MASTER

FOSTER-MILLER ASSOCIATES, INC.

ENGINEERS
350 SECOND AVE.
WALTHAM, MA 02154
617 890-3200/TWX 710-324-1468

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

QUARTERLY REPORT NO. 1

Design and Development of Stirling
Engines for Stationary Power
Generation Applications in the
500 to 3000 Horsepower Range
Contract No. DE-AC02-79ET15208

Submitted by:

Foster-Miller Associates, Inc.
350 Second Avenue
Waltham, Massachusetts 02154

William M. Toscano
Program Manager

Submitted to:

U.S. Department of Energy
Argonne National Laboratory
Chicago Operations and Regional Office
9800 South Cass Avenue
Argonne, Illinois 60439

February, 1980

DISCLAIMER

This book was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

fer

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

TABLE OF CONTENTS

<u>Section</u>	<u>Page No.</u>
1. INTRODUCTION	1-1
2. MARKET ASSESSMENT	2-1
2.1 Present and Future Markets	2-1
2.2 Prime Mover Options	2-2
2.3 Design Specifications	2-4
3. DESIGN METHODOLOGY	3-1
3.1 Description of Design Synthesis	3-1
3.2 First Order Scaling Rules	3-3
3.3 Stirling Engine Configuration Decision Tree	3-3
3.4 Preliminary Engine Design Candidates	3-11
4. ENGINE THERMODYNAMIC PERFORMANCE EVALUATION	4-1
4.1 Second Order Computer Analysis	4-1
4.2 Comparison of Predicted Performance with Actual Performance	4-4
4.3 Sensitivity Analysis	4-4
4.4 Conclusions	4-11
5. PRELIMINARY SYSTEM DESIGN	5-1
5.1 Combustion System	5-1
5.2 Heat Transfer System	5-4
5.3 Preliminary System Configuration	5-4
6. CONCLUSIONS AND REMAINING WORK	6-1
REFERENCES	R-1

LIST OF ILLUSTRATIONS

<u>Figure</u>	<u>Page No.</u>
1-1 Program Organization	1-3
1-2 Schematic View of Small Multi-Fuel Stirling Engine Generator Set for Coal-Derived Clean Gases and Liquids or Current Low- Sulfur HC Fuels of Choice	1-6
1-3 Schematic View of Large Stirling Engine With Fluidized Bed Coal Combustion and Indirect Heat Transfer	1-7
3-1 Flow Diagram of the Design Synthesis	3-2
3-2 Stirling Engine Configuration Decision Tree	3-4
3-3 Fuel/Combustion System Configuration Decision Tree	3-5
3-4 Fuel/Combustion System Configuration Decision Tree (Continued)	3-6
3-5 Heater System Configuration Decision Tree	3-8
3-6 Machine Configuration Decision Tree	3-10
3-7 Single-Acting Manifolded Engine	3-12
3-8 Single-Acting Displacer V-Engine With Pressure Control	3-13
3-9 Double-Acting Rhinia In-Line Engine	3-14
3-10 Double-Acting Rhinia Barrel Engine With Variable Stroke Control	3-15
4-1 Effect of Manifold Gemoetry on Cyclic Heat Transfer and Engine Performance	4-6
4-2 Comparison of Power Predictions With Experi- mental Data	4-7
4-3 Comparison of Heat Input Predictions With Experimental Data	4-8
4-9 Comparison of Brake Efficiency Predictions With Experimental Data	4-9

LIST OF ILLUSTRATIONS (Continued)

<u>Figure</u>	<u>Page No.</u>
4-5 Effect of Control Method on the Stirling Engine Thermodynamic Performance	4-10
4-6 Effect of Heater Wall Temperature on the Stirling Engine Thermodynamic Performance	4-12
5-1 Dual Fuel Burner for the Stirling Engine	5-1
5-2 Atmospheric Fluidized Bed Coal Combustor For Stirling Engine Applications: Preliminary Schematic	5-3
5-3 Schematic Flowsheet: Atmospheric Fluidized Bed Coal Combustor for Stirling Engine	5-5
5-4 Heat Pipe Arrangement for Coal Fired Engine (Viking Design)	5-6
5-5 Schematic View of Small Multi-Fuel Stirling Engine Generator-Set for Coal-Derived Clean Gases and Liquids or Current Low-Sulfur HC Fuels of Choice	5-7
5-6 Schematic View of Large Stirling Engine With Fluidized Bed Coal Combustion and Indirect Heat Transfer	5-9

LIST OF TABLES

<u>Table</u>	<u>Page No.</u>
2-1 Comparison of Prime Mover Options	2-3
2-2 Design Specifications for the Stirling Engine for Stationary Power Generation	2-5
4-1 Governing Factors Influencing Stirling Engine Thermodynamic Performance	4-2
4-2 Comparison of Predicted Performance with Actual Performance	4-5

1. INTRODUCTION

This is the first Quarterly Report of the Design and Development of Stirling engines for Stationary Power Generation Applications in the 500 to 300 Horsepower Range. The report is divided into six sections:

- Introduction
- Market Assessment
- Design Methodology
- Engine Thermodynamic Performance Evaluation
- Preliminary System Design
- Conclusions and Remaining Work.

The market assessment of a Stationary Stirling engine for power generation, which was performed by Argonne National Laboratory [1], identified the need for a stationary power engine in total energy and cogeneration systems. The engine has to demonstrate both high thermal efficiency and fuel flexibility. There have been identified six energy market segments for the near future. For the future market the system characteristics of the total or integrated energy systems are an average size of 25 MW and a Stirling engine as the prime mover. Finally the market assessment established the design specifications of the Stirling engine prime mover.

With the market assessment completed, the program objectives were established.

- Develop a Stirling engine as a potential prime mover in the capacity range 500 to 3000 hp
- Integrate multifuel flexibility into the final design

- Competitive thermodynamic performance, emissions, and life cycle costs.

Foster-Miller Associates, Inc. (FMA) has assembled a team of engine companies and consultants to accomplish successfully, the first phase of the aforementioned program. As shown in Figure 1-1, FMA is responsible for program management, mechanical design, thermodynamic analysis, and the coordination of the subcontractor's efforts. Teledyne Continental Motors (TCM), a diesel engine manufacturer, is responsible for the manufacturing assessment and production cost assessment of the Stirling engine conceptual designs. The Stirling engine consultants, who are Mr. Stig Carlqvist, former managing director of United Stirling of Sweden, and Professor Joseph L. Smith, Jr. of the Massachusetts Institute of Technology, are responsible for providing previous Stirling engine technology and design evaluation. The remaining company participating in the program is Sunpower Corporation which is presided by Dr. William Beale, developer of the free piston Stirling engine. Sunpower is also responsible for providing previous Stirling engine technology and design evaluation. Part of the design evaluation will be the prediction of the engine performance with the Sunpower third order computer program. Finally, FMA has secured the consulting services of Thermacore, Inc. who will be responsible for evaluating heat pipe concepts for Stirling engines.

Not shown in Figure 1-1 are two other company participants. FMA early recognized the importance of a manufacturer assistance in the design and evaluation of fluidized bed combustion systems. Hence FMA has secured the participation of Dorr-Oliver, Inc. and Wormser, Inc.

Figure 1-1. Program Organization

The program approach consists of the following steps:

- Evaluate market assessment
- Design protocol developed for investigating engine configuration alternatives
- Thermodynamic analysis
- Configuration selection
- Manufacturing assessment.

Each of these steps are discussed in detail in the following sections. However, a few words regarding some of these steps. The thermodynamic analysis is important since a large Stirling engine with cylinder capacities of 100 hp and greater has never been built so it is a critical task to extrapolate the performance of the conceptual designs from existing data for Stirling engines with cylinder capacities of 25 hp and less.

The major engine design areas have been identified and include the following:

- Combustion system
- Heat transfer
- Stirling engine mechanical configuration
- Drive train mechanism.

The combustion system will consist of both a clean fuel exhaust gas recirculation system and a high performance atmospheric fluidized bed. For the latter combustion system the indirect heat transfer method will either be a pressurized inert gas circulation loop for the state-of-the-art design or heat pipes for the advanced conceptual design. The Stirling engine

mechanical configuration will include an integrated heater head/heat exchanger configuration which is adaptable to both combustion systems investigated. The drive train mechanism employs proven diesel engine technology.

In Figure 1-2 is presented a schematic view of a small multifuel Stirling engine generator set with a combustion system for coal-derived gases and liquids or low sulfur hydrocarbon fuels. The system is modular with each modular consisting of five cylinders, each developing 100 hp for a total of 500 hp. There are five heater heads, combustors and preheaters. There is a blower for exhaust gas recirculation which is used to preheat the air and vaporize the fuel for efficiency combustion after premixing. Also, there is a diesel type crankcase which includes crosshead and seal modules. The overall system dimensions of the engine module are 2 by 3.5 meters. Of course for a total output of 3000 hp, six engine modules are required.

In Figure 1-3 is shown a schematic view of a large, Stirling engine with an atmospheric fluidized bed coal combustion system with indirect heat transfer. The engine consists of six modules, each developing 500 hp for a total engine output of 3000 hp. There are a total of thirty cylinders. There is an airblower and an air preheater for the combustion system. The indirect heat transfer method is a pressurized inert gas circulation loop which is driven by a pump. The overall system dimensions are 14 by 7 meters.

For both systems the engine salient features are:

- Engine design includes an integrated heater head/heat exchanger which is easily adaptable for efficient coal combustion

Figure 1-2. Schematic View of Small Multi-Fuel Stirling Engine Generator-Set for Coal-Derived Clean Gases and Liquids or Current Low-Sulfur HC Fuels of Choice

Figure 1-3. Schematic View of Large Stirling Engine With Fluidized Bed Coal Combustion and Indirect Heat Transfer

- Monolithic heater head offers high performance and high reliability for large Stirling engines
- In-line, double acting engine is adaptable to different methods of capacity control for various duty cycle requirements
- Modular construction.

The conclusions developed to date include the following:

- For successful penetration into the integrated energy systems market, coal derived fuels must be used to assure fuel availability and low life cycle costs
- It appears that Stirling engines employing clean fossil fuels cannot compete with diesel engines
- Combustion technology exists for the successful burning of coal derived fuels in a large stationary Stirling engine
- Current fluidized bed combustion technology is adaptable for coal burning Stirling engines
- The development of large Stirling engines adapting existing technology for burning coal derived fuels should be completed
- The development team offers a broad range of experience in the development of combustion engines and related machinery, a major manufacturer of diesel and specialty engines, and consultants having extensive Stirling engine experience

- The modular design for the Stirling engine power section and output machinery (generic to diesel) is interchangeable with either option of coal derived fuels or coal combustion
- Large Stirling engines with high thermal efficiency may be extrapolated from previous small Stirling engines
- Modular Stirling engine "cylinder" design can be mated with current diesel crankcase, generator set practice to assure near term penetration into a low volume market
- Stirling engine thermodynamic performance is not strongly dependent on the overall mechanical configuration but rather on the heat transfer system
- The analysis, confirmed with comparison to actual engine data, predicts high thermal efficiency for a design tailored to stationary power generation.

The remaining Phase I tasks consist of the thermodynamic performance evaluation by means of the Sunpower third order analysis, the manufacturing assessment, and the production cost assessment of the conceptual engine designs.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

2. MARKET ASSESSMENT

A market assessment of a Stirling engine for total and integrated energy systems was performed by Argonne National Laboratory [1]. The market assessment revealed that there was a need for a stationary power engine in total energy and cogeneration systems. The engine must offer both high thermal efficiency and fuel flexibility, particularly the use of coal and coal-derived fuels. A summary of the results of that assessment which include present and future markets for stationary power generation is presented in this section. Also, the three viable prime mover options, Stirling, Diesel, and Brayton are examined and the final design specifications are presented.

2.1 Present and Future Markets

Prior to examining the engine options, it is worthwhile to examine both the present and future markets for integrated energy systems. The characteristics of the present market for stationary engines in the 500 to 4000 kW range for total energy systems are the following:

- As of 1974 there were only 232 installations in the United States
- Diesels comprised 5 percent, gas turbines comprised 13 percent, and steam turbines comprised 2 percent of the present market
- There are only two market segments, industrial and residential/commercial; the latter segment dominates
- Most of the total energy systems have capacities less than 10,000 kW.

The future market is expected to be different from the present market. Six energy market segments for total energy systems or integrated energy systems have been identified and are summarized as follows:

<u>Market Segment</u>	<u>Capacity Range</u>
● Total energy systems	0.1 to 60 MW
● Selective energy systems	0.1 to 60 MW
● Grid-connected systems	6 to 200 MW
● Coal using systems	6 to 200 MW
● District Heating and Cooling Systems	20 to 100 MW
● Thermal Transport Systems	20 to 1000 KW

From the results of the market assessment it is anticipated that the typical integrated energy system will have an average size of 25 MW and there will be a cumulative number of 12,000 systems by the year 2000.

2.2 Prime Mover Options

From the results of the market assessment it was discovered that there is a need for a prime mover for Stationary power generation applications. Three prime mover candidates were examined: Stirling, Diesel, and gas turbine. The results of the comparison are shown in Table 2-1. For the near term the Stirling is competitive with the Diesel and gas turbine. However, with respect to fuel flexibility, coal fuel, and life cycle costs, the Stirling is superior to the Diesel and gas

Table 2-1. Comparison of Prime Mover Options

	<u>STIRLING</u>	<u>DIESEL</u>	<u>GAS TURBINE</u>
EFFICIENCY (%)			
NEAR TERM	30 - 37	30 - 36	30 - 34
DEVELOPMENTAL	40 - 48	40 - 50	35 - 38
FUEL FLEXIBILITY	ALL FUELS	DIESEL, NATURAL GAS, POSSIBLE COAL	SPECIFIED LIQUID AND GASEOUS FUELS
LIFE MTBO (hr)	20,000 - 30,000	20,000 - 30,000	10,000 - 50,000
RELATIVE COST OF MAINTENANCE	1.0	1.0	0.8
WEIGHT/POWER CAPACITY (Lb/hp)	6 - 30	6 - 30	4 - 6
VOLUME/POWER CAPACITY (ft ³ /hp)	0.15 - 0.3	0.12 - 0.25	0.05 - 0.2
ELECTRICAL GENERATION COSTS (¢/kW-hr)			
NEAR TERM	4.5 - 5.4	4.6	6.3 - 6.7
DEVELOPMENTAL	3.6 - 4.4	4.4	6.3 - 6.7

turbine. For these three important reasons the Stirling was selected as the candidate prime mover.

The primary advantages of a Stirling engine may be summarized:

- High thermal efficiency
- Good part load characteristics
- Fuel flexibility
- Low emissions
- Good reliability
- Low noise.

2.3 Design Specifications

So from the results of the market assessment and prime mover options comparison, the design specifications were established. A summary of the design specifications for the Stirling engine for stationary power generation is presented in Table 2-2. The power range is between 500 to 3000 hp. The engine efficiency (excluding combustion system efficiency) is to be 40 percent or greater. The engine design should be compatible for both coal combustion and conventional fuels combustion.

Table 2-2. Design Specifications for the Stirling Engine for Stationary Power Generation

- POWER RANGE OF 500 TO 3000 hp
- ENGINE DESIGN SPEED COMPATIBLE WITH A 60 Hz ELECTRIC GENERATOR
- SHAFT POWER OUTPUT EFFICIENCY OF 40 PERCENT OR GREATER (EXCLUDING COMBUSTION EFFICIENCY)
- FUEL FLEXIBILITY - USE OF LOW COST FUELS SUCH AS COAL AND COAL-DERIVED LIQUIDS
- COSTS LESS THAN 150 PERCENT OF A COMPARABLE DIESEL ENGINE AND 180 PERCENT OF A SOLID FUEL ENGINE
- MTBO OF 10,000 hr OR GREATER

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

2-6

3. DESIGN METHODOLOGY

The design methodology consists of the design synthesis, incorporation of scaling rules into the design scaleup, the preparation of a configuration decision tree analysis, and selection of preliminary engine design candidates. Each of these design methodology categories is described in detail below.

3.1 Description of Design Synthesis

A flow diagram depicting the design synthesis is shown in Figure 3-1. The first step in the design synthesis is the evaluation of the design specifications which were derived from the market assessment. The next step is the combination of existing Stirling engine data, i.e., engine specifications and performance data, with standard engine scaling laws. The next step is the preparation of a Stirling engine configuration decision tree. From the decision tree, preliminary engine designs are selected and a thermodynamic parametric analysis is performed to determine the importance of various mechanical and thermodynamic variables on the engine thermodynamic performance, design, and cost. Note that based on the outcome of the thermodynamic analysis, it may be desirable to select other engine configurations. Once the engine configuration has been established both a refined design which incorporates the results of the thermodynamic sensitivity analysis and a final thermodynamic performance prediction are performed. Next detailed final layout drawings of the Stirling engine are prepared. Finally a manufacturing assessment and production cost estimates are performed. Depending on the results of the manufacturing assessment, the layout drawings may be modified.

Figure 3-1. Flow Diagram of the Design Synthesis

3.2 First Order Scaling Rules

In order to initiate a design analysis it was necessary to scale Stirling engines to the large sizes. The preliminary scaling rules are the following:

- Power is proportional to the cross sectional flow area in the regenerator
- The linear dimension is inversely proportional to the engine speed
- The ratio of the heat transfer coefficient to the hydraulic radius is proportional to the ratio of the friction factor to the hydraulic radius and inversely proportional to engine speed.

The assumptions for these scaling rules are one dimensional flow, perfect gas law, Nusselt number and friction factor are a function of Reynolds number, and the engine may be described by pressure, temperature and velocity which are functions of position and time.

3.3 Stirling Engine Configuration Decision Tree

The Stirling engine design may be separated into three subsystems: fuel/combustion, heater, and machine. This is illustrated in Figure 3-2. Each of these subsystems is examined in detail in Figures 3-3 to 3-7.

In Figures 3-3 and 3-4 are presented the configuration decision tree for the fuel/combustion system. The fuel/combustion options are gas, distillate fuel, residual fuel, and coal. Only the first three options are presented in Figure 3-3 and coal is presented in Figure 3-4. For the clean

Figure 3-2. Stirling Engine Configuration Decision Tree

Figure 3-3. Fuel/Combustion System Configuration Decision Tree

Figure 3-4. Fuel/Combustion System Configuration Decision Tree (Continued)

fuels it is preferable to use high exhaust gas recirculation (EGR) with premixing to maintain low NO_x emissions and a short, clean, nonluminous "gas type" flame. Residual fuel with high sulfur content is considered the same as coal. In Figure 3-4 are presented the coal combustion alternatives.

- Direct coal combustion
- Onsite coal gasification
- Pressurized fluidized bed coal combustion
- Atmospheric fluidized bed coal combustion.

For direct combustion, considerable clean-up equipment, i.e., flue gas desulphurization, is required. It is felt that this method is not practical for the short term. For the design capacity of 500 to 3000 hp, onsite coal gasification is not practical economically. Pressurized fluidized bed coal combustion is presently being investigated and it is not expected to be developed in the near term. The only near term viable solid coal combustion method is atmospheric fluidized bed coal combustion (AFBC), and of the many methods for AFBC, it is felt that the Wormser Grate technique is more appropriate for a Stirling engine.

There are two possible methods for heat transfer to the hot end of the Stirling engine: direct heat transfer and indirect heat transfer. In Figure 3-5 is presented the heater system configuration decision tree for the Stirling engine. The results of the analysis for the heater have revealed that for direct heat transfer the high exhaust gas recirculation is the better method for maintaining low NO_x emissions and for indirect heat transfer the inert gas circulation loop is the best method. The heat pipe would be employed for the advanced engine design concepts.

Figure 3-5. Heater System Configuration Decision Tree

The machine configuration is dependent on both the heat exchanger configuration and engine configuration. In Figure 3-6 is presented the machine configuration decision tree. There are three possible ways to arrange the heat exchanger: axial, manifold, and concentric. The axial heat exchanger configuration has minimum dead volume; however, it is inconvenient for mechanical adaptation and it is not advantageous for coupling with normal heater systems. The manifolded configuration has large dead volumes and high temperature swings which promote high cyclic heat transfer losses. The concentric heat exchanger arrangement is nearly compatible with all heater systems.

There are two possible methods for arranging the engine: single acting and double acting. The single acting engine typically had larger cylinders and more moving parts. For a single acting engine either an opposed V or U arrangement or a displacer arrangement may be employed. The single acting displacer engine has demonstrated higher reliability characteristics. For a double acting engine either a Rhinia or an opposed piston arrangement may be employed. The double acting opposed piston arrangement presents hot rod seal problems, also it is not a viable engine configuration candidate. The double acting Rhinia may either be arranged in-line or barrel. It is worth mentioning that all of the engine configurations, if designed properly, will operate at the same thermodynamic efficiency.

There have been listed several methods of capacity control, either phase, pressure, or stroke. Depending on the results of the thermodynamic parametric analysis, the method of capacity control will be selected.

Figure 3-6. Machine Configuration Decision Tree

Preliminary machine configuration conclusions are:

- Approximately the same thermodynamic efficiency is obtained with any machine configuration
- Axisymmetric integrated Stirling engine design is applicable for any drive configuration
- Double acting Rhinia in-line engine with one crank and levers for parallel cylinders with pressure control is the simplest. Two auxiliary cranks are required for stroke control.

3.4 Preliminary Engine Design Candidates

Preliminary engine designs for:

- Single acting manifolded engine
- Single acting displacer, V engine with pressure control
- Double acting Rhinia in-line engine
- Double acting Rhinia barrel engine with variable stroke control

are presented in Figures 3-7 through 3-10, respectively. All of the engine designs employ an integrated piston-regenerator-heater head which has internal thermal insulation. For all of the engine designs, provisions have been made to incorporate conventional fuel systems as well as fluidized bed combustion systems.

The next step in the design synthesis is to evaluate the preliminary engine designs. The engine design characteristic criterion include:

- Thermodynamic performance
- Fuel flexibility

Figure 3-7. Single-Acting Manifolded Engine

Figure 3-9. Double Acting Rhinia In-Line Engine

Figure 3-10. Double Acting Rhinia Barrel Engine
With Variable Stroke Control

- Manufacturing assessment
- Production costs and life cycle costs.

In the next sections each of these design criterion will be discussed and preferred engine designs will be presented.

4. ENGINE THERMODYNAMIC PERFORMANCE EVALUATION

The first step in analyzing the engine is to predict the engine thermodynamic performance. The thermodynamic evaluation included development of a second order analysis, thermodynamic parametric analysis, and prediction of engine performance by means of both a second order and third order analyses. Each step in the Stirling engine thermodynamic evaluation is discussed in the following subsections.

4.1 Second Order Computer Analysis

Stirling engines, like other heat engines, converts heat to useful work. In this conversion process, there are various loss mechanisms due to the non-ideal nature of the device which prevents one from obtaining desired high efficiency. There are three groups of loss mechanisms that influence the Stirling engine performance. The first group of losses occur in the combustion system where fuel is burned to produce heat. Some of the fuel energy is wasted between heat source and engine. The second group of losses are called heat input loss. These losses occur because some of the heat delivered to the engine leaves the engine without producing useful work. The third group is called power loss because these losses occur after the heat input is converted to useful work. These loss mechanisms have to be carefully investigated and minimized in designing a Stirling engine in order to obtain high performance. A list of the loss mechanisms is presented in Table 4-1.

Foster-Miller Associates has prepared a second order analysis for examining the various loss mechanisms in the Stirling engine. In addition, Sunpower Corporation is performing analyses of the engine designs with its third order analysis. The original version of the second order analysis was developed by Professor Smith and Dr. Rios of MIT [2].

Table 4-1. Governing Factors Influencing Stirling
Engine Thermodynamic Performance

LOSS MECHANISMS

POWER LOSS

- **PRESSURE DROP LOSS** { **HEATER**
COOLER
REGENERATOR
- **HEAT TRANSFER LOSS** { **HEATER**
COOLER
- **CYCLIC HEAT TRANSFER LOSS**
- **LEAKAGE LOSSES**
- **MECHANICAL FRICTION LOSSES**

HEAT INPUT LOSS

- **STATIC HEAT TRANSFER LOSS**
- **SHUTTLE HEAT TRANSFER LOSS**
- **CYLINDER-PISTON GAP PUMPING LOSS**
- **REGENERATOR HEAT TRANSFER LOSS**

COMBUSTION SYSTEM LOSS

Basically, the second order analysis assumes:

1. Cylinders are adiabatic, i.e., heat transfer does not occur between the gas and wall inside the cylinders
2. All the heat exchangers are perfect
3. Working gas is a perfect gas.

Based on these assumptions, differential equations for engine pressure are derived and numerically integrated to obtain basic engine performance. The next step is to account for the non-ideal nature of the engine components such as heat transfer inside the cylinder, pressure drop in the heat exchanger, imperfect heat exchange pressures, and all other losses mentioned. The original second order version was modified to include all the losses listed in Table 4-1.

The most salient feature of the FMA second order analysis is the cyclic heat transfer loss mechanism. This loss mechanism is similar to gas spring loss due to cyclic heating and cooling at the gas wall interface. Up to now, this loss mechanism has not been recognized by Stirling engine analysts as a significant loss mechanism for Stirling engines. FMA has identified the significance of this loss mechanism and included it in its second order analysis.

It was discovered that this loss was the largest power loss mechanism in Stirling engines. After the cyclic heat transfer loss was included in the second order analysis most of the persistent discrepancy between the test data and prediction disappeared. It is thought that the failure to recognize this important loss mechanism in the Stirling engine led other investigators to use "fudge" factors in their analysis to match engine test data.

Illustrated in Figure 4-1 is an example of how this loss mechanism can be detrimental to engine performance. The geometry of the manifold for one engine design was changed in order to increase the surface area. The results show that when surface to volume ratio of this manifold quadrupled, the engine power and efficiency dropped approximately 10 percent. The influence of improper sizing of manifolds on performance loss has also been observed by others.

4.2 Comparison of Predicted Performance with Actual Performance

In order to validate the second order analysis, the performance of actual Stirling engines was predicted and compared with actual data as shown in Table 4-2. The second order analysis predicts brake power with maximum error of 15 percent and efficiency with maximum error of 11.6 percent. In Figures 4-2, 4-3, and 4-4 are presented the comparisons of the second order analysis engine performance predictions with actual experimental data of the GPU-3 Stirling engine. The correlation of the second order analysis predictions with actual engine performance data is very good, particularly since the second order model is based solely on the fundamental principals.

4.3 Sensitivity Analysis

With the aid of the second order analysis computer program, a sensitivity analysis of various engine design parameters was performed in order to determine the significance of these parameters on the design of the high efficiency Stirling engine. The sensitivity analysis included the investigation of engine phase angle, heater temperature, pressure, and dead volume on engine performance. As an example of how useful the results of the sensitivity analysis are for mechanical design preparation, Figure 4-5 shows relative merits of various power control options

Table 4-2. Comparison of Predicted Performance With Actual Performance

ENGINE NAME	OPERATING CONDITION			BRAKE POWER/CYLINDER			BRAKE EFFICIENCY		
	TEMP.	MEAN PRESS.	RPM	PREDICTED kW	ACTUAL kW	ERROR %	PREDICTED %	ACTUAL %	ERROR %
GPU-3	HEATER 704°C	400 psi	1000	1.15	1.14	1.0	25.5	24.4	4.5
			1500	1.725	1.50	15.0	28.4	25.5	11.4
	COOLER 15°C		2000	2.25	1.98	13.6	29.3	27.1	8.1
			2500	2.58	2.39	7.9	28.0	26.8	4.5
			3000	2.75	2.61	5.4	25.9	25.5	1.6
4-D1	HEATER 650°C COOLER 75°C	2177 psi	1400	5.0	5.0	0.0	33.1	32.0	3.5
4-D2	HEATER 720°C COOLER 50°C	2177 psi	1000	9.56	9.5	1.0	35.2	34.0	3.5
			1250	11.95	11.0	9.0	35.4	36.0	-1.7
			1500	14.0	12.5	12.0	34.5	33.0	4.5

Figure 4-1. Effect of Manifold Geometry on Cyclic Heat Transfer and Engine Performance

GPU-3 HYDROGEN 704°C/15°C
MEAN PRESSURE 400 psi

LEGEND

- × PREDICTED BY FMA CODE (2ND ORDER)
- PREDICTED BY NASA LERC CODE (3RD ORDER)
- EXPERIMENTAL

Figure 4-2. Comparison of Power Predictions With Experimental Data

GPU-3 HYDROGEN 704°C/15°C
MEAN PRESSURE 400 psi

Figure 4-3. Comparison of Heat Input Predictions With Experimental Data

GPU-3 HYDROGEN 704°C/15°C
 MEAN PRESSURE 400 psi

LEGEND

- + PREDICTED BY FMA CODE (2ND ORDER)
- PREDICTED BY NASA LERC CODE (3RD ORDER)
- EXPERIMENTAL, BASED ON HOT END HEAT BALANCE
- △ EXPERIMENTAL, BASED ON COLD END HEAT BALANCE

Figure 4-4. Comparison of Brake Efficiency Predictions With Experimental Data

Figure 4-5. Effect of Control Method on the Stirling Engine Thermodynamic Performance

for different power capacities for stationary Stirling engines. From the thermodynamic analysis, stroke control appears more favorable as compared to the other capacity control options.

In Figure 4-6 is presented the effect of heater wall temperature variations on engine performances. For 100°C increase in heater wall temperature, approximately 10 percent gain in power and efficiency will occur. This is what led to investigating the means of increasing heater head temperature. The near term solution is the employment of the monolithic cast iron heater head design, and the long term solution is the use of heat pipe technology.

4.4 Conclusions

The thermodynamic engine analysis conclusions may be summarized as follows:

- FMA second order analysis predicts Stirling engine thermodynamic performance within 15 percent error
- FMA second order analysis, based on first principles, is suitable for design analysis and optimization
- For 100°C increase in heater temperature it is possible to obtain a 10 percent increase in efficiency and power
- Stroke control appears to be the best power control method for stationary Stirling engines
- Engine optimized for helium can have the same efficiency as an engine optimized for hydrogen
- Cyclic heat transfer work loss is a major loss mechanism in Stirling engines, and has to be considered in designing the engine internal geometry

4-D2 He 15 MPa 1500 rpm, COOLANT TEMPERATURE 50°C

Figure 4-6. Effect of Heater Wall Temperature on the Stirling Engine Thermodynamic Performance

- In order to decrease the cyclic heat transfer work loss, it is necessary to approach isothermal processes in the cylinder.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

4-14

5. PRELIMINARY SYSTEM DESIGN

The preliminary system design consists of the combustion system, both the clean fuel combustion system and the fluidized bed combustion system, heat transfer system, and the Stirling engine. Each of these systems will be discussed in detail below.

5.1 Combustion System

A burner was designed to accomodate all of the preliminary engine designs. The burner is a dual fuel system, i.e., it will burn either a liquid fuel or a gaseous fuel. A flow schematic of the burner is shown in Figure 5-1. The liquid fuel, which may be conventional low sulfur hydrocarbon fuels or coal derived liquids, is vaporized by means of the exhaust gas recirculation (EGR). The vaporized fuel is then mixed with preheated air from the recuperator and finally is ignited. In a similar fashion, the gaseous fuel which may be either high or low BTU natural gas or coal derived gas, is preheated by means of the exhaust gas recirculation. The preheated gaseous fuel is then mixed with the preheated air and finally is ignited. The burner technology employed is very similar to that developed by FMA for the residential variable firing rate gas and oil burners.

For the burning of solid coal, an atmospheric fluidized bed coal combustion system for a Stirling engine was designed. A preliminary schematic of the combustion system is shown in Figure 5-2. The design is a two stage combustion system which was originally proposed and demonstrated by Wormser, Inc. The advantages of the system as compared to a conventional atmospheric fluidized bed combustion system are:

- The bed temperature can be maintained at 1550°F to assure sulfur retention

Figure 5-1. Dual Fuel Burner for the Stirling Engine

Figure 5-2. Atmospheric Fluidized Bed Coal Combustor for Stirling Engine Applications: Preliminary Schematic

- Greater temperature differences between the heat exchanger tube surfaces and bed can exist
- Turndown ratios of up to 6 to 1 can be achieved.

A schematic of the flow diagram is shown in Figure 5-3. For the proposed atmospheric fluidized bed combustion system, a cyclone, combustion air preheater, and bag house are required. The overall dimensions of the system are 6.7 by 1.83 meters.

5.2 Heat Transfer System

For indirect heat transfer, two methods of transferring heat to the hot portion of the Stirling engine were investigated. The first method, high pressure gas circulation loop, is more appropriate for the state-of-the-art engine designs. The second method, heat pipes, requires development and is more appropriate for the advanced engine design. In Figure 5-4 is shown a schematic of the heat pipe arrangement for a Stirling engine mated with an atmospheric fluidized bed combustion system. The working fluid is sodium and there are 22 bed tubes per bundle.

5.3 Preliminary System Configuration

In Figure 5-5 is presented a schematic view of a small multi-fueled Stirling engine generator set with a combustion system for coal-derived gases and liquids or low sulfur hydrocarbon fuels. The system is modular with each modular consisting of five cylinders, each developing 100 hp for a total of 500 hp. There are five heater heads, combustors, and preheaters. There is a blower for exhaust gas recirculation which is used to preheat the air and vaporize the fuel for efficient combustion after premixing. Also, there is a diesel type crankcase which includes crosshead and seal modules. The overall system dimensions of the engine module are 2 by 3.5 meters. Of course for a total output of 3000 hp, six engine modules are required.

Figure 5-3. Schematic Flowsheet: Atmospheric Fluidized Bed Coal Combustor for Stirling Engine

Figure 5-4. Heat Pipe Arrangement for Coal Fired Engine (Viking Design)

Figure 5-5. Schematic View of Small Multi-Fuel Stirling Engine Generator-Set for Coal-Derived Clean Gases and Liquids or Current Low-Sulfur HC Fuels of Choice

In Figure 5-6 is shown a schematic view of a large, Stirling engine with an atmospheric fluidized bed coal combustion system with indirect heat transfer. The engine consists of 6 modules, each developing 500 hp for a total engine output of 3000 hp. There is an airblower and an air preheater for the combustion system. The indirect heat transfer method is pressurized inert gas circulation loop which is driven by a pump. The overall system dimensions are 14 by 7 meters.

For both systems the engine salient features are:

- Engine design includes an integrated heater head/heat exchanger which is easily adaptable for efficient coal combustion
- Monolithic heater head offers high performance and high reliability for large Stirling engines
- In-line, double acting engine is adaptable to different methods of capacity control for various duty cycle requirements
- Modular construction.

Figure 5-6. Schematic View of Large Stirling Engine With Fluidized Bed Coal Combustion and Indirect Heat Transfer

6. CONCLUSIONS AND REMAINING WORK

The conclusions developed to date include the following:

- For successful penetration into the integrated energy systems market, coal derived fuels must be used to assure fuel availability and low life cycle costs
- It appears that Stirling engines employing clean fossil fuels cannot compete with diesel engines
- Combustion technology exists for the successful burning of coal derived fuels in a large stationary Stirling engine
- Current fluidized bed combustion technology is adaptable for coal burning Stirling engines
- The development of large stationary Stirling engines adapting existing technology for burning coal derived fuels should be completed
- The development team offers a broad range of experience in the development of combustion engines and related machinery, a major manufacturer of diesel and specialty engines, and consultants having extensive Stirling engine experience
- The modular design for the Stirling engine power section and output machinery (generic to diesel) is interchangeable with either option of coal derived fuels or coal combustion

- Large Stirling engines with high thermal efficiency may be extrapolated from previous small Stirling engines
- Modular Stirling engine "cylinder" design can be mated with current diesel crankcase, generator set practice to assure near term penetration into a low volume market
- Stirling engine thermodynamic performance is not strongly dependent on the overall mechanical configuration but rather on the heat transfer system
- The analysis, confirmed with comparison to actual engine data, predicts high thermal efficiency for a design tailored to stationary power generation.

Remaining work consists of the thermodynamic performance evaluation by means of the Sunpower third order analysis and the manufacturing assessment and production cost assessment of the conceptual engine designs.

REFERENCES

1. Marciniak, T. J., Bratio, J. C., David, A., and Lee, C., An Assessment of Stirling Engine Potential in Total and Integrated Energy Systems, Argonne National Laboratory, Argonne, Illinois, ANL/ES-6, February 1979.
2. Rios, P. A., An Analytical and Experimental Investigation of the Stirling Cycle, ScD. Dissertation, Massachusetts Institute of Technology, Cambridge, MA, 1969.

FOSTER-MILLER ASSOCIATES, INC.
ENGINEERS
350 SECOND AVE.
WALHAM, MA 02154
617 890-3200/TWX 710 324-1468

AN EQUAL OPPORTUNITY EMPLOYER.