresponds to an engine pressure increase, and the negative value corresponds to an engine pressure decrease. The present engine design has an α value of .15. For a storage tank with an internal volume of one-quarter the size of the working space, the maximum fractional change in working space pressure at constant stroke is +.031 and -.029.

Figure 4-14 illustrates the response of the system to a two percent decrease in grid voltage and head temperature (solid vectors). Also shown in this figure is the response of the system to a change in engine pressure sufficient enough to return the system to the nominal 15 kW output. The values of engine and power factor angle after the power correction differ from the nominal values of 3.5 cm and 0°, respectively.

4.7.4 Overall System Control

It is anticipated that individual collector units will be arranged in clusters with the number of units per cluster being dependent on the individual unit size and the utility grid requirements. Control of the units in a cluster arrangement is best exercised via a digital computer-based central control system. The advantages of this control approach are as follows:

- Start-up and shutdown of individual units can be sequenced to control grid transients.
- Statistical variations in the stored energy at start-up of the individual units can be used to account for uncertainties in predicted insolation level and cloud cover.
- The output of the cluster can be conveniently matched to the requirement of the grid.
- Duplication of control hardware is eliminated.
- Cluster operating strategy can be conveniently changed in response to experience gained from system operation.
- Monitoring of unit performance for maintenance purposes is simplified.



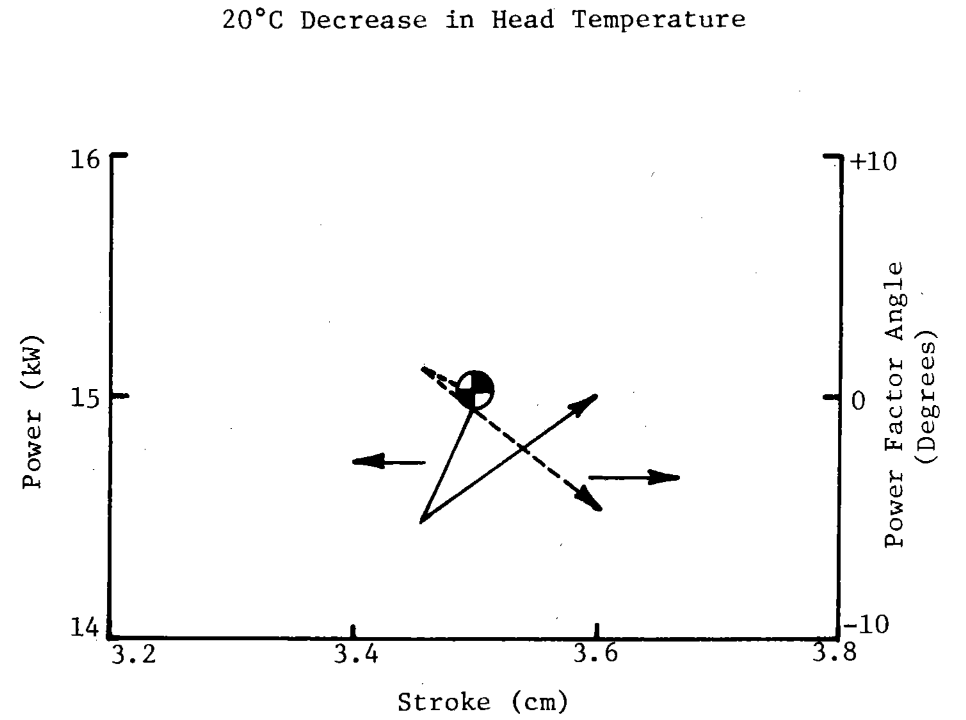


Fig. 4-14 Pressure Control Response Diagram

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A schematic layout for a potential centralized control system is shown in Figure 4-15. The control system communicates to each unit an electrical output requirement, and the units communicate the actual power output and receiver temperature. The power channel also serves to transmit the start-up and shutdown commands.

Basically, control of the system involves the choice of two major parameters for each unit: the initial value of stored energy (start delay), and the output power level. These two parameters provide a reasonable range of control flexibility. In general, for days with discrete clouds, the major cloud parameters are percentage cloud cover and repitition rate. The former tends to set the maximum average output and the latter tends to set the minimum stored energy requirement. If the cloud conditions do not tax the full capacity of the storage system, then to some degree the power output can be traded against the duration of the generating period.

4.7.5 Typical Operating Profile

A typical operating profile for an individual collector unit is shown in Figure 4-16. The profile is based on the assumption that all latent-stored energy has been expended during the previous generator period and that the receiver temperature has been dropping slowly during the overnight shutdown.

In order to reduce this temperature drop to a minimum, the receiver aperture is blocked with a night cover and the engine is isolated from the receiver by the thermal value. Upon acquisition of the sun, the solar input begins to replace the lost sensible energy and the receiver temperature begins to rise. When the receiver reaches the storage temperature, the central computer takes note of the time and from this point forward, the computer monitors the stored energy level by integrating the measured insolation level. When the calculated stored energy reaches a predetermined start value (approximately 65 percent of storage capacity in Figure 4-16. the start command and the output power goal are transmitted to the unit. For the example cited, there is no cloud cover immediately following the start, and the power output level transmitted to the unit is below the balance point value. Consequently, the stored energy continues to rise during the initial operating period but at

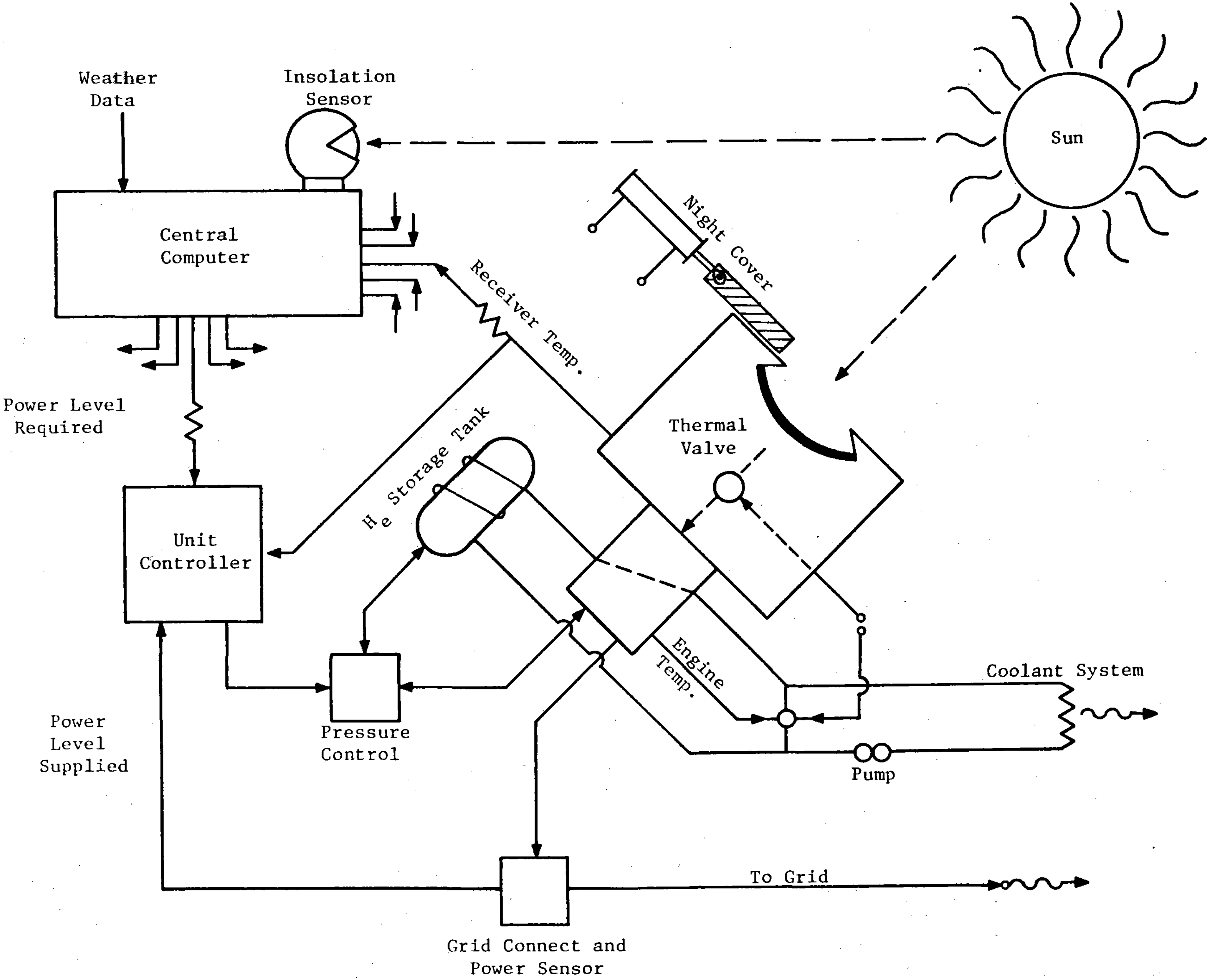

Fig. 4-15 Basic Engine Control System

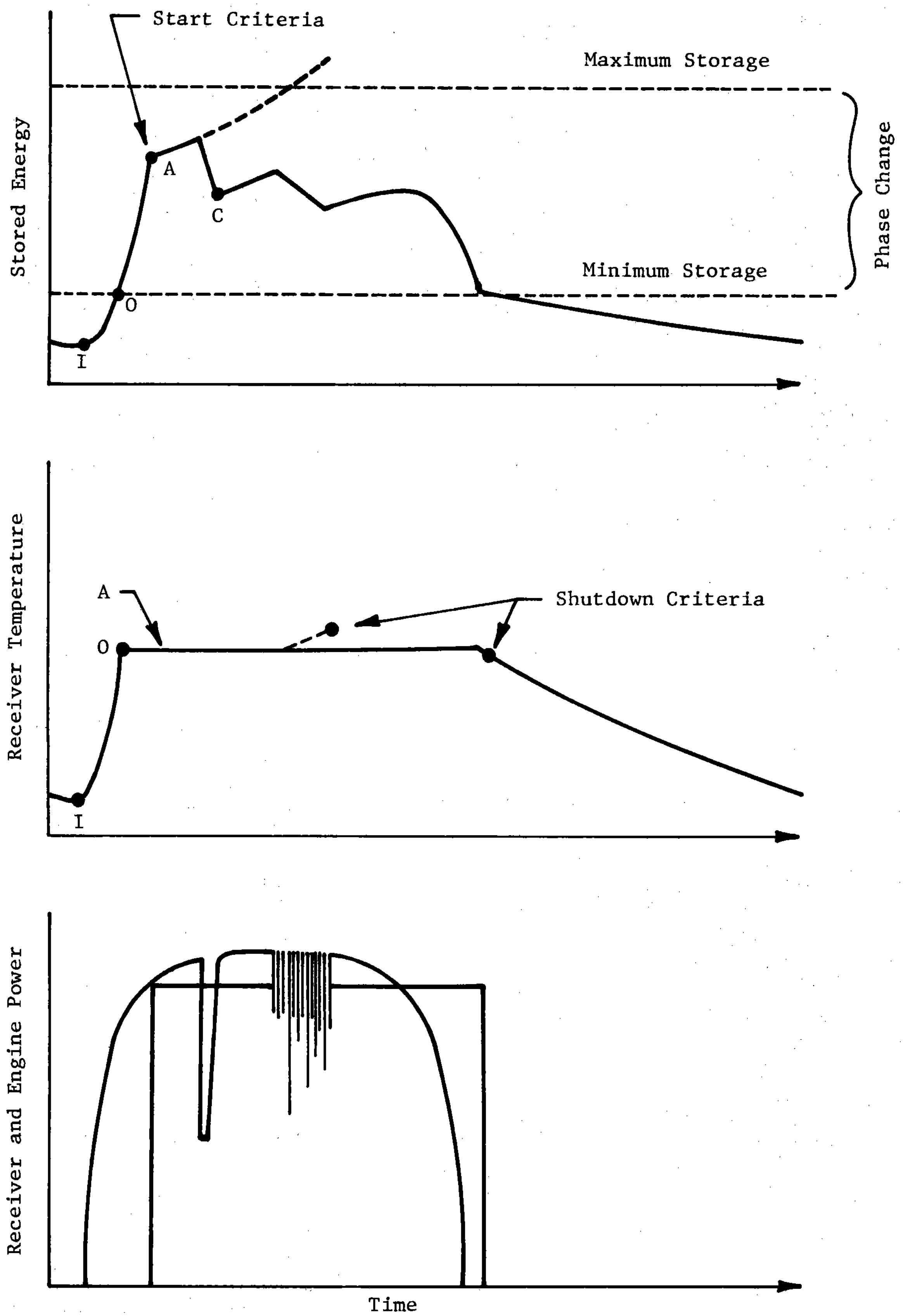

Fig. 4-16 Typical Operating Profiles

a much slower rate than before start-up. The stored energy rises to approximately 75 percent capacity (point A), at which time a sizable cloud temporarily blocks the solar input. The system continues to operate on stored energy and provides constant electrical output. Cloud passage occurs when stored capacity has dropped to 45 percent (point C), and the system again begins to increase stored energy while maintaining constant output. If the solar input had not been blocked, the stored energy would have increased until the receiver storage capacity was exceeded. At that point, the receiver temperature would begin to rise. This temperature rise is communicated to the central computer to update its estimate of stored energy and, to obtain a revised power output requirement. If the central computer does not respond, then the unit initiates a shutdown when the critical receiver temperature is reached.

Returning to the typical profile (solid curve), the second clear period is followed by a period of rapid cloud passage and then by a third clear period. Sunset occurs during this last clear period and shutdown is initiated by the central computer when all the latent energy of the receiver is expended. This occurance is signaled by a drop in receiver temperature. Since a number of units may reach this condition simultaneously, the shutdown signals are appropriately sequenced to moderate the grid transients.

4.8 Engine/System Interface

4.8.1 Engine/GE Heat Receiver

The interaction between the MTI Stirling engine and the GE heat receiver (GE Drawing #707E934) has been investigated, and the two have been found to be compatible without change to either. The MTI engine (Figure 4-17) is illustrated with GE's existing attachment points shown in phantom. The heater head fits easily in the space provided in the receiver, the sealing bellows size and location is satisfactory, and the conical engine mounting ring attached to the receiver is the right size, as can be seen in Figure 4-17. However, due to the diameter of the Stirling engine, it would be desirable to either segment the existing ring or alter the joint structure, as shown, in order to provide extraction clearance.

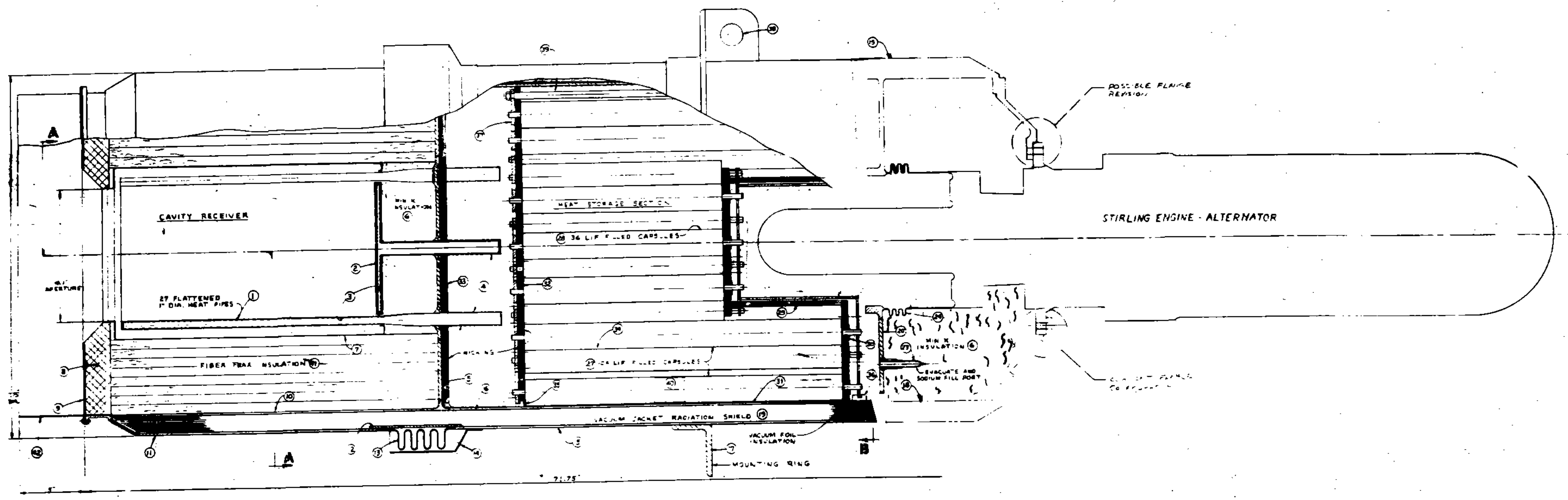

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Fig. 4-17 Stirling Engine/Heat Receiver Interface

After looking at the GE receiver design it became apparent that it would be advantageous to remove the complete Stirling engine from the receiver without having to disassemble either. To that end, an interface was devised which would prevent contamination of both the engine and receiver. A septum which closely conforms to the shape of the heater head would be permanently attached to the receiver, forming a socket into which the heater head could be inserted. The intermediate space would then be filled with a nontoxic, high conductivity metal like tin or zinc which would melt and provide a low loss thermal coupling which could be disassembled without admitting contaminant gasses into the heat pipe system or allowing sodium vapor to escape.

4.8.1.1 Thermal Switch. Once the basic liquid metal interface had been defined another use was suggested. If an appropriate storage vessel and connecting plumbing were supplied, then the interface could also be used as a thermal switch. By removing the liquid metal from the interface to the storage vessel it would be possible to greatly reduce the heat flux from the receiver to the engine. This switch could be used to assist in retaining heat overnight in the receiver and/or to hasten warm up in the morning.

If the liquid metal, e.g., zinc, is replaced by a low conductivity gas, e.g., carbon dioxide, then the conductance will be reduced to a level .0004 of the original value and the main heat transfer mode will be radiation. If the engine is to be held at 300°F, calculations indicate only 4,800 Btu/hr need be removed by the engine cooling system. As a point of reference, the total heat flux when the engine is operating at full power is approximately 40,000 Btu/hr, indicating nearly a 10/1 reduction in heat loss from the thermal storage, without overheating any parts of the engine.

4.8.2 Direct Heat Receiver Concept

In addition to the main requirement that the Stirling engine would interface directly with the GE heat pipe receiver design, MTI was also requested to explore and evaluate a receiver system in which the heater head is heated directly by solar radiation. It was concluded in Subsection 2.3, Heat Input Concept, that the direct radiation system has the following disadvantages:

- Small heat storage capacity
- Uneven heat flux
- More complex heater tube arrangement
- Longer heater tube and more engine dead volume

Ignoring the thermal storage, the direct radiation system was further pursued, starting with heater tube modification. The selection of new heater tube geometry is such that the heat transfer area is increased with an acceptable increase in pumping loss as compared with the optimum configuration selected by the computer program. The pumping loss, including friction, entrance and exit losses, were calculated for the optimum tube geometry. Different tube diameters of different lengths were checked for pumping losses and compared with the optimum geometry. The final selection was a .25 in. ID by .35 in. OD tube arranged as shown in Figure 4-18. The effect on the engine overal thermal efficiency due to the increase in dead volume was not checked by computer program but judged to be small (less than 1 or 2 points of efficiency). To deal with the uneven heat flux, the tube is cast in a block of copper. It is expected that the high thermal conductivity of the copper will distribute the heat more evenly.

Improved System:

The coefficient of expansion and nickel diffusion problems were presented to NASA-Lewis. It was suggested that these problems may be minimized by burying the tube halfway into the copper block. An additional improvement was to allow more tubes to pass the outer diameter of the block where incident flux is the highest. A sketch of the improved version is shown in Figure 4-19.

A heat transfer analysis was attempted for the case of the tube half buried into a copper block. In the process of deriving the equations, it was soon recognized that a three-dimensional finite element heat transfer analysis is required. Limited by available resources, the analysis was further simplified by considering the copper block as a fin. Due to symmetry, only a quarter of the tube section was considered.

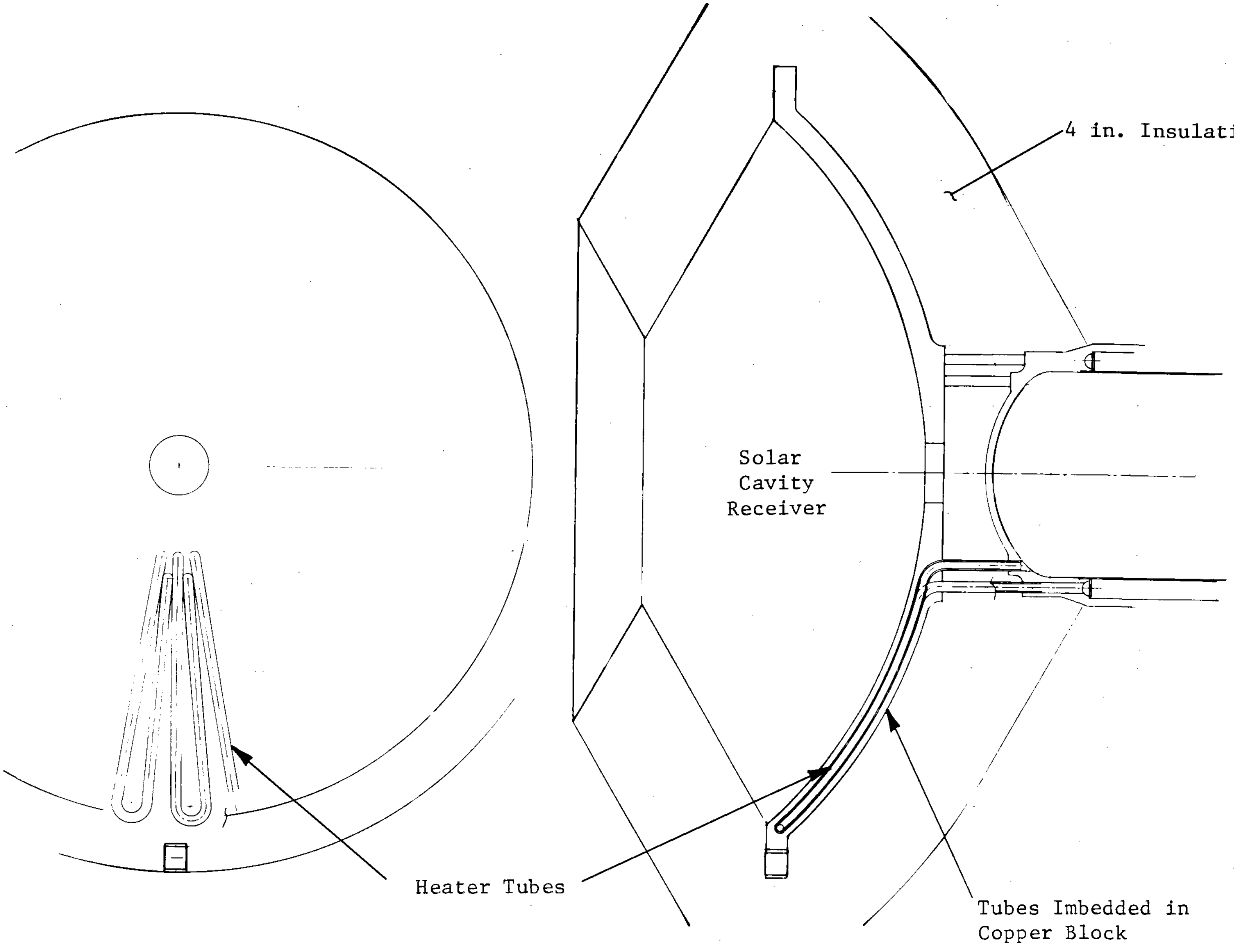

Fig. 4-18 Initial Direct Insolation Heater Head Concept

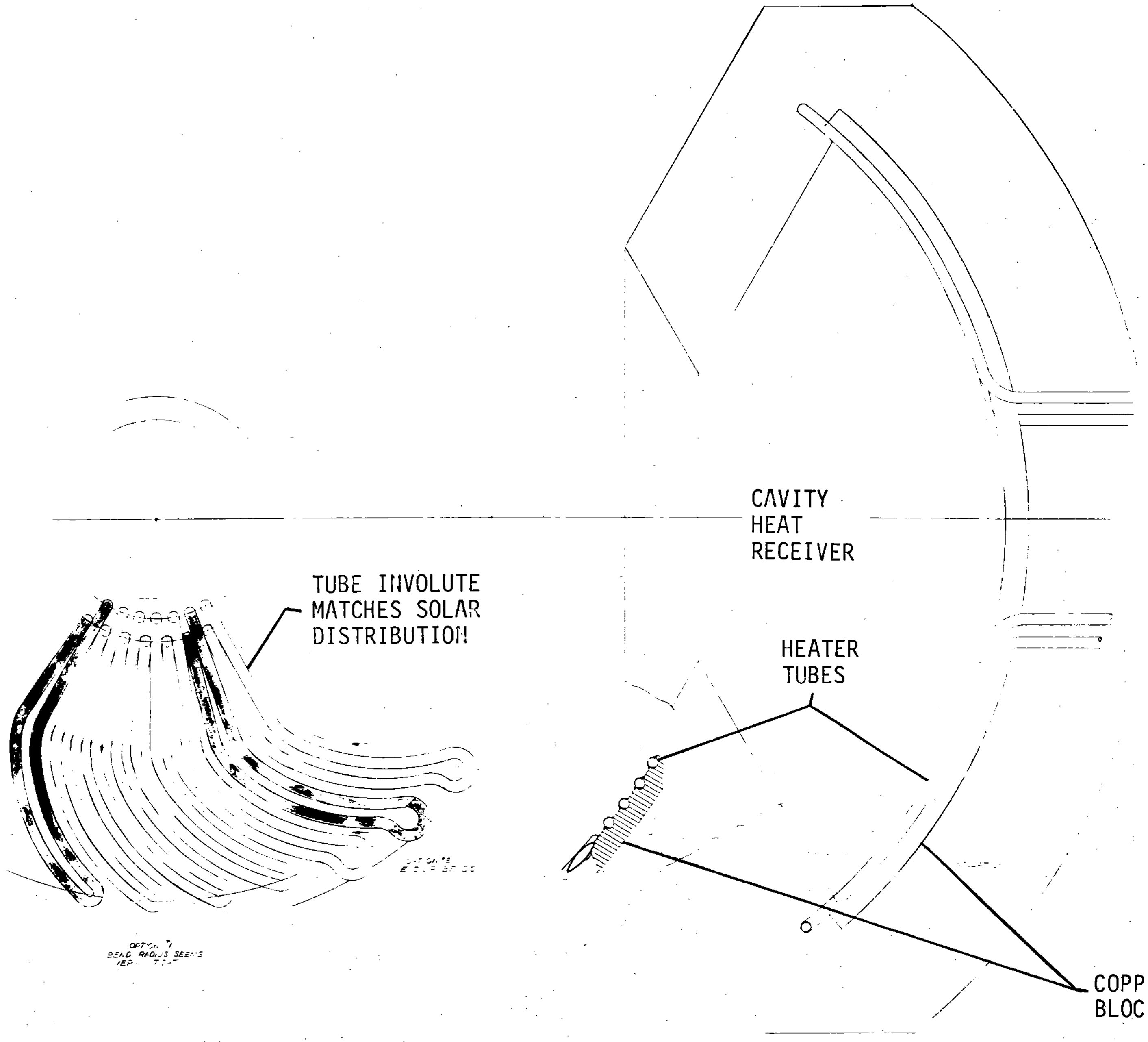

Fig. 4-19 Direct Insolation Heater Concept

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Based on the above analysis, 0.4 in. was selected for the thickness of the copper block. This is a simple and compact receiver with excellent efficiency. Further analysis is recommended for the final design.

4.8.3 Engine/Load Interface

The interface of the engine to the grid is discussed in detail in Subsection 4.7, System Control, of this report.

4.8.4 Auxiliary Heat Source

The purpose of the auxiliary heat source is to allow testing and checkout of the engine independently of a solar energy input. The main design considerations are as follows:

4.8.4.1 Performance Consideration: Solar insolation simulation - The heater must be capable of simulating a solar insolation profile. This means the auxiliary heat source should be very responsive to the rate of heat increase or decrease schedule. To accomplish this, the system should have a very low heat capacity so that little or no heat is given to the engine from the heat source when it is cutoff, or little or no heat is absorbed by the heat source when heat is turned on.

4.8.4.2 Operating Considerations: Reliability, durability, safety and ease of instrumentation are also main considerations. The capability to accurately evaluate the heat input to the engine is essential in this application. Cost and simplicity were also considered.

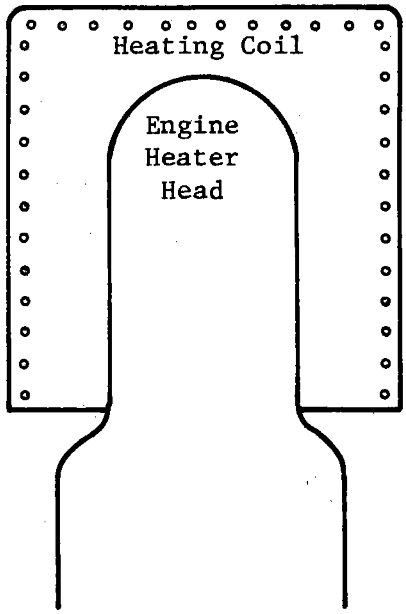
4.8.4.3 System Selection. A variety of heating systems were examined for their merit and the more promising ones were studied in detail to determine whether they should be adopted or rejected. Following is a summary of the evaluation.

a. Direct I^2R Heating - In this technique, the heater head itself is used as a resistance heating element. It is a simple, straightforward technique which is easy to control, instrument, monitor and fabricate. MTI has repeatedly used this technique for several of its engine testing programs.

In all these tests, the engine heater heads were tube-bundle types. Each individual heater tube is brazed on an electric lead which supplies a heavy current to the tubes. MTI has developed a number of electrically heated engines and made this heating system a practical technique.

The heater head geometry of the present system is a cylinder with a hemisphere on its top. All these surfaces are active heat transfer surface areas in operation. It is believed that a uniformly heating surface cannot be achieved by any arrangement of electric power to the head. Therefore this idea was not pursued.

b. Electric Heating Coil Heat Radiation System



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The heating coil has the following limitations:

Surface watt density: 25 watt/in.^{2}

Surface temperature in enclosure: 2200°F

This system was rejected for the following reasons:

1. The necessary heat supply for 16.5 kW output at the alternator requires a surface area of 1833 in.^{2} or 12.7 ft^2. This seems to be too large to be designed around the heater head.

2. Assuming the heater and the heater head are perfect blackbodies and view factors are equal to one, to radiate

heat to the heater head at a temperature of 1600°F, the heating coil has to be maintained at 2620°F which is above limit.

c. Combustion Heating - Brief calculations indicated that a combustion heater can be designed. By properly choosing the flame temperature and gas passages, adequate heat can be transferred to the heater head. In view of the wide range of load variation, it is believed that the control of fuel and airflow to simulate the solar insolation may require some development. This idea was not pursued further.

d. Sodium Boiler System - Sodium vapor heating duplicates the actual operating condition. A sodium boiler can be designed as an auxiliary heat source, but due to safety considerations, this system was also abandoned.

4.8.4.4 Recommended Induction Heating System. An induction heating system at the required power level is commercially available. It can deliver a large quantity of heat to a small space. It has a low system heat capacity, therefore, it has fast response to load changes. It is easy to control, instrument and monitor.

Suppliers of induction heating equipment were contacted for their recommendation. Recommended equipment for this application are:

- One (1) 100 kW 3 kHz solid-state frequency converter
- One (1) Model 23-MH Heat Station
- One (1) refractory-lined induction heating coil
- One (1) water cooled buss assembly
- One (1) temperature instrument/controller

Some development work may be required to design the coil so that the heat at the surface of the hemisphere is uniformly heated. A possible heating coil arrangement and a curve of power input versus power absorbed by heater head are shown in Figure 4-20.

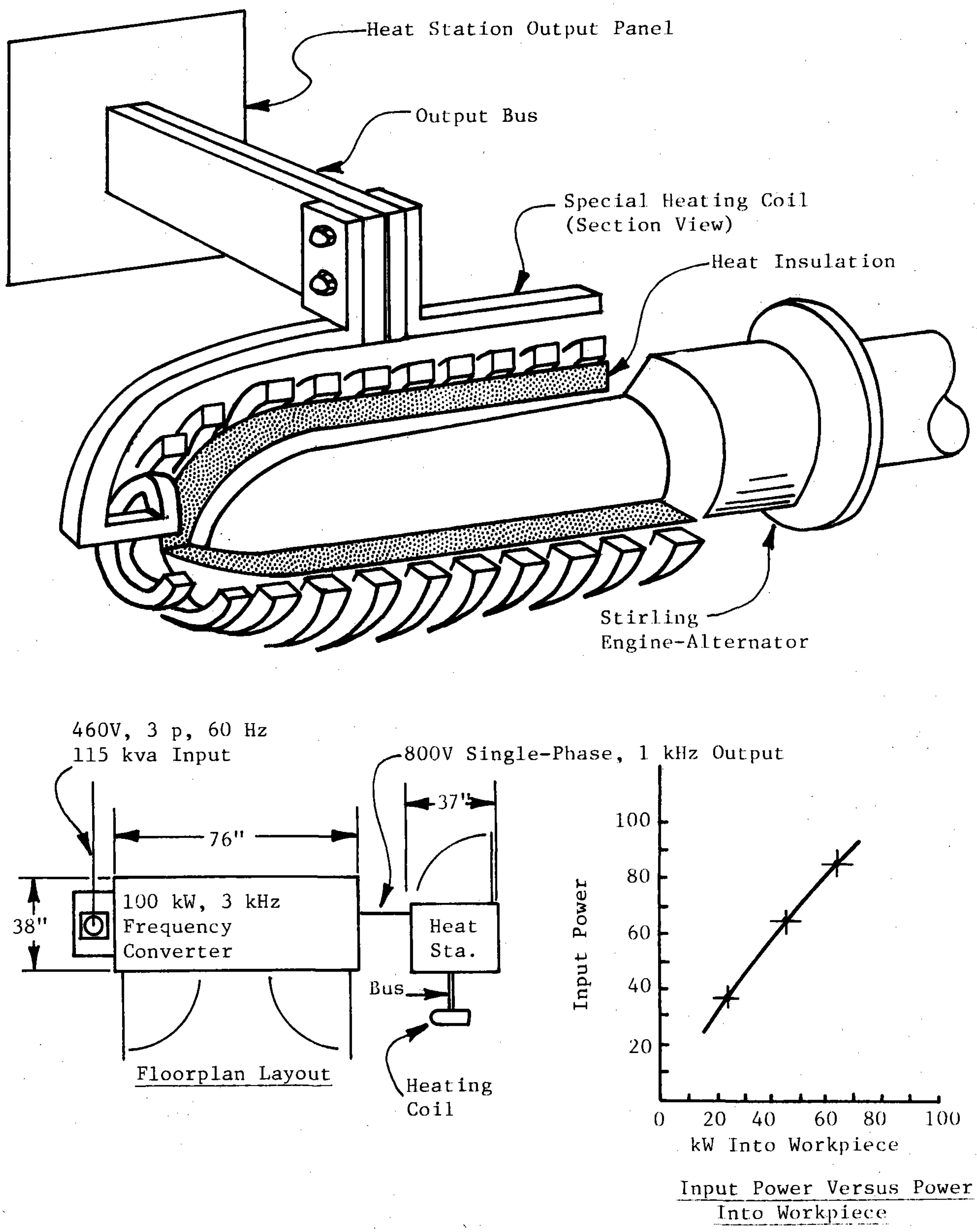

Fig. 4-20 Induction Heating System

This is a compact and fast response heating system. A computer simulation indicated that the heater head can reach 1600°F in thirty seconds (conduction loss ignored). Power control can be easily accomplished and the heat input to the engine can be monitored.

REFERENCES

1. Basiulis, A., and Ewell, G. I., "Heat Pipes for Hostile Environments in Energy Conservation Application", 12th IECEC, 1977.

2. Steiner, John C., "Stirling Thermal Engine Regenerator Cost Reduction Program Report II", General Motors Research Laboratories, July 22, 1965.

3. Blackman, A. W., "The Market Dynamics of Technological Substitutions", Technology Forecasting and Social Change, Vol. 6, 1974, pp. 41-63.