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DESIGN AND DEVELOPMENT OF STIRLING ENGINES FOR
STATIONARY POWER GENERATION APPLICATIONS IN THE
500 TO 3000 HP RANGE

Subtask 1A Report, State-of-the-Art Conceptual Design

MASTER

March 1, 1980

Work Performed Under Contract No. AC02-79ET15209

General Electric Company
Advanced Energy Department
Space Division
Valley Forge Space Center
Philadelphia, Pennsylvania

U. S. DEPARTMENT OF ENERGY

Division of Industrial Energy Conservation

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LARGE STATIONARY STIRLING ENGINE

DESIGN AND DEVELOPMENT OF STIRLING ENGINES
FOR

STATIONARY POWER GENERATION APPLICATIONS IN THE 500 TO 3000 HP RANGE

CONTRACT NO. DE-AC02-79ET15209

SUBTASK 1A REPORT

MARCH 1, 1980

STATE-OF-THE-ART CONCEPTUAL DESIGN

PREPARED FOR:

U.S. DEPARTMENT OF ENERGY

WASHINGTON, D.C.

GENERAL ELECTRIC

ADVANCED ENERGY DEPT.

SPACE DIVISION

Valley Forge Space Center

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TABLE OF CONTENTS

<u>SECTION</u>		<u>PAGE</u>
1	SUMMARY AND OVERVIEW	1
2	STATE-OF-THE-ART CONCEPTUAL DESIGN	6
2.1	Task Definition and Design Requirements	6
2.2	System Description - Schematic	7
2.3	Engine Configuration Selection Summary	11
2.4	Conceptual Design of Candidate Systems	19
2.4.1	Overall Design Philosophy and Basic Engine Characteristics	19
2.4.2	V Engine with Crank Drive	23
2.4.2.1	V Engine Description Summary	24
2.4.2.1.1	Articulated V Engine	24
2.4.2.1.2	Non-Articulated V Engine	26
2.4.2.1.3	Generator Mounting	34
2.4.2.1.4	Accessories	34
2.4.2.1.5	Modularization	34
2.4.2.2	Cylinder Design	37
2.4.2.3	Seal Design	44
2.4.2.4	Heater Head Heat Exchanger	51
2.4.2.4.1	Direct (Gas) Fired	51
2.4.2.4.2	Heat Pipe Heat Exchanger	51
2.4.2.4.3	Pressurized Gas Heat Transport Heat Exchanger	52
2.4.2.5	Regenerator and Cooler	53
2.4.3.6	Balancing, Drive System and Crosshead	58 62
2.4.3	Radial Engine	63
2.4.3.1	Engine Description	63
2.4.3.2	Radial Modularity	64
2.4.4	Swashplate Engine Configuration	68
2.4.4.1	Engine Description	68
2.4.4.2	Drive System Elements	68
2.4.4.3	Modular Approach	70
2.5	Heat Transport Loop	77
2.5.1	Heat Pipe System	77

TABLE OF CONTENTS (Continued)

<u>SECTION</u>	<u>PAGE</u>
2.5.1.1 Overall System Description	77
2.5.1.2 Heat Pipe Description	79
2.5.1.3 Operating Characteristics	82
2.5.2 Pressurized Gas Heat Transport System	83
2.5.2.1 Overall System Description	83
2.5.2.2 Pressurized Gas Loop	83
2.5.2.3 Gas Circulator	86
2.5.3 Insulation and Support	90
2.6 Heat Source Interface Definition	92
2.6.1 Coal Fired Systems	92
2.6.2 Oil Fired System	99
2.6.3 Gas Fired System	99
2.7 Auxiliaries	100
2.7.1 Cooling Loop	100
2.7.2 Lubrication System	101
2.7.3 Helium Supply and Recovery Systems	101
2.8 Control System	103
2.9 Electrical System	107
2.9.1 Generator Starter	107
2.9.2 Switching and Control	107
2.10 Materials	109
2.11 Speed Selection	113
3 SYSTEM EVALUATION AND COMPARISON	114
3.1 Evaluation Criteria and Raging Methodology	114
3.1.1 Concept Evaluation Methodology	114
3.1.2 Levelized Annual Cost	119
3.1.3 Output of Economic Calculation	120
3.2 Performance	121
3.3 Cost Data	124
3.4 Levelized Annual Cost of Energy	145
3.5 Candidate Configuration Selection	146
3.5.1 Engine Cost Comparison	146
3.5.2 Comparison of Qualitative Criteria	147

TABLE OF CONTENTS (Continued)

<u>SECTION</u>	<u>PAGE</u>
3.5.2.1 Non-Articulated V Engine	147
3.5.2.2 Articulated V Engine	147
3.5.2.3 Radial Engine	148
3.5.2.4 Swashplate Engine	148
3.5.2.5 Rhombic Drive Engine	149
3.5.2.6 Heat Transport Loop Rating	149
4 SYSTEM SELECTION AND RECOMMENDATIONS	152
APPENDICES	153
A PIR U-SSE-79-001, LSSE Concept Comparison and Selection	153
B PIR U-SSE-79-002, Evaluation Criteria and Method of Economic Analysis For LSSE	175
C PIR U-SSE-79-003 Materials for LSSE	199
D Foster Wheeler Task 1A.1 Report	234
E Foster Wheeler Task 1A.3 Report	313

LIST OF ILLUSTRATIONS

FIGURE	TITLE	PAGE
1-1	Selected State of the Art 1985 Demo Engine, 1000 HP-V8	3
2.2-1	System Design Requirements	6
2.2-2	Overall Energy Balance	9
2.2-3	System Schematic	10
2.3-1	Configuration Parameters	12
2.3-2	Configuration Parameter Summary	13
2.3-3	Cylinder Arrangement	14
2.3-4	Cylinder Design	15
2.3-5	Heat Transport System	16
2.3-6	Candidate System Selection	17
2.3-7	Preliminary, Relative Cost Rating of Candidate Configuration	18
2.4.1-1	Design Approach Philosophy	21
2.4.1-2	Engine Characteristics	22
2.4.2-1	V Engine-Articulated	27
2.4.2-2	Articulated Stirling	30
2.4.2-3	Articulated Connecting Rod Riria Engine Efficiency Due to Non-Sinusoidal Piston Motion	31
2.4.2-4	Articulated Crank Loads	32
2.4.2-5	V Non-Articulated	33
2.4.2-6	V Modularization	36
2.4.2-7	Maximum Hoop Stress vs. Cylinder Wall Thickness	39
4.2-8	Conventional Cylinder Design	40

LIST OF ILLUSTRATIONS

FIGURE	TITLE	PAGE
2.4.2-9	Double Wall Cylinder	41
2.4.2-10	Water Cooled Cylinder	42
2.4.2-11	Pumping (Appendix) Losses Due to Dual Cylinder Wall Design	43
2.4.2-12	Scraper Seal (or Pumping Ring)	48
2.4.2-13	United Stirling Sliding Seal	47
2.4.2-14	Ford/N. V. Philips Sliding Seal	48
2.4.2-15	Piston Rod Seal Concept	49
2.4.2-16	Cylinder Seal System with Oil Eliminator	50
2.4.2-17	Engine HX and Regenerator Design	53
2.4.2-18	Direct Fired Heater Head Parameters	54
2.4.2-19	Heater Head Design Parameters	55
2.4.2-20	Heat Pipe Transport Loops	56
2.4.2-21	Heater Head Heat Exchanger for Pressurized Gas Heat Transport Loop	57
2.4.2-22	Regenerator Parameters	59
2.4.2-23	Cooler Parameters	60
2.4.2-24	Gas Fired Heater Head	61
2.4.3-1	Radial Engine	65
2.4.3-2	Radial Engine-Cycle Forces	66
2.4.3-3	Radial Modularization	67
2.4.4-1	Swashplate Engine	72
2.4.4-2	Swashplate Engine with Journal Bearings	73
2.4.4-3	Swashplate Slider Bearing Load and Plate Support Total Load	74

LIST OF ILLUSTRATIONS

FIGURE	TITLE	PAGE
2.4.4-4	Swashplate Engine Pressure vs. Phase Angle	75
2.4.4-5	Swashplate Engine Modularization	76
2.5.1-1	Heat Pipe Installation	78
2.5.1-2	Heat Pipe Assembly	80
2.5.1-3	Heat Pipe Characteristics	81
2.5.2-1	Pressurized Heat Transport System	84
2.5.2-2	Heat Transport Loop	85
2.5.2-3	Heat Transport Loop Piping Loss	87
2.5.2-4	Pressurized Gas Transport Loop Tubing	88
2.5.2-5	Heat Transport Loop Pumping Power	89
2.5.3-1	Typically Available Fibrous Insulation	91
2.6.1-1	Combustion Systems Summary Chart	93
2.6.1-2	Approximate Distribution of Cost of Combustion System/KW as a Function of Capacity	98
2.8-1	Pressure Control	104
3.1-1	Qualitative Evaluation Criteria	115
3.1-2	Qualitative Evaluation Criteria Contributing Factors	116
3.1-3	Weighting Factors	118
3.2-1	Performance Summary Engine Design Point	122
3.2-2	Stirling Engine Performance Summary	123
3.3-1	Typical Price Program Input Sheet	125
3.3-2	Typical Weight Calculation Sheet	126
3.3-3	Engine Weight Summary V-NA-NM	127

LIST OF ILLUSTRATIONS

FIGURE	TITLE	PAGE
3.3-4	Engine Weight Summary V-NA-Modular	128
3.3-5	Engine Weight Summary V-Art-NM	129
3.3-6	Engine Weight Summary V-Art-Modular	130
3.3-7	Engine Weight Summary Radial	131
3.3-8	Engine Weight Summary Swashplate	132
3.3-9	Typical Price Program Printout	133
3.3-10	Cost and Weight Comparison for a 1000 HP Unit	143
3.3-11	Specific Weight Cost-Non-Articulated V Engine	144
3.5-1	Engine Weight and Cost Data	150
3.5-2	Qualitative Evaluation Results	151

1.0 SUMMARY AND OVERVIEW

This report summarizes the first portion of the Conceptual Design of Stirling Engines for Stationary Power Application in the 500 to 3000 HP range which was aimed at state-of-the-art stationary Stirling engines for a 1985 hardware demonstration. (Task 1A)

The main goals of this effort were to obtain reliable cost data for a Stationary Stirling engine capable of meeting future needs for total energy/cogeneration systems and to establish a pragmatic and conservative base design for a first generation hardware.

In order to secure a competitive position for this engine in the marketplace, it was clearly recognized that a coal and waste product fired version had to be given primary consideration since a fuel oil or gas fired version would not be cost competitive with a diesel/gas engine. Therefore, the mating of the Stirling engine with an Atmospheric Fluidized Bed (AFB) was a main target.

Since cost effectiveness (or ROI) for total energy/cogeneration systems depends - besides the initial capital investment - mainly on the total operating time per year (high capacity factor), a main objective of the design was not only low cost, but high reliability and good maintainability with engine efficiency being an important but not main consideration.

Starting with an extensive screening effort, four (4) engine types and three (3) heat transport systems were selected:

<u>ENGINE TYPES</u>	<u>HEAT TRANSPORT SYSTEMS</u>
● V-type crank engine	● Heat pipe
● Radial Engine	● Pressurized gas heat transport loop
● Swashplate engine	● Direct gas fired system
● Rhombic drive engine	

After a preliminary layout cycle, the rhombic drive engine was eliminated due to intolerable maintenance difficulties on the push rod seals. "V", Radial and Swashplate engines were taken through a detailed design/layout cycle, to establish all important design features and reliable engine weights, including all auxiliaries.

A costing program - RCA's "PRICE 84" - was employed to establish engine costs (including all engineering development and testing costs) as a function of engine size and yearly production rate ranging from 10 to 10,000.

A summary of this weight and cost analysis is given in Table 3.3. Specific weights range from 9.4 lb/hp (radial) to 12.5 lb/hp (sw: plate) and specific costs were obtained between 230 \$/hp* (308 \$/KW) and 297 \$/hp* (398 \$/KW) for a 1000 HP engine at a production rate of 1000 engines per year. (Comparative data for a 2000 HP locomotive diesel engine: 13.5 lb/hp and 117 \$/hp or 157 \$/KW).

It has to be noted that a quite conservative approach was deliberately chosen for the first generation hardware:

- Helium instead of hydrogen as working fluid - mainly for market acceptability.
- 100 atmospheres mean pressure level instead of 200 ATM now being considered in automotive Stirling engines.
- 1200 RPM engine speed - with a good change for later upgrading to 1800 RPM.
- Bearings and crankshafts consistent with high peak load diesel design practices rather than higher allowables for the more even load distribution on Stirling engines.

After comparing engine layouts and analyzing qualitative and quantitative evaluation criteria, the V-crank engine (Figure 1-1) was chosen as the candidate for a 1985 hardware demonstration. This choice is substantiated by the following advantages:

- State-of-the-art drive with considerable experience and established reliability from large diesel engines for the entire drive train and auxiliary system.
- Existing production facilities (forging, heat treating, machining, grinding and balancing).
- Existing skills for design, assembly and maintenance, no development required for the drive system, crankcase, auxiliaries.
- Wide application range and interface flexibility.
- Good modularity.
- Good component accessibility for easy maintenance.
- Enhanced market acceptability due to familiarity with components and configuration.

The radial engine design with four cylinders (90°) in one plane is ideally suited for a Rinja type engine, has good modularity and excellent weight and cost characteristics (approximately 20% lower than "V" engines). This engine was not chosen since maintainability

* Including all auxiliaries but not the Heat Transport Loop.

considerations make a vertical shaft mandatory which curtails application flexibility and reduces an already lower market acceptance. However, this engine might be a good supplement to an extended production program for second or third generation hardware.

The swashplate engine, even though compact in design, did not exhibit the same weight and cost advantages as the radial engine with considerable reduction in drive system maintainability (limited access to critical bearings). Furthermore, the development of a large swashplate drive would introduce a considerable risk to program cost and schedule.

The proposed "V" engine design incorporates the following novel and unique features:

- Individually mounted water cooled cylinders with integrated seal cartridge for easy exchange and maintenance.
- Seal system without catastrophic failure modes which can tolerate considerable leakage to minimize forced outages.
- Exchangeable heat exchangers, regenerators and coolers with low temperature, non-welded interface joints for easy maintenance and heat source adaptability.
- Interchangeable heat transport systems for fuel flexibility and later fuel conversion capability.
- Extensive use of low cost material through novel design approaches for cylinder, regenerator/cooler and heat transport loop manifold design.
- Novel dead volume control system.
- Integrated single bearing generator for ease of site installation.
- Easy adaptability for advanced hardware and retrofit applications.

The chosen design represents a conservative approach with a high degree of confidence for the viability of a demonstration and first generation hardware.

In an evolutionary process, later generation hardware may be upgraded to more than twice the power level/per cylinder with minor modification in engine hardware by introducing higher pressure levels, 1800 RPM engine speed and hydrogen as working fluid.

Figure 1-1. Selected State-of-the-Art 1985 Demo Engine 1000 HP-V8

For a successful engine demonstration by 1985, the following areas require further attention:

- Reliable seals for this size of engine are not state-of-the-art and further design and development testing is needed before a final design can be chosen.
- Reliable heat source cost data are not available making an economic assessment difficult. Economically viable designs for the total energy/cogeneration market have to be established which fall in the desired output range and address problems such as modularity, turn-down ratio and coal handling equipment for small outputs.
- A heat transport loop is a necessary component to achieve good fuel and system flexibility.

Heat pipes of the required size (250 KW thermal) are not state-of-the-art and exhibit certain operational problems which are not sufficiently resolved.

A pressurized gas heat transport loop suffers from the non-availability of suitable high temperature gas circulators at a pressure level of approximately 40 atmospheres. Furthermore, such a system adds approximately 60 \$/hp to the overall engine cost (vs. 25 \$/hp for the heat pipe system) and has an auxiliary power consumption of approximately 5% of engine shaft power.

A considerable design and development effort is required to close the gap to a truly state-of-the-art heat transport system equipment.

The degree of detail in which this conceptual design was carried out was only possible with the close cooperation and help from many experienced contributors:

- GE-Locomotive Operation contributed many design details taken from existing locomotive diesel hardware, assisted in many valuable reviews and provided all team members a first hand familiarization with similar type hardware, including assembly and test operations.
- GE-Energy Technology operation in Schenectady, NY is responsible for the economic analysis using an existing computer program modified for the specific needs of this program.
- Foster Wheeler Development Corporation contributed a valuable heat source selection study and interface definition as well as conceptual layout for a typical AFB heat source.
- Dr. R. Meijer and B. Ziph - N.V. Philips - provided Stirling component design guidance.
- Thermacore, Inc. assisted in heat pipe definition.

2. STATE OF THE ART CONCEPTUAL DESIGN

2.1 Task Definition and Design Requirements

The task's goal as defined by the work statement is "to identify and evaluate State of the Art large Stationary Stirling Engine concepts and recommend one or more of these concepts for prototype development which have the greatest potential for prototype testing and demonstration by 1985".

The governing requirements and specifications were given in Sections 3.1 and 3.2 of enclosure 2 to DOE RFP NO.

DE-RP02-79 ET 15337. S000 and are summarized in table 2.1-1.

1. GENERAL REQUIREMENTS

- DESIGN FOR FUEL FLEXIBILITY

- Oil
- Gas
- Coal - Conventional and Fluidized Bed

- EMPHASIZE

- Reliability
- Maintainability
- Cost Competitiveness

2. SPECIFIC REQUIREMENTS

Power Range	500 to 3000 HP
Hot Side Temperature	760°C (1400° F)
Coolant Inlet Temperature	80°C Design Point
Shaft Efficiency	20 to 120°C Operating Range ≈ 40%
LIFE	10,000 Between Overhauls (HRS)
SPEED	Compatible With 60 HZ Generator
CONTROLS	For Remote Operation

TABLE 2.1-1 SYSTEM DESIGN REQUIREMENTS

2.2 SYSTEM DESCRIPTION--SCHEMATIC

The Large Stationary Stirling Engine system is shown schematically in figure 2.2-3. The sub-systems to support the engine are discussed individually in subsequent sections of this report, but a brief overview will be presented here.

Heat is delivered from the combustion system (which is not part of this conceptual design) via a heat transport system. Two transport schemes are considered in section 2.5, one being a heat pipe system which requires no electrical power, the second is a pressurized gas system using helium or air which requires auxiliary power for the electrically powered gas circulator.

The only direct fired system considered is a gas fired system (section 2.6-3) where the combustion gases directly heat the heater head heat exchanger.

Heat is transferred to the engine in a separate and exchangeable heater head heat exchanger described in section 2.4.2.4.

The engine is cooled by circulating water which removes heat not only from the cycle coolers but also from a lubricating oil cooler and from the cylinder heads. The coolant and oil pumps are driven by the engine and together constitute less than 1% of the heat supplied to the engine. The coolant loop contains an economizer and a process heat exchanger, as well as an auxiliary cooling heat dump to be used as a backup for the process heat exchanger.

The generator is mechanically driven by the engine, and generates electric power which is deliverable with the exception of two electrically powered parasitic systems; the coal preparation and combustion system, and the heat transport system circulator (not necessary for heat pipe system). As shown in the overall energy balance, figure 2.2-2, the net output is either 33 or 30.5% of the total system input.

The generator losses are fed back into system by using the combustion air as generator cooling medium.

The lubricating oil system consists of an engine shaft driven pump and a heat exchanger; it supplies oil to engine bearings and crossheads and to a seal oil circuit. The seal oil loop contains a second high pressure pump to raise the oil pressure to mean cycle pressure, and a filter which supplies oil to all of the cylinder cycle seals. The seal system also contains a helium and oil separation system, which includes a passive "scrubber". The helium is then fed back into the cycle and the oil is returned to the lubrication system sump. (Section 2.4.2.3)

A helium charging system provides working fluid to the engine through a pressure regulator. The control systems described in section 2.8 are integrated with this system.

Engine startup is achieved by motoring the generator, a system which has proven successful with other Stirling engines. No separate gear mesh or motor is necessary, only the proper power and phasing controller to accelerate the generator.

FIG 2.2-2 OVERALL ENERGY BALANCE

STIRLING COGENERATION/TOTAL ENERGY SYSTEMS
CAN PROVIDE EFFICIENT POWER PRODUCTION FOR
UNITS BELOW 2MW

FIG 2.2-3 SYSTEM SCHEMATIC

LSSE SYSTEM SCHEMATIC

USE OF HEAT TRANSPORT LOOP
RESULTS IN HEAT SOURCE FLEX-
IBILITY AND ENHANCES MAINTAIN-
ABILITY

2.3 ENGINE CONFIGURATION SELECTION SUMMARY

The Configuration Selection is discussed in detail in PIR'S USSE-79-001 and 79-002 which are attached to this report. Therefore only the summary charts will be presented here.

Table 2.3-1 summarizes the configuration parameters considered. These parameters are compared with respect to their advantages and disadvantages for a Stationary Stirling Engine in tables 2.3-2 to 2.3-5.

Evaluating these characteristics a preliminary configuration section was made shown in table 2.3-6.

After preliminary layouts were made of the selected engine configurations a preliminary cost rating was made in order to concentrate the effort on the more promising configurations. This relative rating is summarized in table 2.3-7, and is based on the articulated V configuration which was arbitrarily used as a cost standard (=1.0) in all cost categories: development, production, operating, maintenance and installation cost. In order to arrive at a total cost factor which was related to the annual levelized cost of energy a weighting factor for each cost category was used consistent with the contribution of the specific cost to the cost of energy. This weight factor is shown in the first column of table 2.3-7. As can be seen from the overall results there was no clear drop out from the cost of energy standpoint even though the V and radial engine showed some advantage.

Therefore, all engine configurations except the opposed rhombic drive engine were taken into a final design cycle.

The opposed rhombic drive engine had been originally selected with respect to the possibility of larger output per cylinder (250 HP or 500 HP) because it was not tied to a Rinja configuration whereby necessity the cylinder's output is one quarter of the base (smallest) module output. Since 500 HP was the selected module size to comfortably span the 500 HP to 3000 HP range the cylinder size for all Rinja type engines was 125 HP. Therefore, it was felt that the rhombic drive engine with 250 HP or, 500 HP per cylinder would be a viable contender despite its more complex drive and seal system.

After the preliminary layout, however, it became apparent that the seal maintenance on this type engine was unacceptable and therefore this engine configuration was eliminated from further design effort.

Table 2.3-1 Configuration Parameters

TYPE OF DRIVE

- 1.1 Crank
- 1.2 Swashplate
- 1.3 Rhombic
- 1.4 Free Piston

ARRANGEMENT OF CYLINDERS

- 2.1 In-Line
- 2.2 V-Type
- 2.3 Opposed (V-angle 180°)
- 2.4 Radial
- 2.5 U (2 rows parallel, 2 drives)
- 2.6 Circular
- 2.7 W-configuration
- 2.8 Inverted W

CYLINDER DESIGN

- 3.1 Single Acting
- 3.2 Double Acting

MODULE ARRANGEMENT

- 4.1 In-Line
- 4.2 Double Row
- 4.3 Radial
- 4.4 Circular

HEAT TRANSPORT LOOP

- 5.1 Heat Pipe
- 5.2 Reflux Boiler
- 5.3 pumped Liquid Metal Loop
- 5.4 Pressurized Gas
- 5.5 Direct
- 5.6 Radiation

TABLE 2.3-2 CONFIGURATION PARAMETER SUMMARY

1. DRIVE TRAIN

CONFIGURATION	ADVANTAGES	DISADVANTAGES	COMMENTS
<u>CRANK</u>	<ul style="list-style-type: none"> • Cost Effective • High Mechanical Efficiency • Easy Maintenance • <u>Well Developed</u> • <u>Existing Production Facilities</u> • Trained Skills Available 	<ul style="list-style-type: none"> • Vibration 	<p>Lowest Cost and Schedule Risk.</p> <p><u>Good market acceptability.</u></p>
<u>SWASHPLATE</u>	<ul style="list-style-type: none"> • Compact • Low Vibration • Promising Concept for Advanced Design 	<ul style="list-style-type: none"> • <u>Not state-of-the-art for large engines</u> • Limited access for maintenance • Lower mechanical efficiency • Auxiliary take-offs difficult • Module arrangement (no through drive) 	<p><u>Market acceptability uncertain.</u></p> <p>Difficult to obtain prior experience on large engines.</p>
<u>RHOMBIC</u>	<ul style="list-style-type: none"> • Reduced No. of Cylinders • Low Vibration • No Cross-heads 	<ul style="list-style-type: none"> • Mechanical complexity • Double seal • Seal maintenance difficult 	<p><u>Maintainability of seal major problem.</u></p> <p>Smaller modules may require flywheel.</p>
<u>FREE PISTON</u>	NOT CONSIDERED FOR LARGE ENGINES		Not a viable concept

TABLE 2.3-3 CYLINDER ARRANGEMENT

CONFIGURATION	ADVANTAGES	DISADVANTAGES	COMMENTS
<u>IN-LINE</u>	<ul style="list-style-type: none"> • Good Accessibility • Easy Maintenance 	<ul style="list-style-type: none"> • <u>Excessive Length</u> (block and foundation) • Low torsional stiffness of crankshaft • Integration to heat source more difficult (due to length) 	<p>Practical only for engines not exceeding 12 cylinders.</p> <p>Less viable for Rinia configuration.</p>
<u>"V"</u>	<ul style="list-style-type: none"> • Compact • Good Accessibility • Low Specific Weight • Easy Maintenance 	<ul style="list-style-type: none"> • Cylinder interconnection somewhat longer 	
<u>OPPOSED</u>	<ul style="list-style-type: none"> • Same as "V" 	<ul style="list-style-type: none"> • <u>long cylinder interconnections</u> • Integration to heat source more difficult 	<p>Heat transfer loop may obstruct crank case access.</p>
<u>RADIAL</u>	<ul style="list-style-type: none"> • Short and Compact • Low specific weight • Excellent modularity • Well suited for Rinia configuration 	<ul style="list-style-type: none"> • Integration to heat source more difficult • Limited access to drive 	<p><u>For ease of maintenance, vertical axis mandatory.</u></p> <p>Special foundation support structure required.</p> <p><u>Market acceptability uncertain</u></p>
<u>"U"</u>	<ul style="list-style-type: none"> • Ease of Maintenance • Multiple Auxiliary take-off • Short Cylinder interconnection • Balancing benefits 	<ul style="list-style-type: none"> • Double crankshaft • Connecting gear drive in reverse loading • Bulky block 	<p><u>Advantages only where multiple heater head integration is desirable</u></p>
<u>CIRCULAR</u>	<ul style="list-style-type: none"> • Compactness • Short Cylinder interconnections 	<ul style="list-style-type: none"> • Limited access • Auxiliary drive take-off difficult 	<p>Used on swashplate drive only</p>
<u>"W" or INVERTED "W"</u>	<ul style="list-style-type: none"> • Close Proximity of Cylinders 	<ul style="list-style-type: none"> • Double or triple crankshaft • Bulky • <u>Limited access for maintenance</u> 	<p>Not considered a viable configuration for maintenance reasons.</p>

TABLE 2.3-4- CYLINDER DESIGN

CONFIGURATION	ADVANTAGES	DISADVANTAGES	COMMENTS
<u>SINGLE ACTING</u>		<ul style="list-style-type: none"> • Greater number of seals • Increased engine volume 	Used for rhombic drive only Requires bounce space.
<u>DOUBLE ACTING</u>	<ul style="list-style-type: none"> • Compact Design • <u>Only one rod seal for two cylinder spaces</u> 		

4. MODULE ARRANGEMENT

<u>IN-LINE</u>	<ul style="list-style-type: none"> • <u>Ease of Integration with Heat Source and Generator</u> 	<ul style="list-style-type: none"> • Overall Length 	Either bolt-on or integral block.
<u>DOUBLE ROW</u>	<ul style="list-style-type: none"> • Overall Length Reduced • Reduced Torque on Crank-shaft 	<ul style="list-style-type: none"> • More difficult to integrate with heat source • <u>Gear box required for single generator</u> 	
<u>RADIAL/ CIRCULAR</u>	<ul style="list-style-type: none"> • Small Foundation 	<ul style="list-style-type: none"> • <u>Requires special gear box for single generator</u> 	Circular arrangement considered for swashplate engines only.

TABLE 2.3-5 HEAT TRANSPORT SYSTEM

CONFIGURATION	ADVANTAGES	DISADVANTAGES	COMMENTS
<u>HEAT PIPE</u>	<ul style="list-style-type: none"> • Low ΔT • Small Liquid Metal Inventory • <u>No auxiliary power required</u> 	<ul style="list-style-type: none"> • Slight Safety Problem • Thermal decoupling from heat source difficult • <u>Exchangeability difficult</u> • <u>No field repair possible</u> • Vapor pressure of Na above atmospheric for $T > 1640^\circ\text{F}$ 	<p><u>Required size not quite state-of-the-art.</u></p> <p>Gas controlled pipe may be required for ease of maintenance.</p>
<u>REFLUX BOILER</u>	<ul style="list-style-type: none"> • Low ΔT • No auxiliary power 	<ul style="list-style-type: none"> • <u>Safety problem due to increased liquid metal inventory</u> • Significant thermal stress in evaporator during transients 	<p>Reflux boiler principle would be used in heater head HX only.</p>
<u>PUMPED LIQUID METAL</u>	NOT CONSIDERED DUE TO SAFETY PROBLEM WITH PERSONNEL AND HARDWARE		<u>Low market acceptability.</u>
<u>PRESSURIZED HELIUM</u>	<ul style="list-style-type: none"> • Safe operation • Good temperature control • <u>Easy to decouple from heat source</u> or engine switch to redundant heat source. • Good maintainability • Relative positioning to heat source less critical 	<ul style="list-style-type: none"> • <u>Auxiliary power required</u> • Increased ΔT • <u>High temp. gas circulator required</u> • Lower heat transfer rates • Increased He consumption • Heavy wall piping 	<p>May require multiple circuits per engine for increased reliability (He leak!)</p>
<u>RADIATION</u>	<ul style="list-style-type: none"> • Simplicity 	<ul style="list-style-type: none"> • Requires close proximity to heat source • Low heat flux • <u>Requires high source temp.</u> 	<p>Only viable for special applications.</p>
<u>DIRECT</u>	<ul style="list-style-type: none"> • Simplicity 	<ul style="list-style-type: none"> • <u>Contamination of heater tubes</u> • Low heat transfer rates • Close spacing requirements to heat source 	<p>Considered only for gas fired units.</p>

TABLE 2.3-6 CANDIDATE SYSTEM SELECTION

SYSTEM	1	2	3	4	5	6
<u>TYPE OF DRIVE</u>						
Crank	X	X	X			X
Swashplate				X		
Rhombic					X	
Free Piston						
<u>ARRANGEMENT OF CYLINDERS</u>						
In-Line						
V-Type	X	X	X			
Opposed (V-angle 180°)					X	
Radial						X
U (2 rows parallel, 2 drives)						
Circular				X		
W-Configuration						
Inverted W						
<u>CYLINDER DESIGN</u>						
Single Acting					X	
Double Acting	X	X	X	X		X
<u>MODULE ARRANGEMENT</u>						
In-Line	X	X	X		X	X
Double Row						
Radial						
Circular				X		
<u>HEAT TRANSPORT LOOP</u>						
Heat Pipe	X					
Reflux Boiler						
Pumped Liquid Metal Loop						
Pressurized Gas		X				
Direct (Gas Fired)			X			
Radiation						

TABLE 2.3-7 PRELIMINARY, RELATIVE COST RATING OF CANDIDATE CONFIGURATIONS

COST	* WEIGHT FACTOR	V-ARTICULATED	V-NON-ARTICULATED	SWASHPLATE-JOURNAL BRG.	SWASHPLATE-N.F. BRG.	RADIAL	OPPOSED RHOMBIC	He-TRANSPORT LOOP	HEAT PIPE
DEVELOPMENT	.01	1 1 .01	1 1 .01	1.7 1.7 .017	1.6 1.6 .016	1.1 1.1 .011	1.5 1.5 .015	1 1 .01	1.5 1.5 .015
PRODUCTION	.24	1 1 .24	.98 1 .235	1.2 1.2 .288	1.3 1.3 .312	9 5 1 .24	1.3 1.3 .312	1 1 .24	1.2 1.2 .288
OPERATION	.61	1 1 .61	1 1 .61	1.05 1.05 .64	1.05 1.05 .64	1 1 .61	1 1 .61		
MAINTENANCE	.09	1 1 .09	.95 1 .0855	1.4 1.4 .126	1.5 1.5 .135	1.1 1.1 .099	1.6 1.6 .144	1 1 .09	1.0 1.0 .09
INSTALLATION	.05	1 1 .05	1 1 .05	1.2 1.2 .06	1.2 1.2 .06	1.2 1.2 .06	1.2 1.2 .06	1 1 .05	.8 .8 .04
		1.0	.990	1.131	1.163	1.068	1.141	.39	.433
RANKING		2	1	4	6	3	5		

LEGEND:

RELATIVE RATING	COST FACTOR
--------------------	----------------

* BASED ON CONTRIBUTION TO ANNUAL
LEVELIZED COST OF ENERGY

2.4 CONCEPTUAL DESIGN OF CANDIDATE CONFIGURATION

2.4.1 Overall Design Philosophy and Basic Engine Characteristics

The introduction of a cogeneration system is only viable for engines which have high reliability (availability factor > 90%) and are used a large fraction of the time (> 60%). This dictates high reliability combined with a fast and easy replacement of parts, i.e. good accessibility and interchangeability of all components. Therefore, much emphasis was placed on high reliability and good maintainability since earlier economic studies (ETAS) have clearly indicated that the total operating time is a more important economic parameter for a total energy/cogeneration system than performance.

When oil or gas is used as fuel, diesel engines will most likely outrank the Stirling engine with respect to the return on investment (or levelized cost of energy) due to the increased cost for the heat source integration complexity. Therefore, the most viable application for large stationary Stirling engines will be a coal or refuse fired version where the reduction in fuel costs more than offsets the higher hardware and installation costs. Therefore, a prime consideration in the design approach was the integration to a coal fired heat source since this is considered the most viable path for the commercialization of large stationary Stirling engines.

A summary of groundrules used for the conceptual design are given in Table 2.4.1-1.

Based on these groundrules a detailed conceptual design was made for the following engines:

1. 90°V crank articulated *
2. 90°V crank non-articulated **
3. 90°Radial
4. Swashplate--4 cylinders
 - a. with journal bearing
 - b. with N.F.--bearings

Utilized cylinders and seal cartridges were the same for all engines all heat-exchangers, regenerators and coolers are of the same design.

As Base-Module size, a 500 HP unit was chosen, which results in a cylinder size of 125 HP for the selected Rinja configuration--same for all engines.

In order to access the module interface design all layouts were based on a 2 Module (1000 HP) design, except for the swashplate engine where a 4 cylinder and 8 cylinder alternate approach was considered.

The choice of helium as working fluid was not based on performance or other trade off analysis but solely on safety/market acceptability considerations for the first generation engines. In the preliminary

* In plane cylinders, two different connecting rods

** Offset cylinders equal connecting rods, same crank

design effort final engine parameters will be selected such that later switchover to hydrogen can be accommodated with only a minor hardware modification such that hydrogen can be introduced at a later time for advanced configurations in an evolutionary development process.

Other common engine characteristics are summarized in Table 2.4.1-2.

TABLE 2.4.1-1.

DESIGN APPROACH PHILOSOPHY

1. CONCENTRATE ON COAL FIRED VERSION (BUT: FIRST TEST UNIT MAY BE GAS FIRED)
2. RELIABILITY MOST IMPORTANT PARAMETER (WE HAVE TO BE EQUAL TO DIESEL ON-LINE AVAILABILITY $\approx$ 95%)
 - SIMPLICITY
 - USE AS MANY PROVEN TECHNOLOGIES AND DESIGNS AS POSSIBLE
3. GOOD MAINTAINABILITY
 - ALL PARTS INTERCHANGEABLE & EASILY ACCESSIBLE
 - AS FEW DIFFERENT PARTS AS POSSIBLE (SPARE PART INVENTORY)
 - MOST REPAIRS SHOULD TAKE LESS THAN ONE DAY
 - FAST SEAL EXCHANGE
4. COST EFFECTIVE DESIGN (ROI)
 - SIMPLE DESIGN - EASE OF MANUFACTURING
 - LOW COST MATERIALS
 - AS FEW ON-SITE OPERATIONS AS POSSIBLE
 - USE OF AVAILABLE PRODUCTION FACILITIES
5. GOOD MARKET AVAILABILITY
 - NO REAL OR PSYCHOLOGICAL SAFETY HAZARDS
 - SIMPLE INSTALLATION REQUIREMENTS
 - LOW SKILL LEVEL FOR INSTALLATION & MAINTENANCE
 - FOLLOW CODES OR USE AS GUIDELINE
 - SIMPLE SIZE VARIATION (500 HP INCREMENT)

TABLE 2.4.1-2.
ENGINE CHARACTERISTICS

POWER - 125 HORSEPOWER PER CYLINDER

STROKE - 4.00 INCHES

BORE - 7.00 INCHES

DISPLACEMENT - 153.94 CU. IN. PER CYLINDER

1.23 CU. IN. PER HORSEPOWER

SPEED - 1200 RPM

WORKING FLUID - HELIUM

PRESSURE - 100 ATMOSPHERES MEAN,
APPROXIMATELY 135 MAXIMUM,
65 MINIMUM

ENGINE TYPE - RINIA

PHASE ANGLE - 90°

2.4.2 V ENGINE WITH CRANK DRIVE

The V engine configuration has gained acceptance through-out the engine industry as being an extremely successful approach to engine design. Aside from the obvious advantage of being shaped to provide an ideal interface with the automobile engine space allocation, the V engine provides a very low specific weight crankcase and drive, very good dynamic balancing potential, excellent servicing access, convenient take-off for auxiliaries, a stiff drive due to short crankshaft and a lower number of main bearings than in-line engines require.

Specifically for the Large Stationary Stirling Engine, the V configuration offers the advantages of having overall dimensions well suited for AFB heat source interfacing, being compatible with the 90° phase angle required for the Rinia configuration, having cylinders in close proximity to allow Rinia interconnections and being similar in size and design to the GE locomotive diesel engine and thus allowing use of existing design and manufacturing technologies as well as facilities.

The Erie facility presently produces several hundred diesel engines per year each having either eight, twelve or sixteen cylinders in a V configuration, with approximately 250 HP per cylinder. These engines are used primarily in locomotive applications, although they also are finding use in marine and stationary applications. The GE diesel engine for locomotive use is designed to idle at 450 rpm, and have a full load speed of between 900 and 1200 rpm, the latter exactly matching the selected LSSE design point. Overall sizes for the diesel are also similar to the conceptual Stirling design, with a height of seven feet, width of six feet and lengths of eleven, thirteen or sixteen feet for the various number of cylinders.

Several design features which have been incorporated into the LSSE design are a direct result of the diesel engine experience:

- o Interchangeable individual cylinder modules, easy maintenance and exchange.
- o No water or cycle gas contact occurs with the main frame, thus helping to eliminate corrosion and its effects from the primary structural member.
- o Shaft driven oil and water pumps.
- o Combination quick removal access and pressure relief doors sized to provide ease of maintenance are located on both sides of the frame.
- o Lubricating oil passages are cast directly into the frame with occasional cleaning ports located for ease of servicing.
- o Single bearing flanged generator.

Many other design features have been implemented as suggested by key engineering personnel from the Erie locomotive facility, and will be identified in the following description sections.

2.4.2.1 V ENGINE DESCRIPTION SUMMARY

The V engines selected for this conceptual design phase are of two basic types, one having an articulated drive as shown in figure 2.4.2-1, the other having individual, identical connecting rods for each cylinder. The merits of each will be discussed in the applicable section.

2.4.2.1.1 ARTICULATED V ENGINE

The GE locomotive diesel engine incorporates this approach, which permits the use of the minimum length crankshaft and crankcase by allowing the pairs of cylinders to be in the same plane. The other advantage to in-plane cylinders is the reduction of axial imbalance couples. Some redistribution of connecting rod loads occurs thus modifying the crankshaft bearing load profile.

As with any slider crank mechanism, the motion imposed on the piston is not purely sinusoidal since the distance between the crank pin and the wrist pin is fixed, and the spacing along the cylinder axis varies as the leg of a triangle having the connecting rod as the hypotenuse, and the perpendicular offset of the crankpin as the second leg. The result of this effect for the master rod in this configuration is identical to a standard nonarticulated engine, while for the articulated piston the effect is amplified by the exaggerated crank end offset as well as the shorter hypotenuse (connecting rod). As a result of these variations in displacement versus crank angle, the acceleration effects produce second, fourth and higher even numbered harmonic dynamic inputs to the engine and result in some noise and vibration. Additionally, the thermodynamic cycle is affected by the displacement variations which also result in a slight phase shift as shown in figure 2.4.2-2. Since the Rinia approach leads to expansion and compression spaces from different cylinders being used in combination, the result of this effect will be a variation in power output between separate cycles. Figure 2.4.2-2 presents a plot of motion variations between a sinusoidal stroke and the articulated motion of the LSSE. The plot of figure 2.4.2-3 indicates the reduction in output of Stirling cycle with the LSSE articulation from a purely sinusoidal drive cycle and is the result of a first order computer program developed specifically for the LSSE evaluation.

Parametric studies indicate that the best performance is achieved when the master rod bore for the articulated wrist pin is located at the same angle as the V (90°). The effect on dynamic loads on the crankshaft and other drive elements is small for the geometry selected and is considered negligible.

Figure 2.4.2-4 defines the total forces on the drive system bearings, and includes cycle as well as acceleration forces. The individual rod loads represent the axial force in the rod, while the total crank bearing load defines the maximum radial force on that bearing.

The V engine shown in figure 2.4.2-1 indicates the use of an articulated drive system and shows the relatively large cross section of the master connecting rod. The removal of the master rod from the top side of the engine offers the advantage of not requiring oil pan removal for rod replacement, a major time saver in field servicing. In order to remove the articulated master rod, the crosshead diameter for this version was increased to nine inches compared with seven inches required for side load capability. The result is an increase in crosshead weight, acceleration loads and overall engine size. (The nonarticulated V engine uses a 7" crosshead as discussed in section 2.4.2.1-2).

The use of a crosshead in all of the LSSE engine configurations results from the need for a seal between the crankcase and the compression space, with no lubrication existing above the seal. The drive imposes significant lateral loads on the piston (or crosshead) and in order to react these loads either a crosshead is used or the piston rod can react these loads on a bearing surface below the seal. Because of the magnitudes of these loads, the crosshead approach is the only viable solution since the piston rod would become prohibitively large if it were sized to react the load in a sleeve bearing. The crosshead imposes a relatively large initial load penalty on the engine drive, but enhances servicing by providing a piston rod joint and a bore adequately large to permit connecting rod removal.

The seal cartridge and water cooled cylinder are attached to the crankcase with four bolts; an access hole through the crosshead bore provides access to the piston rod joint in the crosshead, allowing rapid removal of the piston, cylinders and seal cartridge without any further system disassembly.

One major difference between the drive for a diesel and a Stirling engine is that the diesel cycle load is primarily unidirectional, whereas the Rinja design provides reversing loads as indicated in figure 2.4.2-4. As a result of this, the articulated design is better fitted with a diesel, since the articulated connecting rod bearing is basically asymmetrical and favors a larger bearing surface area in the direction toward the crankshaft.

Inputs from the GE diesel engineering personnel also tend to favor the nonarticulated approach, even suggesting that if a new diesel were to be designed, the use of a non-articulated approach would be recommended.

2.4.2.1-2 NON-ARTICULATED V ENGINE

Two design approaches are available to implement a nonarticulated V engine. One utilizes a block with cylinders in common axial planes and combinations of standard connecting rods and forked rods which straddle them. The second approach is the common automobile engine design, where cylinders are slightly offset from each other in the axial direction, and all connecting rods and bearings are identical. The latter simplifies the design and servicing capability, and the offset for the LSSE engine is only 2.25 inches.

Fig 2.4.2-5 shows a partial cross/section for this engine, only showing features which are different from the articulated V engine.

The crosshead diameter is made identical to the cylinder bore, seven inches, and allows adequate clearance for connecting rod removal through the top. The same type of piston rod to crosshead joint is implemented to permit removal of the piston along with the seal cartridge and cylinder after removing four bolts. Access to the crankshaft is available through the quick removal doors on the side of the crankcase, and hence connecting rods and even main bearings can be replaced without oil-pan removal or crankcase disassembly.

FIG 2.4.2-1
V ENGINE - ARTICULATED

ADVANTAGES

- o LOW OVERALL LENGTH
- o IN PLANE (AXIAL) CYCLINDERS REDUCE IMBALANCE COUPLE
- o ALLOWS "BOLT ON " MODULARITY
- o COMPATIBLE WITH BOLT TOGETHER CONFIGURATIONS FOR EXTENDED OUTPUT

DISADVANTAGES

- o HIGHER HARMONIC IMBALANCE DUE TO ARTICULATED CRANK
- o COMPLEX CONNECTING ROD DESIGN
- o LARGER CROSSHEAD DIAMETER

FIG 2.4.2-1 CONT'D.
V- ENGINE ARTICULATED

FIG 2.4.2-1 (Con't.)
V ENGINE - ARTICULATED
PARTS LIST

1. Heat Pipes
2. Engine Heater Head
3. Regenerator
4. Cooler
5. Hot Side Manifold
6. Cold side Manifold
7. Water Cooled Cylinder
8. Piston
9. Seal Cartridge
10. Crossed
11. Master Connecting Rod
12. Articulated Connecting Rod
13. Connecting Rod Bearing
14. Balance Weight
15. Main Frame (Crank case)
16. Access Port
17. Balance Shaft
18. Coolant Pump
19. Oil Pan
20. Generator
21. Generator Shaft Joint

FIG 2.4.2-2

ARTICULATED STIRLING

90° PHASE ANGLE, 4" STROKE
13.285" PRIMARY CON ROD LG
8.9555" ARTICULATED CON ROD LG
4.90" THROW OFFSET @ 90°

FIG 2.4.2-3 ARTICULATED CONNECTING ROD RINIA ENGINE EFFICIENCY
DECREASE DUE TO NON SINUSOIDAL PISTON MOTION
90° PHASE ANGLE

FIG 2.4.2-4

ARTICULATED CRANK LOADS

FIG 2.4.2-5 VEE - NON ARTICULATED

ADVANTAGES

- SMALLER NUMBER OF PARTS (SPARES)
- IMPROVED BALANCING
- SMALLER CROSSHEAD DIAMETER

DISADVANTAGES

- INCREASED OVERALL LENGTH
- STAGGERED CYLINDER ARRANGEMENT

2.4.2.1.3 GENERATOR MOUNTING

The approach used on the GE locomotive diesel engine is quit appropriate for LSSE generators also. The last main bearing on the engine crankshaft serves as the forward bearing for the single bearing generator. The advantages to this approach are the elimination of a flexible coupling between the two shafts and the repeatability of the mounting for field replacement. Elimination of the field alignment eases installation and decreases foundation size requirements. The generator tolerances permit replacement using original shims without additional alignment verifications.

Initial alignment is achieved by bolting the flanged shafts together, and rotating one full revolution while monitoring deflection of the end crank throw. Required shim thicknesses and location are determined from this deflection measurement.

2.4.2.1.4 ACCESSORIES

The lubricating oil pump and water pump are located on the forward face of the engine with gear drives used to connect them to the crankshaft. Similar to the GE diesel approach, an additional crankshaft take-off is provided for specific applications where shaft power is required. The use of gear drives also allows the pump drive speed to be varied to permit optimization or utilization of an existing pump.

The lubricating oil passages are primarily located in the crankcase, but where external runs are required, such as for seal scavenge oil return, the use of tubing and mechanical fittings will be utilized.

Water lines must connect to each cylinder and cooler from a common manifold, and non-metallic hoses can be implemented for this relatively low temperature, low pressure application to enhance servicing.

2.4.2.1.5 MODULARIZATION

The required power range for the LSSE program is 500 to 3000 horsepower. Thus a natural size for a base module is a 500 HP unit.

Several approaches for modularization have been investigated and are discussed below. Which method will be finally chosen is subject to a more finalized design with market requirements more firmly established.

1. The present approach for the GE locomotive diesel modularization is a dedicated block and crankshaft (for 8, 12 a

16 cylinders) with all other hardware being common (except for the size of water and oil pumps). This results in the smallest and lighter engine and also is a very cost effective approach provided that the production quantity in each power range is large enough to warrant dedicated block and crankshaft sizes. This approach is shown in fig 2.4.2-6 as "non-modular".

2. The use of a four cylinder module which has a main frame and crankshaft designed to mate with adjacent units or auxiliaries allows more commonality of parts and provides a higher degree of system flexibility. The initial design and manufacturing cost is reduced since only one type of crankshaft and main frame is required, and spare part stocking is simplified. Production costs are further reduced by the increased number of identical parts (accelerated learning curve). This might be a most cost effective approach for the first generation engines.
3. Each four cylinder (500 HP) module can be equipped with its own generator. The result for larger systems is an increase in base cost due to high combined costs as a function of redundancy of all supporting systems, and higher combined generator and switchgear costs. Also, the heat source interfacing could become more difficult because of increased space requirements. Even though this approach increases the system flexibility, its increased cost and installation complexity makes it a less attractive version.
4. Groups of engines could be interfaced with a gearbox driving a single generator. The substantial gearbox cost (approximately \$30.00 HP) and resulting inefficiencies as well as engine spacing are major draw-backs. A slight advantage is the use of a speed increasing gearbox with an 1800 rpm generator at somewhat reduced cost.

This approach appears attractive only for special applications, where mechanical power outputs above 3000 HP are required.

FIG 2.4.2-6
VEE MODULARIZATION

- FLEXIBLE DESIGN
- MINIMAL SPARE PARTS

ARTICULATED
 MODULAR V-12

- SAME AS ABOVE

NON-ARTICULATED
 MODULAR V-12

- SHORTEST LENGTH
- LIGHTEST WEIGHT

NON-MODULAR V-12

2.4.2.2 CYLINDER DESIGN

The cylinders for Stirling engines have characteristically been designed as thick walled pressure vessels designed to maintain the cycle pressure while subjected to the hot side gas temperature. To minimize conduction losses, the thickness can be tailored to the temperature profile, but the mass of material remains quite high and the use of expensive, high temperature materials is the rule. Other factors, such as thermal stresses and thermal inertia (increased servicing time), become increasingly important on large engines as well. Figure 2.4.2-7 indicates the insensitivity of pressure induced hoop stress to wall thickness, and figure 2.4.2-8 shows the design approach utilized on most smaller Stirling engines. In designs of this type the governing parameter is generally creep or creep rupture failure in the high pressure area. The water cooling jacket near the base maintains a relatively cool surface for the piston seal to ride against, and connections at the hot end are generally welded or brazed.

Figure 2.4.2-9 shows a preliminary approach solving some of the previously mentioned problems, wherein a separate liner and outer shell are used, separated by a layer of insulating material. The insulation space is maintained at mean system pressure via small passages at the cold end interface of outer and inner cylinder, thus the hot inner wall is subjected to cyclic pressure stresses only, the outer shell is thermally isolated and carries the differential between mean pressure and the external environment. The fabrication of a cylinder of this design permits the use of low cost materials, since the hot wall is subjected to relatively low stress levels, and there is little corrosion potential on the hot wall as well.

The expansion joint shown in figure 2.4.2-9 compensates for the differential thermal and pressure expansions between the two walls. Main disadvantages of this approach are the difficulty to install the insulation and high temperature on the heat changer interface flange. To overcome these disadvantages a water cooled cylinder was chosen shown in figure 2.4.2.10. The design uses a thin inner liner which is perforated and communicates with a small gap filled with metal screen or ceramic material. This results in no pressure differential across the inner hot liner, with the full pressure being reacted by the water cooled outer cylinder. The increased heat loss (approximately 2%), used for process heat, is well compensated by decreased material costs (approximately \$16.00/hp). The cooling water also circulates under the hot side flange, and the liner extends past the flange surface, reducing the temperature at the seal interface to enhance the joint reliability. An axial expansion joint in the inner liner accommodates differential growths, and simplifies assembly.

The amount of material used in this approach is lower due to the elimination of all loads from the hot liner and also because of the small diameter and low temperature of the pressure shell. The configuration of the pressure shell is quite simple and would lend itself well to high production fabrication techniques, including ductile cast iron.

The gap between inner and outer cylinders is an appendix loss and therefore the free volume of this gap was to be kept small. An estimate of these losses as a function of gap size is shown in figure 2.4.2-11.

FIG. 2.4.2-7

MAXIMUM HOOP STRESS VS. CYLINDER WALL THICKNESS

PRESSURE = 14.70 PSI
MARGIN OF SAFETY = 2.25
INSIDE DIAMETER = 7.4"

FIG 2.4.2-8 CONVENTIONAL CYLINDER DESIGN

ADVANTAGES

- SIMPLE (MINDED) DESIGN

DISADVANTAGES

- HIGH COST OF MATERIALS
(MULTIMET $\approx$ \$12/#)
- EXCESSIVE THERMAL STRESS
- HIGH TEMPERATURE HX INTER-
FACE SEAL
- WEIGHT

FOR INCREASED COST EFFECTIVENESS
DESIGN HAS TO ACCOMODATE LOW COST MATERIALS

FIG 2.4.2-9 DOUBLE WALL CYLINDER

ADVANTAGES

- LOW COST MATERIALS
(316 L \$1.50/#)
- REDUCED WEIGHT

DISADVANTAGES

- HOT LINER EXPOSED TO + 600 psi
PRESSURE OSCILLATIONS
- HIGH TEMPERATURE HX INTERFACE
SEAL
- INSTALLATION OF INSULATION
DIFFICULT

FIG 2.4.2-10 WATER COOLED CYLINDER

ADVANTAGES

- LOW COST MATERIAL EVENTUALLY DUCTILE CAST IRON
- MINIMUM WEIGHT
- LOW TEMPERATURE HX INTER-FACE SEAL (INCREASED RELIABILITY)
- NO CYCLIC PRESSURE LOADS ON LINE

DISADVANTAGES

- INCREASED HEAT LOSS
- APPENDIX LOSSES

INCREASED THERMAL LOSSES WELL
COMPENSATED BY CAPITAL COST REDUCTION

FIG 2.4.2-11 PUMPING (APPENDIX) LOSSES DUE
TO DUAL CYLINDER WALL DESIGN

2.4.2.3 Seal Design

The function of the rod seal in a Stirling engine is to contain the working fluid in the cylinder by preventing leakage into the crankcase, as well as to preclude the migration of crankcase contained fluids into the working space. The problem is complicated by the fluctuating high pressure of the working fluid, by slight lateral translation of the rod during the stroke, by the long life requirements under reciprocating conditions and by the sensitivity of the engine hot side to relatively small quantities of crankcase fluids.

Two basic approaches have been followed in the past; a positive seal consisting of a rolling diaphragm element and any of several forms of rubbing or sliding seals. The rolling element or rollsock seal has been pursued by N.V. Philips, with some test results exceeding 10,000 hours of life, but many failures at shorter times prevent the seal from being credited with a reliability which would allow it to be used in this state of the art program. Factors such as debris in the seal area can drastically reduce seal life, and extreme care and high skill levels are required for installation and servicing.

Additional complications arise from the use of a pressurized oil backup system to reduce the tensile stress on the diaphragm. Thus this seal system is complex and vulnerable and is prone for sudden catastrophic failures unsuitable for stationary cogeneration systems.

Sliding seals fall into many categories, and combinations of types can be selected to provide specific sealing requirements. Gland or compression packings exert a mechanical force between the surfaces to be sealed and generally provide an adjustment which distinguishes them from the molded packings, including lip type and squeeze type seals. These seals function by providing one or more surfaces in contact with the reciprocating member, and in general can be fabricated from materials which permit them to function without supplementary lubrication.

In addition to these seal types, a metallic scraper or pumping ring has been developed, and depending on its orientation, can function in either capacity. This device consists of a metal ring with the inner bore slightly tapered with the smaller end being nearly equal to the shaft diameter as shown in Figure 2.4.2-12. If oil is drawn into the bevel side, hydrodynamic forces cause the ring to be slightly expanded and oil is dragged through the gap. If oil exists on the side opposite the bevel, the scraping action of the sharp lip precludes the flow of oil into the ring. In practice, these seals can be used individually to serve either purpose, but are often used in opposed sets to maintain a reservoir of pressurized oil between them. The pumping ring is placed facing the oil volume of the engine and pumps oil into the cavity between it and the scraper ring, which acts as a termination of the intermediate reservoir. This approach has been used successfully as a method of eliminating the oil pump in rollsock seal systems, and pressures of over 1000 atmospheres have been reported.

SLIDING SEAL SYSTEMS

Some information has been released on at least two systems which have been developed and tested in Stirling engines. The United Stirling seal is shown in Figure 2.4.2-13 and consists of an upper self-lubricated molded seal whose purpose is to significantly mitigate the pressure excursions within the cycle, but not to serve as a positive seal. This "Kapseal" has been developed by N.V. Philips and exhaustively tested. Below this seal is a chamber which is pressurized with working fluid and contains a compression spring to provide axial force to the lower seal cartridges. These consist of a metal scraper seal to impede oil transport into the chamber, and a working fluid seal element below an intermediate pressure tap. This element passes some oil and some working fluid, which are drawn off using the pressure tap, filtered and the working fluid is returned to the engine space. As reported in the October, 1979 U.S. DOE Highway Vehicle Systems Contractors Coordination Meeting, efforts are underway to improve the oil sealing function of this seal concept. The problem reported is oil migration along the rod to the upper seal, and one method of eliminating this problem is being developed. It consists of a rubber diaphragm between the rod and seal housing and also prevents wear debris from settling onto the lower seals. Testing of the system must include the vulnerability of the diaphragm, increased assembly and maintenance time, and difficulty in determining when and if an internal failure has occurred (diaphragm leakage).

The second sliding seal system is the Ford/N.V. Philips seal, which contains four sealing elements. From the working fluid side, the first seal is a cap seal (may be same as Kapseal mentioned previously) to provide relatively constant pressure in the adjacent spring chamber. The next seal is a scraper seal of unorthodox configuration which is loaded by the axial spring. The primary sliding seal is next, and it uses the wedging action of the spring as well as the pressure drop it maintains to enhance rod preload. The fourth and last seal is a typical scraper ring which is adjacent to an oil filled cavity which provides rod cooling by introducing a flow of oil which bleeds out through a lower rod gap. An oil and working fluid outlet tap is provided in the area of the primary sliding seal. Pressure in that area is maintained just above the minimum cycle pressure by means of a working fluid check valve. The fluids recovered from the seal are processed through an oil separator and then the working fluid through a demister to condense any oil vapor or mist remaining. The oil collects in a float relieved reservoir which returns it to the crankcase. Figure 2.4.2-14 shows this seal.

PROPOSED LSSE SEAL

For a stationary engine application, the design of the seal must consider not only seal reliability, but degraded mode operation in order to minimize forced outages and servicing of the seal and engine as well. Since the size is significantly larger than engines for which existing seals have been designed, some unique design problems (such as stroke length and significant rod lateral motion) exist.

so, due to the large power output per cylinder, the losses encountered in accepting a finite leakage through the seal system may become acceptable, permitting some variation in seal design.

The design shown in Figure 2.4.2-15 is proposed as a conceptual approach to meeting the LSSE requirements. The design consists of a Kapseal (or equivalent) at the working fluid end, again used to eliminate the pressure wave variations from the internals of the seal cartridges. A small buffer volume is next in line with a gas inlet and a labyrinth seal following. This labyrinth seal serves as a series back-up to the Kapseal for isolating pressure variations and may also serve as a substitute for a finite period of time (at higher power losses) until a failed Kapseal can be replaced. Also, the labyrinth assists in preventing oil migration into the working space by providing high gas velocities relative to the rod (especially in the upstroke direction when transport is most likely to occur) to produce a scrubbing action. The labyrinth is positioned relative to the rod by means of a floating ring to accommodate rod eccentricities. Below the seal is a chamber of adequate length to assure that the portion of the rod that contacts oil in the lower seals does not enter the labyrinth area.

A fluid removal line is used to extract any working fluid (either leaked past the Kapseal or added above the labyrinth to promote flow) and oil that may exist in the volume. This mixture of helium and oil vapor or mist is processed through an oil mist/vapor separator shown in Figure 2.4.2-16 and Section 2.7.3 and returned to the working space and oil reservoir respectively.

The next and final seal is a scraper seal, also in a laterally floating housing. This scraper removes most oil from the rod before it is carried into the seal cavity. Immediately below the scraper is a cavity into which oil is pumped from a remote source to provide cooling to the rod and also to reduce the pressure drop across the seal and consequently reduce working fluid loss to the crankcase. The fit of the outlet with respect to the rod provides the desired internal pressure and adequate oil flow. The seals are housed in a cartridge to facilitate replacement or servicing, and a lower flange provides access to the internal chambers via ports. The pressure induced axial force is carried through the flange and reacted in the external flange bolts.

Approximate power consumption predictions for this seal system are 0.32 horsepower for cooling oil pumping and .5 horsepower for labyrinth seal flow pumping, using a seal chamber pressure of 88 atmospheres. Main objective of this seal design approach is to avoid forced outages by providing a capability to tolerate seal leakage, without shutdown. Gas flow indicators in individual cylinder return lines can monitor the state of the seal and thus provide information on necessary scheduled maintenance.

Figure 2.4.2-13--United Stirling Sliding Seal

Figure 2.4.2-12 Scraper Seal (or Pumping Ring)

SLIDING SEAL CARTRIDGE CROSS SECTION

Figure 2.4.2-14 Ford/N.V. Philips Sliding Seal

FIGURE 2.4.2-15

PISTON ROD SEAL CONCEPT

Figure 2.4.2-16

CYLINDER SEAL SYSTEM
WITH OIL ELIMINATOR

2.4 Heater Head Heat Exchanger

The purpose of the heater head heat exchanger is to exchange heat from the external heat supply or transport system to the Stirling engines working fluid. The engine's heat demand is approximately 850,000 BTU's per hour per cylinder; a small temperature drop across the exchanger serves to minimize the transport system temperature and material selection difficulty and to obtain high system efficiency. Further constraints are imposed on the exchanger design by the regenerator interface and by the engine working fluid pressure drop, which also directly affects engine performance. Limits of void volume and flow symmetry also influence the design.

Three variations are considered in this design, one each for the direct gas fired engine, the pressurized gas heat transport system and the heat pipe system. In all cases, the required internal heat transfer between the tube walls and the pulsating engine working fluid (helium) is identical, but because of variations in external heat transfer the exchanger designs are different. A summary of important HX parameters is given in Table 2.4.2-17.

2.4.2.4.1 Direct (Gas) Fired

The conceptual design selected for this application is a gaseous fuel fired combustor with either a radiation cooled porous plate burner or a ribbon type plate burner. The heater head is a nested finned tube arrangement and is integrated with the combustor as shown in Figure 2.4.2-24. The Advanced Energy Department has a well developed and demonstrated capability to design and fabricate this type of system for Stirling engines, a technology developed in the ongoing Activated Heat Pump Program, sponsored jointly by the American Gas Association and DOE. The key attributes of this design approach are the uniform flow and heat distribution which are available, the simply implemented inlet air preheater and a relatively low cost and compact package. Figure 2.4.2-18 defines the parameters of the engine heater tubes required to meet the necessary heat exchange level, as a function of the working fluid pressure drop. The data are based on the use of fins on the outside of the tubes to enhance heat transfer from the combustion products to the tubes. The fins selected for use have a 0.50 inch diameter, are 0.040 inch thick and are spaced with a 0.051 inch gap. The configuration selected for the conceptual design is compatible with the Vee engine cylinder spacing of twelve inches between centerlines, which dictated the maximum exchanger outside diameter. The heat exchanger consists of 50 tubes having a 0.25 inch inside diameter and an active heat transfer (finned) length of 22 inches. The overall length of each tube will be approximately 29 inches, from regenerator to manifold. The tube bank is surrounded by a convoluted counterflow air preheater with a exit temperature of approximately 800°C (425°C).

2.4.2.4.2 Heat Pipe Heat Exchanger

The exchanger design for the heat pipe heat transport system consists of an array of tubes which transport the working fluid internally condensing sodium on the outside.

Figure 2.4.2-19 presents the design parameters and the selected design of 300 tubes with an inside diameter of 0.13 inches and a length of 15.25 inches. The resulting pressure drop is about 4 psi and the helium volume is 61.0 cubic inches. The heat transfer in this design is limited by the internal tube wall heat transfer coefficient. The selected configuration is shown in Figure 2.4.2-20.

2.4.2.4.3 Pressurized Gas Heat Transport Heat Exchanger

The engine heater head heat exchanger for the pressurized gas heat transport system has been designed to be compatible with the regenerator configuration selected. The design for the heat pipe version has been used as a baseline, having 300 tubes with an inside diameter of 0.130 inches and a length of 15.00 inches. Calculations indicate that for this pressurized gas system the length should be increased to 15.25 inches to provide the proper level of exchange on the tube inside wall. The configuration selected has an axial flow tube shell heat exchanger, providing an adequate performance with reasonable levels of pumping power for both helium and air heat transport systems. As shown in Figure 2.4.2.21 circumferential manifolds are used at each end of the exchanger to provide proper flow distribution.

TABLE 2.4.2-17

ENGINE HX AND REGENERATOR DESIGN

● HEATER HEAD

	<u>HEAT PIPE</u>	<u>FOSSIL FUEL COMBUSTOR</u>
- NO. OF TUBES/CYL.	300	50
- TUBE I.D.	.15"	.25"
- ACTIVE HEAT TRANSFER LENGTH	15"	21"

● REGENERATOR

- NO./CYLINDER	1
- LENGTH	2.5"
- DIAMETER	5.5"
- FILLING FACTOR	0.20
- WIRE SIZE	.005"

● COOLER

- NO. OF TUBES/CYL	600
- TUBE I.D.	.07"
- ACTIVE HEAT TRANSFER LENGTH	5"

FIG. 2.4.2-18
DIRECT FIRED HEATER HEAD PARAMETERS

FIG 2.4.2.19

HEATER HEAD DESIGN PARAMETER

FIG 2.4.2-20 HEAT PIPE TRANSPORT LOOP

FIG 2.4.2-21 HEATERHEAD HEAT EXCHANGER FOR PRESSURIZED GAS HEAT TRANSPORT

2.4.2.5 REGENERATOR AND COOLER

The regenerator design for a large engine can not simply be "scaled-up" from small size Stirling engines, since the pressure drop is proportional to the regenerator length. Consequently, the diameter of a single regenerator per cylinder has to be selected according to the heater head and cooler diameter. The regenerator matrix material selected is randomly stacked .005 stainless steel fiber having a porosity of 80% based on previous regenerator design experience. Figure 2.4.2-22 indicates the design point used to provide a 97% efficiency with a 5.5 inch diameter unit. The corresponding pressure drop is approximately .5 PSI and the void volume is approximately 50 cubic inches.

The cooler design selected for this conceptual design is the conventional tube-shell design, with helium flowing inside the tubes while water is circulated on the outside surfaces. Figure 2.4.2-23 indicates the required five inch long cooler design to provide a transfer of 550,000 BTU per hour from cylinder to the coolant water (assumed to be at 80°C). A selection of 600 tubes provides a pressure drop of 9 psi, and permits use of a 0.069 diameter (inside) tube. Since the heat transfer coefficient from the helium to the tube is low (approximately 37% of the water side) a method of enhancing the transfer and thus reducing the required number of tubes has been developed and successfully implemented in the gas fired heat pump engine. A wire spring coil is placed in each tube to alter the helium flow, and thus enhance the heat transfer at the same pressure drop (9 psi). The number of tubes is reduced from 600 to 360, and the internal tube diameter is increased to 0.124 inches. The void volume is increased from 11.2 to 21.7 cubic inches, however, this has no impact on the engine performance. This approach will be further pursued during the preliminary design for increased cost effectiveness. The housing for the regenerator cooler has been configured to reduce the severity of thermal stresses in the wall of the regenerator section. Conventional regenerators are surrounded by a pressure tight structural wall, but the relatively small diameter permits a thin section to be used and results in a fairly compliant section which reduces the effects of the extremely high axial temperature gradients. The 5.5 inch inside diameter of the LSSE regenerator would result in a very thick wall of rather expensive material if it were designed to resist pressure induced creep rupture failure for the life of the engine, and the resulting rigidity would create high thermal stresses. The approach shown in figure 2.4.2-20 permits the use of low cost materials by isolating the load carrying members from the high temperature gas. The regenerator is surrounded by a thin cylinder which is free to slide axially at one end, and provides only a partial seal. The external cavity remains near mean cycle pressure, such that the load carrying wall is relatively long and thus the axial gradient effect is significantly reduced.

FIG 2.4.2-22

REGENERATOR PARAMETERS
ONE PER CYLINDER
.005" STACKED FIBER
80% POROSITY

FIG 2.4.2-23 COOLER

PARAMETERS

TUBE LENGTH = 5"
TUBE-SHELL HX
H₂O FLOW = 22.6 GPM

FIGURE 2.4.2-24
GAS FIRED HEATER HEAD

2.4.2.6 BALANCING, DRIVE SYSTEM AND CROSSHEAD

The V engine is comprised of slider crank mechanisms operating with a fixed phasing between them. Each cylinder generates inertial forces by virtue of the reciprocating motion imposed by the crank mechanism. The piston and crosshead are being accelerated along the cylinder axis, resulting in acceleration forces occurring at the same frequency as the rotational speed, as well as forces at even multiples of the frequency due to a deviation from pure sinusoidal motion as a result of the slider crank drive. These secondary forces increase inversely with connecting rod length, and are increased by the use of articulated connecting rods. The general approach to balancing used in reciprocating engines will be applied to the Stirling engine, with crankshaft counterweights being used to counterbalance the rotating mass of the crankshaft and part of the connecting rod weight. Connecting rods are counterweighted at the extremes to provide a torque balancing (assuming constant rotational speed). By virtue of the use of a 90° angle, the secondary forces and moments for most of the configurations selected will be inherently balanced, with the exception of the four cylinder V engines which will contain counter rotating balance shafts operating at twice the crankshaft speed. Since the magnitude of the harmonic imbalances decreases quite rapidly with increasing order, the effects of the fourth and higher orders are normally considered negligible for reasonable connecting rod length to stroke ratios. The GE locomotive diesel engine contains balancing shafts for their shortest block, due in part to the articulated drive and also the use of a 60° V angle, which results in a secondary shaking force in the eight cylinder configuration. The articulated drive creates a slight phase shift in the even numbered harmonics also, and results in a slight residual imbalance even with active balancing implemented.

A torsional damper of the viscous variety has been shown in use on the Stirling engine. This is in keeping with the general use of these devices to assist in internal distribution of the output torque of the engine to produce a reasonable fatigue stress level on the crankshaft. The torque is fully reacted on the frame, between the engine and generator housings, and is not reflected to the engine mount.

Since the heat transport to the engine is conducted in tubes of fairly sizeable cross sections, it is necessary to minimize the excursion of the engines due to exciting forces (imbalance or others). This reduces the strain imposed on the transport systems, and reduces the flexibility and consequently the complexity of the interfaces. The engine should be fairly rigid relative to the facility in order to obtain these goals, and the proper level of balancing is critical to permit this to be feasible.

2.4.3 RADIAL ENGINE

2.4.3.1 Engine Description

The radial engine shown in Figure 2.4.3-1 is an excellent match for the Rinia system, having the required 90° phase angle between adjacent cylinders. The four cylinders lie in a common plane, and the crankcase is very simple and was the lowest weight per cylinder of all engines investigated. The access doors between cylinder mounting faces permit servicing and inspection of the drive elements, while removal of the piston rod from the crosshead for cylinder exchange is accomplished through each crosshead sleeve.

The drive consists of one master rod whose axis always passes through the wrist pin and crankshaft lobe axis, and three articulated rods. The piston motion induced by the master rod deviates slightly from sinusoidal travel, as does any slider crank mechanism motion, while the articulated rod piston displays a more prominent deviation. The two pistons at 90° to the master rod have a slight phase shift and total travel deviation similar to that of the articulated Vee engine, while the piston opposite the master rod has only the mid cycle variation, although more pronounced. The effects of these variations are twofold; first, the thermodynamic cycle is affected by the variations in displacement versus crankangle, and secondly the piston and crosshead inertial forces are varied. The first results in slightly reduced output, and also in slight mismatch between the cycles in the four cylinder engine. The second effect creates second and higher even numbered harmonic imbalances. The severity of these effects is obviously related to the length of the connecting rods, with longer length reducing their magnitude. The radial connecting rods are somewhat longer than those employed on the Vee engine, for reasons of crankcase access, and it is felt that the engine performance anomalies will be practically negligible. The crankshaft of the radial engine is also very light in weight, due to its short length, and it also can be made to be very stiff. Primary force balancing can be achieved using a pair of counterweights opposite the crank offset.

The drive system bearings have been sized in accordance with the sum of cycle forces as presented in Figure 2.4.3-1 and inertial forces. The inertial loads of opposed cylinders are additive at the crank bearing, while cycle forces tend to be opposite signs due to their 180° phase relationship. The master rod crank bearing load plot indicates this effect by the fairly constant maximum radial cycle load on the crank bearing plot.

The master connecting rod houses the crankpin bearing as well as the three articulated rod bearings, and redistributes the imposed loads. As a consequence, the rod end is relatively large and would result in a large crosshead diameter for easy removal of crosshead and connecting rod. Therefore spacers are used under the caps to permit the crosshead bore to be sized for load bearing capability only.

The removal of cylinders and pistons for servicing can be viewed as a problem for the lower cylinder (or cylinders) on a horizontal shaft mounted engine; consequently a vertical shaft installation is almost mandatory. This is fortified by an evaluation of the heat transport system interface, which indicates a large vertical distance between banks of heat transport tubes as well as significant crowding. In the vertical shaft configuration the elevation span is significantly reduced, and due to the vertical spacing of cylinders the transport tubing is less restricted. The generator can be mounted on the upper face of the engine to simplify removal by crane, while the auxiliaries can be located on the underside, the lower weight allowing removal by a forklift.

The main advantage of the radial engine is its low specific weight and compactness which results in low specific cost ($\approx 25\%$ lower than V-engine) as discussed in Section 3.3. Its main drawback is the vertical axis orientation requirement which limits the application and affects the market acceptability. It is however possible--since all heat transfer and cylinder/seal hardware is the same as for the V-engines and the crankshaft and housing hardware is inherently simple (no camshafts) that for applications where the vertical axis has no adverse effects this type of engine can be used quite cost effectively. Thus for second generation hardware radial engines could be used within a V-production line in order to enhance cost competitiveness for certain applications.

2.4.3.2. Radial Modularity

The radial engine presents an ideal interface as shown in figure 2.4.3-2. Each four cylinder block and crankshaft can be identical with mating faces at each end. The units can be attached by means of tie bolts which are installed from one end. In this manner a significant amount of flexibility can be maintained since servicing can be much simplified.

An alternative approach would incorporate a block and crankshaft for each specific engine configuration in the power range, but for the radial engine this approach does not seem warranted owing to the simplicity of the modular assembly.

FIG 2.4.3-1 RADIAL ENGINE

ADVANTAGES

SIMPLE CRANKSHAFT

"STIFF" DRIVE

LOW SPECIFIC WEIGHT

SIMPLE BALANCING
(2 COUNTERWEIGHTS)

LOW SPECIFIC VOLUME

GOOD MODULARITY

DISADVANTAGES

ARTICULATED PISTON MOTION
NON-SINUSOIDAL

SLIGHT IMBALANCE DUE TO
NON-SINUSOIDAL ASPECT
OF ARTICULATED DRIVE
(PRIMARY ROD).

VERTICAL SHAFT
(MAINTENANCE)

RADIAL ENGINE ---- CYCLE FORCES

Figure 2.4.3-2

Figure 2.4.3 3

TWO OR MORE FOUR CYLINDER ENGINES STACKED WITH VERTICAL SHAFT AXIS
MODULAR APPROACH WITH EITHER BOLT JOINTED OR ONE PIECE CRANKSHAFT
VERTICAL SHAFT ORIENTATION PROVIDES BEST MAINTENANCE

2.4.4 SWASHPLATE, ENGINE CONFIGURATION

2.4.4.1 Engine Description

The Swashplate drive can be used in conjunction with the Rinia principle with any number of cylinders which will yield acceptable phase angle relationships. The reciprocating motion is transformed into rotary motion of a shaft parallel to and centered relative to a circle containing the cylinders by means of a disk permanently attached at an angle to the shaft. Each piston rod contacts the plate via a recessed crosshead which contains a bearing on either side of the disk. As the shaft and disk rotate, harmonic motion is transferred to the pistons.

The Swashplate engine described in figure 2.4.4-1 accomodates the same cylinder, seal and HX assemblies as previously described on V and radial engines. A four cylinder (90° phase angle) engine is shown, but other combinations will be discussed in section 2.4.4.3. The geometry lends itself well to a horizontal shaft orientation for easy interface to the heattransport system to provide a uniform pipe length without cumbersome bends. Cylinder/seal maintenance is reasonably simple, since the heat transport system can be translated horizontally for cylinder removal; the generator is also directly accessible. The crosshead and swashplate servicing is more difficult, since the crankcase must be separated to provide internal access.

2.4.4.2 Drive System Elements

The swashplate is driven in rotation by the axial forces of the piston rods (due to cycle gas and acceleration forces) acting on the non-perpendicular surface of the plate, resulting in a tangential acceleration of the shaft as well as a radial force vector. The direction and magnitude of this force change significantly throughout the rotation (cycle) of the engine, and the slider bearings which transmit the forces between disk and reciprocating mass are critical to the design. The bearings pads shown in the figure 2.4.4-1 are essentially hemispheres which are seated in almost matching recesses in the crosshead as the flat face mates with the disk. In order to create a wedge film effect required for hydrodynamic lubrication, two approaches can be taken:

The "flat" face of the bearing pad can be made convex by a few thousandths of an inch,

The surface of the swashplate disk can be machined to a slightly conical configuration.

The crosshead is notched to accomodate the swashplate, and as a

result an eccentric load path is created. When a tensile load is created in the rod by acceleration and cycle forces, it must be reacted by the lower bearing. This results in a moment being transmitted through the reduced section of the crossheads and causes a bending deflection to occur. Due to fairly critical bearing clearances between the crosshead and bore, these deflections must be limited to prevent wedging of the crosshead and resulting high friction losses. Methods of reducing this effect include, 1) reducing the thickness of the swashplate and consequently the crosshead cutout length, 2) use of a high modulus crosshead material to limit deflection 3) offsetting the swashplate bearings with respect to the piston centerline to increase the cross section in the cutout area and 4) increasing the crosshead diameter and offsetting it to further increase the bending cross section.

The swashplate thickness can be reduced until strength requirements become marginal, especially in this size engine where counterbalancing effects are relatively small.

Making the crosshead from a steel alloy in lieu of aluminum triples its stiffness, but also triples its weight. Figure 2.4.4-2 indicates that as the reciprocating mass increases, the peak forces on the slide bearings are decreased. The disadvantage is obviously an increase in required balance weight mass and overall cost.

The third method listed allows the bearings to be located inboard of the rod axis which results in a torsional load on the crosshead. The bearing surface on the face of the swashplate can be used to react this load from the recessed face of the crosshead, but the existence of an additional load and the resulting friction and complexity are somewhat detrimental.

By offsetting the crosshead itself relative to the cylinder axis, a significant gain in crosshead stiffness can be realized. The option to increase the diameter also exists with the limitation that this increases overall engine size and weight as well as friction losses.

The swashplate support bearings are a critical design element. Two basic choices exist, since either rolling element or sleeve bearings can be used for support. The use of tapered roller or spherical roller bearings for radial and thrust support requires a larger volume than sleeve bearings, but exhibit some advantages. Frequent start-up cycles are better handled by roller bearings, unless pressurized lubrication is available from an externally driven pump. Also, a steep angle tapered roller bearing can be used which will carry both the thrust and radial loads and thus simplifies the bearing system. Since a pressurized lubrication system is mandated for the crosshead and since heat removal from the

swashplate bearings will be required, the overall system cannot be significantly simplified through the use of roller bearings. Sleeve journal bearings can be used to greatly reduce the bearing housing size, but the thrust bearing will be more complex since a tilting pad segmented approach will probably be required. A swashplate engine with journal bearings is shown in figure 2.4.4-2.

The swashplate Rinia engine approach provides almost constant shaft bearing loads due to the symmetry of the cycles, the only variations result from the slight non-sinusoidal effects of the gas forces (drive is truly sinusoidal).

Figure 2.4.4-3 shows the bearing loads vs. swashplate rotation angle. The radial load vector travels through the full circle (360°) and leads the plate high point approximately 25° .

The bearing loads were calculated using the instantaneous cycle pressures based on a modified Schmidt analysis and are shown in fig. 2.4.4-4.

Balancing of the swashplate engine is quite simple because of the pure sinusoidal motion. All primary forces due to accelerations of the reciprocating masses are inherently balanced, since their motions are identical and their composite C. G. passes through the engine centerline. There is a resulting inertial couple, however, since the opposed motion masses are not coplanar. The centrifugal couple created by the rotating swashplate partially cancels this effect, since it is 180° out of phase with it, but additional counterweights are used to provide total balance (smaller engines can often be balanced by simply altering the plate thickness or mass).

2.4.4.3 Modular Approach

The swashplate engine lends itself well to several modular approaches. The four cylinder configuration shown in figure 2.4.4-5 provides a 500 horsepower unit with its own generator. Multiples of these units could be used to satisfy the full range up to 3,000 horsepower, with the obvious advantage of on-line redundancy, inherent incremental control and fabrication simplicity. The failure of one mover in a multi-engine installation would not result in system shutdown, but would allow continued operation at reduced power. The drawbacks are the need for increased switchgear, multiple auxiliaries and generators, more complex installation, and increased floor space requirements.

Another approach is the use of a gearbox to drive a single generator. This would reduce the complexity of the switchgear, but the gearbox cost has been estimated to be in the range of approximately \$30.00 per horsepower, a significant cost burden for the engine. A more favorable approach is to build a 1000 HP, eight cylinder swashplate engine/generator unit employing a 90° phase angle by interconnecting every other

cylinder to form a pair of Rania configurations. The 125 horsepower cylinder is maintained, and a 500 horsepower module is obtained by using only 4 cylinders, sealing the remaining interface ports and revising the balancing weights. In this manner, the redundancy feature in large systems is maintained, while the cost disadvantages of the large number of redundant generators, switchgear and auxiliaries are significantly relieved.

Larger numbers of cylinders can of course be implemented, but the increased fabrication related costs resulting from the increased number of different parts are significant. Therefore the most cost effective and recommended approach is a 1000 HP eight cylinder engine/generator unit; to satisfy the the full range of power requirements multiples of the 1000 HP unit are used while the 500 HP unit uses the same drive system but only 4 cylinders and a smaller generator.

FIG 2.4.4-1 SWASHPLATE ENGINE

ADVANTAGES

- COMPACT ENGINE PACKAGE
- SHORT CYLINDER INTERCONNECTION
- EXCELLENT BALANCING POTENTIAL
- SINUSOIDAL PISTON MOTION
- ROTARY BEARINGS ARE SUBJECTED TO A CONSTANT LOAD, AND CAN BE EITHER ROLLING ELEMENT OR SLEEVE JOURNAL CONFIGURATION
- HORIZONTAL SHAFT ORIENTATION ENHANCES GENERATOR AND HEAT TRANSPORT SYSTEM ACCESS

DISADVANTAGES

- HEAVY CROSSHEAD AND HIGH DYNAMIC LOADS
- CRANKCASE ACCESS REQUIRES SEPARATION
- ACCESSORY POWER TAKE-OFF DIFFICULT
- SLIDER BEARING DEVELOPMENT REQUIRED
- MARKET ACCEPTABILITY UNCERTAIN
- REMOTE WATER PUMP

FIG 2.4.4-2 SWASHPLATE ENGINE WITH JOURNAL BEARINGS

IFG. 2.4.4-3

SWASHPLATE SLIDER BEARING LOAD & PLATE SUPPORT TOTAL LOAD

LSSE ENGINE
 $P_m = 100$ ATMOS
 125 HP/CYL
 7.00" BORE, 4.00 STROKE
 1200 RPM

SWASHPLATE SUPPORT LOADS & CROSSHEAD SLIDER

BRG LOAD
 -POUNDS-

FIG 2.4.4-4 SWASHPLATE ENGINE PRESSURE VS. ϕ (TYPICAL CYLINDER)

LSSE ENGINE
PM = 100 ATMOS.
125 HP PER CYLINDER

- 5 -

FIG 2.4.4-5 SWASHPLATE ENGINE MODULARIZATION

FOUR CYLINDER ENGINE WITH
INDIVIDUAL GENERATOR

SEVERAL FOUR CYLINDER ENGINES
DRIVING ONE ALTERNATOR THROUGH
USE OF A GEARBOX

EIGHT CYLINDER ENGINE WITH
1,000 HP GENERATOR, F
CYLINDERS CAN BE USED
500 HP OUTPUT.

5 HEAT TRANSFER LOOP

During initial configuration selection the only viable approaches possible for the heat transfer loop were a Sodium heat pipe and a pressurized gas circulation system.

Both systems have been investigated in some detail, but within the scope of the present conceptual design effort a clear choice did not emerge. A heat pipe system appears simple and does not require auxiliary power; with approximately \$25 per horsepower it is a more cost effective choice. Operational and field maintenance/repair problems as well as unknown reliability and long life characteristics and a slight safety concern do not make it a clear choice, especially for the 1985 demo engine.

A pressurized gas heat transport loop has the obvious operational advantage of an easy thermal separation (by switching off the gas circulator) which enhances maintenance and repair operation and also gives the overall system a higher flexibility with respect to engine positioning/heat source interface. Also a switchover to another heat source (redundancy) is possible without undue complexity.

However, high pressure (approximately 40 atmospheres), high temperature (1650°F) piping is expensive and double wall construction is required to obtain reasonable wall thickness (discussed in section 2.5.2). Furthermore a suitable gas circulator is not available off the shelf and considerable development is required for the first unit even though all technologies are available. The cost in production rates of approximately 10/year is projected as \$25 per horsepower with the entire system cost exceeding \$60 per horsepower. Auxiliary power requirement is approximately 5% which adds over 10% on fuel consumption and effectively \$15 per horsepower for equivalent engine cost.

Both systems are discussed in more detail in the following sections:

2.5.1 HEAT PIPE SYSTEM

2.5.1.1 Overall System Description

Considering that the operation of the heat pipe requires no auxiliary power and large heat fluxes are obtained at low temperature differences, the heat pipe is an ideal match for a large Stationary Stirling engine, powered by an AFB-heat source.

Fig 2.5.1-1 shows a sketch of a heat pipe system installation for a 1000 HP V-8 engine. Each of the pipes transports approximately 250 KW_{th} (approximately 850,000 BTU/hr) and feeds one cylinder.

From a preliminary sizing of the AFB it appears that the engine length and the bed width are quite well matched and a straight heat pipe can be used for all cylinders of a V-engine.

FIG 2.5.1-1 HEAT PIPE INSTALLATION

In order to arrive at equal length heat pipe units for both cylinder rows the heaterhead HX are placed such that all heat pipes are equal size.

To satisfy the total heat input requirement to one heat pipe (approximately 250 KW Thermal) a second pipe is needed in the AFB and therefore a second branch is added to the evaporator section. This requires that both heat pipes are separated 18" vertically which is unacceptable for the heaterhead HX (dead volume). Therefore the lower heat pipe is somewhat tilted (approximately 11°). This tilt provides sufficient clearance for a removal of either heat pipe from the engine without disassemblies of other components.

In order to enhance heat transfer rate in the evaporator section, heat transfer fins .5" high and .1" thick are added with a spacing of 6 fins per inch.

All heat pipe assemblies are supported by a constant force support system; this allows horizontal movement due to differential Thermal expansion without significant stress at the heaterhead HX interface.

2.5.1-2 Heat Pipe Description

Fig 2.5.1-2 shows a sketch of the heat pipe assembly, table 2.5.1-3 summarizes system characteristics. The evaporator consists of two 5" OD pipes approximately 5 feet long which are joined together near the furnace wall.

A 3.5" OD adiabatic section approximately 8 feet long connects the evaporator section with the heaterhead HX which is the condenser section. This section is designed as a 6" ID, 12" long reflux boiler where the sodium vapor coming from the evaporator section condenses on the heaterhead HX tubes. The sodium condensate collects on the bottom of the HX where it is returned through capillary action to the evaporator section by approximately 20 arteries. These arteries are flexible wound screen tubes with approximately 1/4" OD.

In order to assure an even sodium charge in both evaporator branches the upper branch is connected to 12 arteries while the bottom pipe, favored by gravity, uses only 8 arteries.

The location and attachment of the arteries is shown in fig 2.5.1-2 (bottom) for the adiabatic and evaporator section. Materials used are summarized in table 2.10-1.

One of the major material selection concerns is hydrogen permeation due to atmospheric water vapor on the outside of the heat pipe and several coatings and materials (small traces of aluminum) are presently under investigation to resolve this problem. Another concern is the erosion of pipe and fin material in the AFB due to the abrasive action of the bed material. Very few data are presently

FIG. 2.5.1-2

HEAT PIPE ASSEMBLY

TABLE 2.5.1-3
HEAT PIPE CHARACTERISTICS

THERMAL CAPACITY	850,000 BTU/hr	250 KW _{th}
	SINGLE PIPE	PER CYLINDER
TRANSPORT MEDIUM	SODIUM	
	t _m = 97.5°C	(207°F)
	t _b = 880 °C	(1620°F)
SODIUM INVENTORY	6.6 lb. (3.1 ltr., .8 gal.)	
MATERIALS PIPE	PIPE INCONEL 800	
	SCREEN & ARTERIES	316 L
	FINS	800 H
EVAPORATOR SECTION	2 PIPES 5.5" OD	5' LONG
	6 FINS/INCH	.5" HIGH .1" THICK
ADIABATIC SECTION	SINGLE PIPE 4" OD	~8' LONG
	20 ARTERIES	.25 OD
CONDENSER	HEATER HEAD HX WITH GRAVITY RETURN ARTERIES SUBMERGED IN SODIUM SUMP ABOVE REGENERATOR WALL.	
MAX. NEGATIVE TILT	≈ 6.5° (18" AGAINST GRAVITY)	
WEIGHT	HEAT PIPE	210 lb.
	FINS	<u>206 lb.</u>
	TOTAL	416 lb.
COST	PIPE	\$ 2150
	FINS	<u>960-</u>
	TOTAL	\$ 3110

available on this subject but with increasing field experience of this type of combustion system reliable data should become available soon. Ceramic coating systems could provide an effective barrier for bed erosion. Carbon precipitation in the exposed welds inside the AFB has to be also investigated but is not considered a serious problem.

2.5.1.3 Operating Characteristics

One of the most advantageous operating characteristics of the heat pipe is the low temperature differential between evaporator and condenser section. A preliminary analysis showed a total temperature difference from evaporator outside tube wall to the heaterhead HX tube wall of 28°C at full power (evaporator $\Delta T_e = 16^{\circ}\text{C}$, adiabatic section $\Delta T_a = 5^{\circ}\text{C}$, condenser section $\Delta T_c = 5^{\circ}\text{C}$).

This low temperature differential can become a problem where maintenance or repair requires work on high temperature engine components since the entire heat source has to cool down before work can begin. This can probably be sped up by using the generator as a motor (if type of failure on the engine permits) to increase cool down rate of the AFB.

Another method to overcome this problem is the use of a gas controlled heat pipe where an inert gas (helium) regulates the location of the vapor front. Thus, it can prevent the sodium vapor from entering the condenser section and shut off the heat transfer. However, this increases the complexity of the heat pipe system and adds additional failure modes. The same method could be used to prevent the evaporator section from boiling dry during cool down since the condenser and adiabatic section are more and more blocked off by the inert gas (due to the decreasing sodium vapor pressure) thus avoiding excessive sodium deposit on the cooler sections (sodium freezing point 98°C , approximately 210°F).

A detailed design has to address this problem during the preliminary design phase, where transient temperature histories are investigated.

A further operational / maintenance consideration is the impossibility for field repair of a failed (leak) heat pipe. Vacuum pinch off, bake out treatment and processing requires high skill level and specialized processing and heat treat facilities. Therefore, heat source wall penetration design and engine interface has to be aimed at fast exchange of pipe units. Specialized shipping containers are required to overcome potential safety hazards during shipments.

2.5.2 PRESSURIZED GAS HEAT TRANSPORT SYSTEM

2.5.2.1 Overall System Description

Figure 2.5.2-1 shows a pressurized gas heat transport system for an eight cylinder (1000 horsepower) engine, having a capability of transporting $6.8 (10)^6$ BTU per hour of heat from the fluidized bed to the engine. The system consists of a bank of exchanger tubes which will be immersed in the fluidized bed and are connected to two header tubes which run from the AFB to the vicinity of the engine. Feeder tubes are used as the interface between the headers and the engine, one from each header to every cylinder. In order to provide gas flow, a circulator is located in the return header. The circulator is electrically driven, and can be shut-off to stop heat flow to the engine in the event of a failure.

2.5.2.2 Pressurized Gas Loop

An evaluation of the use of air to replace helium in the transport system resulted in the parameters shown in table 2.5.2-2. The pressure level for air is twice that required for helium, and the flow rate is higher resulting in a larger power consumption. However, the use of air greatly simplifies the design by permitting higher leak rates to be accepted, which with helium would increase operating costs. The tube sizes were held constant for both fluids, having been selected from data shown in figure 2.5.2-3 for helium. Six inch inside diameter headers are used between the engine feed tubes and the combustor. The circulating gas is at approximately 1625° F (hot side) and 1425° F (return), thus the header wall must be designed to resist a creep rupture failure. Figure 2.5.2-4 indicates the wall thickness required, and does not include any safety factor at all.

To preclude the use of these thick walls, the use of a light gauge inner shell (6" diameter) with vent holes to an external insulation chamber and then a pressure tight tube outside of the insulation, as shown in detail D of figure 2.5.2-1 is proposed. In this manner, low cost materials can be used for the relatively cool pressure vessel, and the wall thickness can be significantly reduced due to the higher material allowables at the lower temperature levels. The feeder tubes between the headers and cylinders each have a 2.5 inch inside diameter, and are made from 310 stainless steel. The insulation is external to these tubes, with the interface shown in detail D of figure 2.5.2-1 being typical for these tubes as well as for the AFB immersed tubes. Each tube interfaces with the engine cylinder via a bolted flange with a high temperature face seal.

The heat exchanger tubes immersed in the AFB span the headers and exchange the heat from the AFB to the pressurized gas.

TABLE 2.5.2-2

HEAT TRANSPORT LOOPPRESSURIZED GAS LOOPCHARACTERISTICS

	<u>He</u>	<u>AIR</u>	
PRESSURE LEVEL	450	900	psi
FLOW RATE PER CYLINDER	3800	20000 lb./hr.	
POWER CONSUMPTION * PER CYLINDER	4.9	5.9	HP
HEADER SIZE ID	6"	6"	
FEED PIPE SIZE ID	2.5"	2.5"	
GAS CIRCULATOR ΔP	1.3	4.7	psi
GAS CIRCULATOR SPEED RPM	11,600		
GAS CIRCULATOR COST **	25K\$		

* $\eta_p = .5$

** FOR A 8 CYLINDER 1000 HP UNIT

The tubes shown are 1.60 inches inside diameter with an average length of twelve feet, and 120 are needed for an eight cylinder engine.

The cost of the tubing, supports and required insulation is estimated to be approximately \$50.00 per kilowatt.

2.5.2.3 Gas Circulator

Several manufacturers were contacted in an unsuccessful attempt to identify a circulator which could be used for this application. However, two companies responded with proposals to develop a unit for the helium system. Both propose the use of gas bearings using the helium as the fluid, and also the use of magnetic couplings to eliminate the use of a rotating seal or immersed motor. The development cost is quite high (approximately \$1 million), and predicted unit costs of \$25,000 each for a quantity of ten result in an additional cost of \$33.00 per kilowatt for the circulator, raising the pressurized gas heat transport system total cost to over \$80.00 per kilowatt. Figure 2.5.2-5 defines the power consumed by a typical blower for an air or helium system. While not pursued as part of this activity, it is believed that an air system circulator may be lower in development and unit cost since the use of more conventional hardware may be feasible if slight leakage is acceptable. For example, the use of a rotating shaft seal may permit the rotor shaft to be externally supported, thus precluding the need for gas bearings.

FIG. 2.5.2-3

HEAT TRANSPORT LOOP PIPING LOSS
POWER REQUIREMENT

FIG. 2.5.2-4

LSSE PRESSURIZED GAS HEAT TRANSPORT LOOP TUBING

MAT'L - HOT SIDE = INCOLOY 800H
RETURN SIDE = 316 CRES STEEL

$T_n = 1625^{\circ}\text{F}$
 $T_c = 1440^{\circ}\text{F}$

M.S. = 0

FIG. 2.5.2-5

HEAT TRANSPORT LOOP PUMPING POWER

AIR FLOW 20,000 LB/HR CYL

HE FLOW 3,800 LB/HR CYL

$\eta_p = .5$

HEADER 6" ID

FEED PIPE 2.5" ID

2.5.3 INSULATION AND SUPPORT

The support of the heat transport loop is quite important due to the high temperature and consequently the sensitivity to creep and fatigue failures. The loop itself is quite massive and each support point represents a potential heat leak. The selection of points is made to maintain stresses at appropriate levels, not only from the static load of the loop, but from the dynamic inputs arising from engine operation as well. The use of constant force supports consisting of relatively low spring rate tensioning devices will provide support with the minimum amount of rigidity, thus providing a high degree of support flexibility and reducing the hazard of high stresses due to differential expansion. Supports will also be required within the AFB, since the exchanger array will be quite massive. These supports can have wires penetrating the combustor walls with the spring device located remotely in a cool area.

Both the heat pipe and pressurized gas heat transport loops will require insulation between the AFB insulated wall and the engine heat exchanger.

Since the heat pipe system experiences a very small differential pressure across the pipe wall no pressure support wall is required and a conventional sleeve insulation surrounding the pipe connecting each cylinder to the AFB will be sufficient as insulation system.

The pressurized gas loop with its two pipes per cylinder operates with a relatively high internal pressure and for the 8" header pipes the use of a double wall, similar to the engine cylinder approach, is the only economically feasible approach to permit the use of smaller wall thickness and lower cost (lower temperature) alloys for structural containment. The insulation for this approach is contained between the walls, and is subjected to the gas pressure. The insulation for the smaller diameter feeder pipes is external to the pipes and can be assembled in the conventional manner. This approach is shown in Fig 2.5.2-1 detail D.

Selection of insulation material will be based on commercial availability in appropriate geometric sizes and forms and cost effectiveness with respect to heat loss and material cost; for the insulation contained on the inside of the (pressure retention) pipe consideration has to be given to the possibility of loose fibers or particles which can be transported with the heat transfer medium (helium or air) and cause problems in heat exchangers, gas bearings and rotor of the gas circulator. Table 2.5.3-1 lists several candidate materials which will be selected during the preliminary design stages. For costing purposes Kaowool (Babcock and Wilcox) was selected in the present design.

TABLE 2.5.3-1 TYPICAL COMMERCIALY AVAILABLE FIBROUS INSULATIONS

Name	Manufacturer	Type	Maximum temperature	
			°F.	°C.
Fiberfax	Carborundum Co.	Aluminum silicate	2300	1260
Kaowool	Babcock and Wilcox	Aluminum silicate	2300	1260
Refrasil	H. I. Thompson	Silica fibers	1800-2000	980-1090
Zirconia fibers	H. I. Thompson	Zirconia fibers	3000	1650
Dyna Quartz	Johns-Manville	Silica fibers	2750	1510
Thermoflex	Johns-Manville	Aluminum silicate	2300	1260
Tipersul	E. I. du Pont	Potassium titaxate	2200	1200
Sapphire wool	Thermokinetics Fiber, Inc.	Aluminum fibers	3200	1760

2.6 HEAT SOURCE INTERFACE DEFINITION

2.6-1 Coal Fired Systems

An evaluation of several combustion systems was performed to determine the best match for a Stirling engine in the 500 to 3000 horsepower range. Foster Wheeler Development Corporation employed their background and knowledge of the field to address the problems of matching the Stirling engine with the various combustors, including emissions levels, combustor life problems due to corrosion, slagging problems, heat transfer rates, process efficiency, control and turndown capability, carbon loss, power consumption, approximate cost, fuel flexibility, coal preparation and handling maintenance and others.

Appendix D and E address these problems and consideration in detail; only a short summary will be given in table 2.6-1.

Most systems investigated suffer either from low efficiencies or too large capacities (thermal input for 500 to 3000 HP engines range from 3.4×10^6 to 20×10^6 BTU/HR).

The fluidized bed appears to be the best choice of all the coal combustion systems for mating with the Large Stationary Stirling Engine.

A fluidized bed provides the advantages of fuel flexibility, thus extending its capability beyond coal and providing the potential to burn combustibles available at specific sites. Since the exchangers are immersed in the expanded bed, the heat transfer rate to the heat transport system is considerably increased. Both the heat pipe and pressurized gas systems interface well with the fluidized bed. Although large quantities of fly ash are generated, sulfur removal is simplified by combustion capture. The bed response time is quite slow, but the potential for high turndown ratios exists.

A cost analysis of the fluidized bed system for this application is difficult to develop, since there is no precedent for a combustor in this size range. Also, the coal handling and preparation system capacity requirements are lower than on existing systems. Therefore, the components may need to be specially designed and consequently these costs are not available. Existing unit costs have been extrapolated in an attempt to define a system cost range as shown in figure 2.6.1-2, where the large effect of the unknowns are apparent.

TABLE 2.6.1-1

Combustion Systems Summary Chart

COAL

<u>System</u>	<u>Advantages</u>	<u>Disadvantages</u>	<u>Comments</u>
Stoker	<ul style="list-style-type: none"> No preparation of coal (such as pulverizing or drying) required. Particle size 1-1/4" x 0--no more than 40% <1/4". Simple design. 1.4×10^4 Btu/h--500×10^4 Btu/h size range. Furnace exit about 2300°F Eastern coals; 1950°F Western coals. 	<ul style="list-style-type: none"> Low heat-transfer rate.* Max. air preheat temperature 200 to 350°F. Process efficiency = 43%.† Air preheat 200° max. Excess air = 250%. Low to high carbon loss. High excess air. Potential corrosion of air preheater between 1000 and 1500°F. 	<ul style="list-style-type: none"> Suitable for a pressurized He heat-transfer loop. Development required for feed systems of the small sizes needed. Sulfur recovery back end required. Impractical for rapid load swings. $\Delta T_m = 336$ to 232°F.‡

* $h_g = 5$ to 12 Btu/h·ft²·°F (Lower value parallel flow); at $V_g = 100$ ft/s (Upper value cross flow); Parallel flow may have to be used at the lower capacities to accommodate the bundle.

†All processes limited to 350°F flue gas exit temperature. Calculated efficiencies are only approximate.

Process efficiency may be further restricted by upper limit of minimum air preheat and the use of excess air to cool flue gases to temperatures less than 2300°F for ash deposit control.

Process efficiency calculated at 1300°F air preheat; $\eta = 74\%$; if 1000°F limit imposed, $\eta = 64\%$; if 750°F limit imposed, $\eta = 62\%$.

Typical Losses at 350°F Exit Temperature:

	<u>Stoker, P.C., Cyclone</u>	<u>Fluidized Bed</u>
Dry Gas	5.49	5.49
Fuel H ₂	5.85	5.85
Air H ₂	0.15	0.15
Combustion Loss	0.30	0.30
Dust	0.00	0.33
Radiation	0.28	0.28
Manufacturer's Margin	1.5	1.5
TOTAL	13.57	13.9
$\eta =$	86.42	86.1

‡ $\Delta T_m = 336^\circ\text{F}$ (Furnace exit = 2300°F, based on 100°F approach temperature).

$\Delta T_m = 232^\circ\text{F}$ (Furnace exit = 1950°F, based on 100°F approach temperature).

TABLE 2.6.1-1 (CONTD.)
Combustion Systems Summary Chart (Cont)

COAL

<u>System</u>	<u>Advantages</u>	<u>Disadvantages</u>	<u>Comments</u>
Pulverized Coal	<ul style="list-style-type: none"> • Fast transient. • Good turndown. • Can use 1300°F air preheat with special burner design.[†] • Low carbon loss 0.2 to 2%. • Furnace exits about 2300°F Eastern coals; 1950°F Western coals. 	<ul style="list-style-type: none"> • Low heat-transfer rate.* • High cost of pulverizer system. • High power requirements. • Process efficiency = 74%.[‡] • Air preheat 1300° max. Excess air =250%. • Potential corrosion of air preheater at 1000 to 1500°F. 	<ul style="list-style-type: none"> • Sulfur removal by wet scrubber. • Present capacity range >100 x 10⁶ Btu/h. • Lower ranges require specially engineered mills. • Small sizes--turndown limited to capability of single burner. • 50 to 80% of all ash passes through boiler. • Coals limited to <0.4% Na₂O + K₂O dry coal basis and ash softening temperature >2200°F. • Best suited for He loop; sulfur recovery back end. • $\Delta T_m = 336 \text{ to } 232^\circ\text{F}.$[§]

See p. 1 ().

†Air preheaters are normally designed to $T_{\text{metal max}}$ 750°F using carbon steel. To 1000°F special materials must be selected with an incremental jump in cost. Above 1000°F the technology is not yet developed for proper material selection. The fluidized bed system has the best chance of capturing undesirable impurities that would subject the air preheater to corrosive abuse.

‡See p. 1 (†).

§See p. 1 (§).

TABLE 2.6.1-1. (CONTD.)
Combustion Systems Summary Chart (Cont)

COAL

<u>System</u>	<u>Advantages</u>	<u>Disadvantages</u>	<u>Comments</u>
Fluidized Bed	<ul style="list-style-type: none"> • Fuel flexibility <ul style="list-style-type: none"> - Coal - Wood - Oil - Gas - Combustibles $H > 3000 \text{ Btu/lb.}$ • High heat-transfer rate $h_g = 100 \text{ Btu/h-ft}^2\text{-}^\circ\text{F.}$ • Dry ash handling system. • No corrosion of heat-transfer surface in flue gas in the heat-recovery zone above the bed. • Excess air - 10 to 20%. • Sulfur recovered during combustion, end product--dry ash. 	<ul style="list-style-type: none"> • Low temperature $< 1750^\circ\text{F.}$ • Slow transient. • Air preheat temperature over 1000°F difficult. • Disposal of large quantities of fly ash. • Ca/S ratio about 6:1 at low gas velocities to maintain high heat-transfer rates. • Process efficiency $= 74\%.*$ • Air preheat 1300°F max. 	<ul style="list-style-type: none"> • Good turndown ratio requires specific attention. • Ideally suited for use with heat pipe transport loop. • Can also be used with He loop. • $\Delta T_{in} = 250^\circ\text{F.}$

*See p. 1 (+).

TABLE 2.6.1-1 (CONTD.)

Combustion Systems Summary Chart (Cont)

OIL

<u>System</u>	<u>Advantages</u>	<u>Disadvantages</u>	<u>Comments</u>
Residual Oil Burner	<ul style="list-style-type: none"> • Low cost. • Simple control and fuel handling. • Can use 1300°F air with special burner. 	<ul style="list-style-type: none"> • Low heat-transfer rate. • Vanadium treatment required. Must use Mg additive in a ratio of 3 Mg/1V for V > 0.5 ppm. Na must be washed to levels not exceeding 1 ppm. • Air preheater subjected to same corrosive abuse as turbine blading in the temperature range of 1050 to 1500°F. • Process efficiency = 74%. • Air preheat 1300°F Excess air 250%. 	<ul style="list-style-type: none"> • Gas turbine type of combustion system. • Sulfur may be controlled with an additive plus back-end cleanup. • $\Delta T_m = 336^\circ\text{F}$.

TABLE 2.6.1-1 (CONTD.)
Combustion Systems Summary Chart (Cont)

COAL

<u>System</u>	<u>Advantages</u>	<u>Disadvantages</u>	<u>Comments</u>
Cyclone	<ul style="list-style-type: none"> • No preparation of coal (such as pulverizing or drying) required. • High flexibility for low-grade coal. • Low carbon loss--<0.1%. 	<ul style="list-style-type: none"> • High maintenance cost. • Low heat-transfer rate.* • Process efficiency = 74%.† • Air preheat 1300°F Excess air = 250%.‡ • Potential corrosion of air preheater surface 1000 to 1500°F. 	<ul style="list-style-type: none"> • Liquid slag discharges; requires ash fusion temperature <2400°F. • Present capacity range >100 x 10⁶ Btu/h. • Best suited for He loop. • Sulfur recovery back end required. • $\Delta T_m = 336$ to 232°F.†

See p. 1 ().

†See p. 1 (†).

‡Only 10 to 15% excess air is required for combustion. Approximately 250% is added downstream of combustor and upstream of heat-transfer surface for flue gas temperature control to avoid deposit formation.

§See p. 1 (§).

FIG. 2.6.1-2

Approximate Distribution of Cost of Combustion System/kW as a Function of Capacity
(Based on studies of similar systems)

2.6.2. Oil Fired System

The residual fuel oil burner in which suspension firing is used is presently available for use in systems of the desired size. The technology is developed and the system cost will be low, with control and fuel handling considerations much simpler than with coal systems, but emissions control will require additives and exhaust cleanups. Fuels with concentrations of sodium, lead, calcium and especially vanadium above specified levels would require washing (or additives in the case of vanadium).

The most likely approach would be utilizing a fluidized bed, with a heat transport system in order to protect the heaterhead HX from the solids and other contaminant and also ease the problem of sulphur removal.

2.6.3 Gas Fired Systems

The use of a gaseous fuel provides a cleaner burning system than with either coal or oil, and direct fired, separate burners would be used for each cylinder. Figure 2.6.3-1 defines a concept in which a sheet metal housing is used with a corrugated outer ring serves as a parallel flow heat exchanger to preheat the inlet air. The gaseous fuel is mixed with the preheated inlet air via the nozzle at the top of the cylinder, and it then flows through either the ribbon type burner shown or a porous burner. Ignition takes place on the outside of the burner, where a laminar flame is supported a short distance from the face. A portion of the liberated energy is conducted by the gas/air mixture to the burner, from which it is radiated to the engine tubes. The remainder of the energy is carried by forced convection to the heater tube bundle. After passing through the bundle, the still hot combustion products are circulated through the air preheater and then to the exhaust stack and economizer.

A design of this type is presently undergoing test as a part of the GE Gas Fired Heat Pump Program.

2.7 AUXILIARIES

2.7.1 Cooling Loop

This system provides active cooling for the engine by means of a circulating, pressurized water loop. The system is responsible for removing in excess of 55% of the heat supplied to the engine, and therefore must circulate over 88 gallons of water per minute per four cylinder engine block to maintain a 50°C temperature rise in the water. The engine helium cooler is the source of over 90% of this heat, with the oil and cylinder coolers contributing the remainder. Water circulation may be provided by a single pump, which is driven directly by the engine on all but the swashplate configuration, which utilizes a remote electric motor driven water pump since an additional power take off presents significant difficulties. This latter concept, while being less efficient, offers the advantages of speed control and circulation without engine operation. During engine operation at reduced power output, required engine speed is still 1200 rpm to provide 60 hertz power; but heat input and rejection are reduced, therefore water pump flow rate can be reduced. The electrically driven pump permits the flow to be controlled, but directly driven engine mounted pumps do not provide this flexibility. If a centrifugal pump is used (typical selection), the flow can be controlled by means of an increase in head (baffle or orifice) and the pump power consumption is reduced and outlet temperature can be maintained at desirable levels. Since the pump efficiency is only slightly lower, the net result is a reduction in pumping power. Another approach would be to implement a primary and a secondary pump with a clutch drive, having the ability to switch off the secondary pump below a specific power level. In order to preserve engine cylinder output symmetry the coolant inlet to each cylinder should be at the same temperature, thus precluding a series connection of cylinder coolant lines.

To maintain lubricating oil properties at a reasonable level, the coolant should reach the oil cooler prior to receiving heat from the other systems. The cylinder heads are not as sensitive to final temperature variations, and thus can be on a series path with oil cooler. As shown in figure 2.2-1, an economizer can also be placed in the the cooling loop to recover some of the stack losses. Again assuming a 50°C water temperature rise an additional 85 gallons of coolant flow per 500 horsepower engine module to provide the economizer recovery shown in figure 2.2-2. The total pump output then becomes 173 gpm per module.

The heat from the cooling system is transported to a process heat exchanger to supply process and facility heat. In addition, an auxiliary cooling dump exchanger is included to provide a means of heat dumping in the event of a sudden reduction in process heat demand. The auxiliary loop would be valve actuated, and would only be used if required due to overtemperature condition.

2.7.2. Lubrication System

The lubricating oil is circulated throughout the engine to provide cooling as well as lubrication of the moving parts. As is typical for large reciprocating engines, oil passages are built into the crankcase, crankshaft, connecting rods and crosshead (or Piston) to provide adequate oil flow and pressure to maintain lubrication and cooling. The Stirling engine lubrication system is somewhat simpler than a four cycle IC engine system, since camshafts, valves and timing gears are non-existent. The flow system defined for V and radial engines consists of a primary axial flow artery, which branches off at each main bearing web to supply the crankshaft. The crankshaft oil channel will be continuous, such that supply ports will be redundant and oil will be fed to each connecting rod by means of radial grooves. Passages in the connecting rods will feed the wrist pin bearings as well as crosshead chambers.

The swashplate engine lubrication system differs only in the distribution portion. The swashplate rotary and thrust bearing are independently supplied with oil, and each crosshead is surrounded with an oil supply groove which feeds crosshead ports which supply lubrication to both surfaces of the slider bearings.

In all engine types the oil pressure is generated by a rotary pump which is driven directly by the engine. Oil is supplied from a crankcase sump, filtered and cooled before reaching the pump. The cooler exchanges the heat from the oil with circulating coolant supplied by the system's primary cooling loop. Also fed into the pump supply line is the oil returned from the seal scavenging system's oil separation sump. (Section 2.4.2.3)

2.7.3 Helium Supply and Recovery System

The engine contains over one hundred standard cubic feet of helium per 500 horsepower module, at a mean pressure of 100 atmospheres. The helium charging system will provide the gas needed for initial pressurization as well as make-up and control requirements.

Helium is commercially available in 524 SCF containers (10" diameter x 56" high) at 6000 psig, having a purity of 99.995% and a dew point of -76°F. The storage can be maintained in a bank of these cylinders with a regulated outlet connection to the charging system. For economic reasons the use of an in line compressor to provide better helium utilization may be warranted. A low pressure bottle should be maintained for helium recovery.

The conceptual design of the cylinder seal described in section 2.4.2. of this report described a seal which returned a mixture of helium and oil to a separating system after each had leaked from their respective chambers into the seal cartridge. The separator consists of a commercially available mist eliminator cartridge through which the mixture is circulated. The glass fiberfilled element within the eliminator captures droplets down to the micron ranges, and an activated carbon bed located down stream will serve to remove remaining oil vapor. The oil removed is collected by gravity in an oil sump having a level

indicator; when the upper limit is reached, engine oil is returned to the sump at a rate controlled by an orifice. Since the seal oil and helium pressure is near the engine mean pressure of 100 atmospheres the separator must be enclosed in a pressure vessel. For a 1,000 horsepower engine, the vessel size would be approximately twelve inches in diameter and four feet long. Helium which passes through the separator is fed back into the compression space of the cylinders by means of check valves which serve to maintain a uniform pressure in all cylinders.

2.8 CONTROL SYSTEM

Several methods of controlling Stirling engines are available, each having merits for specific applications. The use of a synchronous generator precludes the use of all speed related control methods except for start up and narrows the controls to the parameters of stroke, temperature, pressure volume and phase angle.

The phase angle control technique does not lend itself to this application since the use of a Rinia engine dictates certain phase relationships between cylinders.

Temperature control can be implemented on either hot side, cold side or both, but it significantly affects system efficiency if the control band is wide. Also, the response time of such a control system is slow, due to relatively large masses of metal in the engine heat exchangers.

Stroke control has been effectively used as control in wobble plate engines, but it is not easily implemented in engines with fixed kinematic linkages, such as the V, radial, swash-plate or rhombic drive. While it does potentially provide the control characteristics required, the mechanical complexity required for implementation rules it out as a viable solution for large stationary engines.

The use of variable change pressure as a control approach provides a very broad turn-down ratio and very rapid response, but it does require a helium recovery system to conserve the working fluid vented from the system. This would consist of storage tanks and a compressor. The implementation of the system could be accomplished as shown in figure 2.8-1, which indicates the use of two valves per cylinder plus the feed valve and a bypass valve used to quickly disable the engine. With this system, the efficiency penalty is small over a wide control range, but the complexity of the numerous valves and compressors must be considered. Temperature control on the hot side is maintained by heat source control with a "feed forward" system. This approach regulates fuel input according to instantaneous engine power setting (before a temperature change on the hot side occurs) and only trims the fuel input for long term temperature adjustment. For power control see section 2.9.3.

The last method to be discussed is the volume control approach. This can be accomplished by either changing the effective coupling to a supplemental volume as shown in figure 2.8-2, or by a piston chamber shown in figure 2.8-3 and described in the following paragraph.

PRESSURE CONTROL

FIG 2.8-1

POWER

PRESSURE

VOLUME CONTROL

CHANGE EFFECTIVE COUPLING TO VOID VOLUME

FIG 2.8-2

P/P_{MAX}

PISTON CHAMBER

FIG 2.8-3

LSSE VOLUME CONTROL MECHANISM

To provide Stirling engine control capability while maintaining a relatively high engine efficiency, a system which provides a variable dead volume and thus a variable pressure ratio can be used. The system is greatly simplified when used in conjunction with a multi-cylinder engine by utilizing a means of controlling the volume variation using cycle gas pressure as the control force. Figure 2.8.3 shows the technique schematically, which contains a cylindrical pressure vessel that can be made as an integral part of the crankcase. The vessel is divided into one chamber per cylinder by means of fixed baffles, and a traveling piston assembly consisting of the same number of sliding baffles fixed to a common rod is used to provide allocation of the volume tied to the engine cold side.

Moving the common piston rod will increase or decrease the total dead volume and thus change the engine power output. Since the pistons are large in diameter (approximately 10 inches) the force to move the piston would be significant, and variable according to the instantaneous cycle pressure. Therefore, four pistons of a Rinja module are coupled together and through a small leak (across the piston seals) average cycle pressure is maintained on the down stream side of each piston. Since the pressure wave in each of the four cylinders is 90 degrees out of phase with the adjacent one the piston forces are balanced between each other and the resulting force is the average cycle pressure. Since this is balanced by the average pressure on the down stream side of each cylinder the net force on the connecting control piston rod is theoretically zero.

To move the piston rod either a small hydraulic cylinder could be used or the down stream pressure is unbalanced by either increasing or decreasing the pressure with maximum or minimum cycle (cylinders) pressure maintained by small reservoir connected by check valves to the cold space of a cylinder, as shown in figure 2.8.3.

This control scheme requires more detailed analysis and design effort to study transients and mass balance effects which is not within the scope of this contract.

LSSE

2.9 ELECTRICAL SYSTEM

2.9.1 Generator/Starter

The preferred generator for LSSE application is a synchronous type with shaft driven brushless excitation. This type unit is produced by several manufacturers, avoids slipping, offers the advantage of reactive power control when compared with a less expensive induction type unit and generally has the most efficient type construction.

A 6 pole 1200 RPM unit is selected for direct engine connection in single bearing construction in order to maximize inexpensive factory assembly and test. The necessary starting duty up to the 1/3 speed at which the LSSE prime mover is self sustaining can readily be provided via operation of the generator as an unexcited synchronous motor. A torque of 40% of machine rating is typically available, which is more than sufficient for start up.

2.9.2 Switching and Control

Readily available motor starter contactors will be used for connection and load break duty on the generator, with fusing or circuit breaker short circuit protection when the system available fault current exceeds the unfused rating of the contactor. Generator ratings below 500 horsepower will be at 480 volts 3 phase, 60 hertz while those above 500 horsepower will have 2300 or 4160 volt windings in order to reduce current and avoid large bus bar or cable. General Electric Company Limitamp[®] motor control equipment is an example of the preferred equipment.

A source of electric power is assumed to be available in order to provide starting energy, power auxiliaries and give a speed reference (grid frequency) in order that a power control algorithm can be implemented. The relatively flat LSSE engine torque characteristics should provide minimal pulsating excitation and avoid the need for a smoothing flywheel, except for the torque damper which is provided in the engine assembly.

Starting of the engine through the generator is no different than starting a typical synchronous motor and connected load. The field is not excited and is provided with a discharge resistive path, then the starter winding is switched across the grid. High impedance grid connections will need reactor starters where voltage dip would be excessive, with the penalty of loss of some torque ability.

When the engine reaches self sustaining speed, it can contribute to the generator acceleration until synchronous speed is almost reached. Full synchronous speed is reached by increasing engine power and energizing the generator field at the appropriate angle through standard equipment designed for that purpose. Multiple engine generator installations can share a central control equipment rack and can be started in sequence. (Figure 2.9.1)

FIG 2.9-1

TYPICAL 3000 KW LSSE ELECTRICAL SYSTEM

Engine torque control is based on the assumed available electrical grid providing a speed reference due to its constant frequency control plus a source/sink capability for power. Engine torque is modulated by controlling effective displacer chamber volume in response to an error signal derived from a power reference and actual power output.

Reactive power is controlled using standard equipment that interfaces with the field voltage regulator capable of operating in other constant VAR or constant power factor mode for system flexibility.

2.10 MATERIALS

The Stirling engines have characteristically employed superalloys in the high temperature heat exchanger in order to permit the use of relatively thin walls to enhance heat transfer. Engines designed previously have generally been aimed at maximum efficiency, and have generally been designed at a temperature and pressure combination which ruled out the use of all but the more exotic alloys. Furthermore, low-power levels require more exotic design approaches in order to achieve high efficiency levels. For reasons of economics as well as availability, the LSSE design has been aimed at utilization of low alloy materials wherever possible. Sacrifice in efficiency and specific power have been made where justified to eliminate the need for the more exotic materials, and the more common austenitic stainless steels have been substituted. The information contained in Appendix F of this report defines the properties and characteristics of many of the available high temperature alloys as they relate to the LSSE utilization, and selection criteria are supported for the specific ranges of importance. Table 2.10.1 is a summary for the material selection for critical parts of the state of the art conceptual engine. Table 2.10.2 and 2.10.3 a property and cost comparison of candidate materials (from Appendix F).

TABLE 2.10-1
MATERIAL SELECTION FOR CRITICAL COMPONENTS

COMPONENT	TEMPERATURE	PRESSURE ATM	MATERIAL SELECTED
CYLINDER--OUTER WALL	≈ 250°	100+	316L
CYLINDER--HOT INNER LINER	1500° MAX.	SMALL DIFFERENTIAL	316L
CYLINDER--RUNNING LINER	300° MAX.	SMALL DIFFERENTIAL	4340 OR NITRALLOY
HEATER HEAD HX TUBES	1500° MAX.	100 +	INCOLOY 802
HOTSIDE CONNECTING TUBES	1500°	100 +	INCOLOY 802
REGENERATOR HOUSING	1400° MAX.	100 +	INCOLOY 802
REGENERATOR MATRIX	1400° MAX.	SMALL DIFFERENTIAL	ANY 300 SERIES S.S.
COOLER TUBES	400° MAX.	100 +	316L STAINLESS
COOLER HOUSING	250°	5	316L STAINLESS
HEAT PIPE TUBING	1750° MAX.	< 1	INCOLOY 800
HEAT PIPE UMBILICAL & WICK MATERIAL	1500° MAX.	SMALL DIFFERENTIAL	300 SERIES STAINLESS
PRESSURIZED GAS HEAT TRANSPORT LOOP HEADER PIPES (INNER WALL)	1650°	SMALL DIFFERENTIAL	316L
HEADER PIPES (OUTER WALL)	400°	45	316L
FEEDER PIPES	1650°	45	310 STAINLESS
HEAT EXCHANGER (AFB)	1750°	45	310 STAINLESS
HEAT TRANSFER FINS	1750°	0	316 STAINLESS
AIR PREHEATER (HIGH TEMP. SEC.)	1450°	SMALL DIFFERENTIAL	316L STAINLESS
AIR PREHEATER (LOW TEMP. SEC.)	1200°	SMALL DIFFERENTIAL	316L STAINLESS
GAS FIRED HEATER HEAD HX TUBES	1750°	100 +	INCOLOY 802
GAS FIRED HEATER HEAD FINS	1900° MAX.	0	INCOLOY 800

TABLE 2.10-2
COMPARISON OF CANDIDATE MATERIALS (1)

ATTRIBUTE	STAINLESS STEEL		INCOLOYS				INCONELS			HAYNES
	316(L)	310(S)	800	800H	802	825	601	625	617	183
BASE MAT'L COST (X=\$1.50/lb.)	X	1.2X	2X	2X	2.3X	2.5X	3.1X	5.6X	6.3X	8X
MILL DELIVERY (MONTHS)	3	4	6	6	8	10	10	17	10	10
AVAILABLE FORMS	ALL	ALL	ALL	ALL	FEW	FEW	ALL	ALL	FEW	FEW
THERMAL STABILITY (2)	2	4	4	5	5	3	5	4	1	2
RESISTANCE TO COAL GAS (2)	3	2	2	3	4	3	7	6	5	1
RESISTANCE TO H ₂ Permeation (2)	3	5	3	5	5	4	3	4	2	1
RESISTANCE TO Na (2)	2	3	2	3	4	3	3	3	2	1
RESISTANCE TO 0.1 VOL.% H ₂ S (2)	3	2	3	4	4	3	5	7	6	1
RESISTANCE TO 1.0 VOL.% H ₂ S (2)	5	2	2	3	4	3	6	8	7	1
RESISTANCE TO OXIDATION (2)	6	3	4	4	5	5	1	2	2	1
EASE OF WELDING (2)	1	6	6	6	5	4	3	2	4	4
EASE OF BRAZING (2)	1	2	5	5	3	5	6	2	5	4
10,000 HR. @ 1500°F RUPTURE STR. (3) $\times 10^{-3}$	3.05	2.8	4.0	5.2	8.2	-	4.0	9.5	12.0	10.0
RESISTANCE TO RUPTURE (1500°F) (3)	6	7	5	4	3	5	5	2	1	2
FATIGUE STR. (1400°F) (4) $\times 10^{-3}$	-	-	22	-	37	-	-	55	-	-
RESISTANCE TO FATIGUE (2)	6	7	5	4	3	5	5	1	2	2
THERMAL CONDUCTIVITY (5)	113	98	80	80	82	76	73	68	94	100
THERMAL EXPANSION (6)	8.9	8.0	7.9	7.9	7.8	7.8	7.6	7.1	6.4	6.6
RECOMMENDATIONS FOR LSSE SERVICE:										
COAL GAS/FLUIDIZED BED		1	3			2				
HOT SIDE CYL/SINGLE WALL AND/OR HEAT EXCHANGER TUBING					2			1	3	4
HOT SIDE CYL/DUAL WALL	1	2			3					
FLNS (AIR @ ~1750°F)			1				2	3		

(1) Comparisons made at 1500°F, unless otherwise specified.

(2) Rating: (1) Best to (9) Worst.

(3) Rupture Strength $\times 10^3$ PSI.

(4) Rotating Beam, $A = \infty$, $R = -1$, $\times 10^3$ PSI.

(5) Thermal Conductivity = BTU/FT²/IN./HR./°F

(6) Thermal Expansion = IN./IN./°F $\times 10^{-6}$

TABLE 2.10.3

NOMINAL CHEMISTRY (WT %), COST AND DELIVERY OF CANDIDATE ALLOYS

ALLOY	Fe	Ni	Cr	C	Co	Mo	Cb	Al	Ti	W	La	BASE MATERIAL COST \$/LB.	DELIVERY MONTHS
316(L) S.S.	Bal.	12	17	.03	--	2.5	--	--	--	-	--	1.50	1 to 3
310(S) S.S.	Bal.	20	25	.08	--	-	--	--	--	-	--	1.75	2 to 4
(1) INCOLOY 800 800H	Bal.	33	21	.03 1.0	--	-	--	.3	.3	-	--	2.90	4 to 8
INCOLOY 802	Bal.	33	22	.40	--	-	--	--	--	-	--	3.44	6 to 10
INCOLOY 825 (2)	30	Bal.	21	.05	.10	3.0	--	0.1	0.9	-	--	3.75	8 to 12
INCONEL 601	16	Bal.	23	.07	--	-	--	1.3	1.3	-	--	4.77	8 to 12
INCONEL 625	2.5	Bal.	22	.05	--	9.0	3.6	.02	.02	-	--	8.66	16 to 18
INCONEL 617	-	Bal.	22	.07	12.5	9.0	--	--	1.0	-	--	9.52	8 to 12
HAYNES 188	3.0	22	22	.10	Bal.	-	--	--	--	14	.10	~12.00	8 to 12

NOTES: (1) Incoloy 800 and 804 have same chemistry except "C" is .07 for Incoloy 800H.

(2) 2.25% Cu included in Incoloy 825 chemistry.

2.11 Speed Selection

The selection of a 1200 rpm operating speed for the engine was made early in the conceptual design phase, since it plays a significant role in engine sizing. Speed selection must consider the desire to use a direct drive to the generator, and consequently the synchronous speed of 3600 rpm (60 Hertz) must be an even multiple of engine speed to permit use of available generators. The primary effects of speed variations are drive system inertial load changes and the resulting friction loss and vibration effects due to imbalances. The relatively massive reciprocating weight of the piston and Crosshead assembly exacerbates these effects, and requires a more conservative approach than for a conventional internal combustion engine speed selection. This is partially, offset by the less violent cycle force pattern since the cycle pressure is almost sinusoidal for the Stirling and not spiked as in IC engine cycle pressure time history. Figure 2.11-1 shows a plot of many gasoline and diesel engine rated speed versus power, and indicates that for the 125 horsepower per cylinder level the 1200 rpm speed point selected is consistent, but somewhat conservative with existing engine design. Other speeds compatible with a directly driven generator would be 900 and 1800 rpm; the lower value is certainly too conservative, while the 1800 rpm speed would be near the edge of the existing design range, for the proposed state of the art engine design, 1200 rpm appears to be a good choice, but for the advanced design, the use of 1800 rpm should be considered to permit further engine cost reduction.

FIGURE 2.11-1
ENGINE RATED SPEED VS. POWER PER CYLINDER

3. SYSTEM EVALUATION AND COMPARISON

3.1 Evaluation Criteria and Rating Methodology

After conceptual layouts of candidate systems had been developed, an evaluation of these concepts was made to identify those which best satisfy the program requirements and goals. The results of this evaluation are shown in section 3.5. The principles and techniques which were employed to evaluate the concepts are discussed in detail in PIR SSE-79-002 which is attached as Appendix B. This section will only present a short summary.

The evaluation criteria which have been identified for the LSSE are presented in Table 3.1-1; they are divided into three categories - performance, costs, and qualitative assessment criteria. The contributing factors which are considered for the qualitative evaluation criteria are listed in Table 3.1-2.

3.1.1 Concept Evaluation Methodology

1. Since engine performance directly enters into the operating costs and thus is reflected in the levelized annual energy cost (see 3.1.2) performance is not used as a separate rating parameter. For all practical purposes the efficiency of all candidate engines is almost equal since all cylinders hardware and heat transfer equipment are identical for all engines; dead volumes are the same and the mechanical efficiencies of the drive train for all crank engines are sufficiently close. The mechanical efficiency of the swashplate may be slightly lower but since there are very few reliable data (none for large engines) and a rigorous drive train efficiency analysis would have been beyond the scope of this phase the efficiency of the swashplate was also assumed equal. Efficiency, therefore is not a meaningful selection criteria for the configurations chosen.
2. All systems are ranked separately: a) on engine cost or levelized annual energy cost and b) on the qualitative assessment criteria as listed in table 3.1-2.
3. For the qualitative assessment criteria, the assigned weighting factors are primarily subjective. Several qualified individuals were approached to establish weighting factors based on their assessment.

The result is shown in table 3.1-3. Based on these results, the weighting factors selected are shown in the last column of 3.1-3.

TABLE 3.1-1

PERFORMANCE	COSTS	QUALITATIVE ASSESSMENT CRITERIA
Peak and Part Load Efficiency	Levelized Annual Energy Cost* Development Production Maintenance Operation Installation	Reliability Maintainability Application/Interface Flexibility Fuel Flexibility Noise/Vibration Market Acceptability *Viability for 1985 Demonstration Adaptability for Advanced Development

* Only for Final Candidate Systems

TABLE 3.1-2

CRITERIA	CONTRIBUTING FACTORS
1) Reliability	No. of moving parts No. of wear related parts No. of brazed/welded joints - total base area Control complexity Maturity of hardware Factor of safety in design Uniformity of mass flow and heat transfer Experience on selected materials Number size and temperature of seals
2) Maintainability	Component accessibility Component exchangeability (no. of operations, down time) Special tooling required Consumables Estimated routine maintenance intervals No. of spare parts required Exchange of seals
3) Application/Interface Flexibility	Range of power/heat ratio Adaptability to load requirements and load profile Adaptability to heat source

TABLE 3.1-2 (Con't)

CRITERIA	CONTRIBUTING FACTORS
4) Fuel Flexibility	<ul style="list-style-type: none"> Capability to burn alternate fuels Degree of difficulty to implement change-over
5) Noise/Vibfation	<ul style="list-style-type: none"> Comparative noise estimate based on RPM Vibration levels based on achievable mass balance
6) Market Acceptability	<ul style="list-style-type: none"> Ease of installation Modularity Compliance with existing codes Required training/experience level of maintenance personnel Power quality
7) Viability for 1985 Demonstration	<ul style="list-style-type: none"> Technology risk Schedule risk
8) Adaptability to Advanced Development	<ul style="list-style-type: none"> Ease of integrating advanced technologies in an existing design

Table 3.1-3 WEIGHTING FACTORS FOR QUALITATIVE ASSESSMENT CRITERIA

	Individuals							Average	Selected Weight Factor
	1	2	3	4	5	6	7		
Reliability	.25	.31	.3	.31	.22	.25	.19	.27	.3
Maintainability	.2	.24	.2	.18	.13	.20	.22	.2	.2
Application/Interface Flexibility	.1	.12	.05	.10	.04	.15	.15	.11	.1
Fuel Flexibility	.15	.06	.1	.13	.13	.1	.1	.11	.1
Noise/Vibration	.05	.06	.05	.05	.13	.15	.12	.09	.05
Market Acceptability	.2	.15	.25	.13	.22	.05	.12	.16	.15
Viability for 85 Demo	.05	.06	.05	.1	---	.1	.1	.06	.05
Adaptability for Av. Development					.13				.05

4. Rating in these qualitative categories are based on the following method:

<u>Attribute</u>	<u>Rating</u>
Meets Ideal Standard (all goals)	8 to 10
Acceptable (meets requirement but not goals)	4 to 8
Unacceptable (does not meet requirements)	1 to 4

These subjective ratings are multiplied by the weighting factors and summed for all categories in one system. Highest numbers = highest ranking. The final results for the candidate systems investigated are listed in table 3.5-2.

3.1.2 Levelized Annual Cost

The Levelized Annual Cost (sometimes termed annualized cost) is the levelized dollar amount required during each year of the life of the system to:

1. Pay operating and maintenance expenses
2. Pay taxes, including corporation (where applicable)
3. Pay prorated interest on debt
4. Pay prorated return to stockholders
5. Build a capital fund that will allow for periodic component replacement or overhaul and retirement of debt.

Computation of Levelized Annual Costs (LAC) begins with calculation of year-by-year system costs. The present value (at the start year) of all system costs is then computed using after-tax cost of capital (r) as the appropriate discount rate. The present value is analogous to that amount which, if deposited in an account earning interest, (r), would permit withdrawals to pay all yearly costs and diminish to zero at the end of system life. There is a related amount which, if deposited annually in the same account would permit the same withdrawals. This is the Levelized Annual Cost. It is derived from the present value by using a capital recover factor (CRF) of the form:

$$CRF = \frac{r(1+r)^N}{(1+r)^N - 1}$$

where r = after-tax discount rate

N = system life

and $LAC = CRF \times PRESENT\ VALUE$

The Levelized Annual Cost is analogous to the annual mortgage payment (at interest r) on a loan equal to the present value of future costs.

To calculate the levelized annual energy cost for the candidate system, a computer program is used which was developed at GE Energy Technology Operation and modified for the recently completed Cogeneration Technology Alternative Study (CTAS) Program.

The economic ground rules and other required inputs used in this program are summarized in table 6 of PIR SSE 69-002, appendix B.

3.1. Output of Economic Calculation

A typical output is shown in table 9 to 12 of appendix B containing the following information:

Table 9 (report 5.1)	Fuel energy saved
Table 10 (report 5.3)	Capital costs
Table 11 (report 5.4)	Economic sensitivity report
vary Capital Cost of Cogen ECS	-20% to +60% in 20%
vary Cogen Fuel Cost	-20% to +60% in 20%
vary Electric Power Cost	-20% to +60% in 20%
Table 12 (report 6.1)	Fuel, emissions, and utility savings

In order to obtain a comparison with other cogeneration/total energy systems the following systems will be analyzed:

<u>System</u>	<u>Data Source</u>
1. Coal fired AFB Stirling	LSSE
2. Residual Fired Stirling	LSSE
3. Steam Turbine/AFB STM Coal AF	CTAS #1
4. Gas Turbine Residual CTAC12	CTAS #15
5. Diesel Residual DEADV	CTAS #25

3.2 Performance

The engine performance prediction has been determined by means of a second order analysis (1), a method which has been demonstrated accurate within $\pm 10\%$ of actual test data when appropriate correction factors are employed. This accuracy is deemed sufficient for a conceptual design phase, with the elaborate third order techniques to be used in the preliminary and final design phases.

Table 3.2-1 summarized the predictions for an individual cylinder of the conceptual design phase engine presented in this report, indicating a 126 horsepower (94KW) shaft (brake) output.

The prediction includes flow friction losses based on the V engine configuration, mechanical friction losses, effect of non-sinusoidal piston travel and pumping and static conduction losses. These last two are the largest heat losses due to the effects of the water cooled dual wall cylinder and the dual wall regenerator. The heat loss for smaller Stirling Engines usually falls in the 20 to 30% range, while the prediction here indicates that only 11% of the basic heat input is lost. This is generally true of all types of heat engines and is one of the prime reasons for a better performance for a larger size engine.

The brake engine efficiency as predicted here is compared with measured performances of previously existing engines, as shown in Fig. 3.2-2. It can be seen that the performance estimation of the present engine design is within the range of the previous engine measurement but on the conservative side.

(1) W. R. Martini and B. A. Ross, "An Isothermal Second Order Stirling Engine Calculation Method", paper #799237, IECEC, August 1979, and also "Stirling Engine Design Manual", NASA CR-135382, April, 1978

Table 3.2-1 Performance Summary at Engine Design Point

Engine Speed	1200 rpm
Mean Pressure	1500 psi
Hot Side Metal Temperature	760°C
Cold Side Metal Temperature	80°C
<hr/>	
Power (KW)/Cylinder	
Basic Power Output	127.9
Flow Friction	17.0
Indicated Power	110.9
Mechanical Friction	16.9
Brake Power	94.0
Heat Input (KW)/Cylinder	
Basic Heat Input	212.9
Reheat Loss	2.2
Shuttle Loss	1.3
Static Conduction	8.4
Pumping Loss	13.1
Temperature Swing Loss	1.9
Flow Friction Credits	-3.5
Net Heat Input	236.3
Indicated Efficiency	46.9%
Brake Efficiency	39.7%

FIGURE 3.2-2
STIRLING ENGINE PERFORMANCE SUMMARY

3.3 COST DATA

A cost analysis of the engine including auxiliaries, the heat transport system and electrical system components but not the combustor system has been prepared for use in a system life cycle analysis.

The detailed costing of the engine was performed using the RCA Price 84 computer program, which provides unit price and assembly to any predetermined detail level based on 29 inputs for each item desired. The inputs include the weight of the item, which has been calculated from design layouts in this case, a volume estimate, production quantity definition, material, tolerances, type classification, complexity estimate, schedule requirements, tooling requirements, reliability and many other factors as shown on the typical input sheet displayed in figure 3.3-1. The engine component weights were calculated for all seven engine configurations costed, to the level indicated in the sample computation sheet presented in figure 3.3-2, which is a breakdown of the drive system for a non-articulated, non-modular eight cylinder V engine. Figures 3.3-3 through 3.3-8 describe weight predictions of the many configurations considered in the costing exercise, and the weight range for an eight cylinder engine runs from 9388 pounds for a radial configuration to 13132 for an articulated modular V configuration. The price program output provides a cost evaluation to the performing operation level for development and production tasks and includes supplemental costs and schedule data. Figure 3.3-9 presents the program cost print out for a typical engine configuration with a production rate of 1000 engines per year after a development program including the fabrication of two prototype units. Schedule integration was provided to meet LSSE program requirements and feasibility was confirmed by the program.

The costs for eight cylinder engines are summarized in figure 3.3-10, and the radial engine proves to be the least expensive, the eight cylinder swashplate with journal bearings second and the non-articulated, non-modular V engine third. These are amortized costs based on three years of production at the specified rates, do not include the heat transport system and generator but do include engine auxiliaries. Since the four cylinder swashplate engine requires the use of two generators and some switch-gear for a 1000 horsepower output (two engine systems), a correction factor was added to those line costs to permit a realistic evaluation to be made. Figure 3.3-11 presents the specific cost of non-articulated V engines for various production rates, and also indicates the specific weight. Radial engine specific costs for an eight cylinder version are shown for comparison purposes.

FIG 3.3-1

PRICE

Worksheet

Basic ModesFile name: SECO
Sheet of

**PRICE 84 (This must be used only as the first line of the file.)

Title: V8-NA/11M BLOCK ASSEMBLYDate: 1-4-79

General A	Production Quantity QTY	Prototypes PROTOS	Weight (lbs) WT	Volume (ft ³) VOL	MODE	
	<u>(30)</u>	<u>2</u>	<u>3739</u>	<u>25</u>	<u>2</u>	
General B	Quantity/Next Higher Assembly QTYNHA	NHA Integration Factors Electronic INTEGE	Structural INTEGS	Specification Level PLTFM	Year of Economics YRECON	Year/Type of Technology YRTECH
	<u>1</u>	<u>0</u>	<u>.6</u>	<u>1</u>	<u>1980</u>	<u>1980</u>
Mechanical/ Structural	Structure Weight WS	Manufacturing Complexity MCPLXS	New Structure NEWST	Design Repeat DESRPS	Equipment Classification MECID	Mechanical Reliability MREL
	<u>3739</u>	<u>(5.043)</u>	<u>.6</u>	<u>0</u>	<u>100</u>	<u>0</u>
Electronics	Electronics Weight/ft ³ WECF	Manufacturing Complexity MCPLXE	New Electronics NEWEL	Design Repeat DESRPE	Equipment Classification CMPID	Electronic Reliability EREL
	<u> </u>	<u> </u>	<u> </u>	<u> </u>	<u> </u>	<u> </u>
Development	Development Start DSTART	1st Prototype Complete DFPRO 7 MO, 4.676	Development Complete DLPRO	Engineering Complexity ECMPLX	Tooling & Test Equip. DTLGTS	Prototype Activity PROSUP
	<u>1080</u>	<u>(1281)</u>	<u>C</u>	<u>(1.0)</u>	<u>0</u>	<u>0</u>
Production	Production Start PSTART	First Article Delivery PFAD	Production Complete PEND	Cost-Process Factor CPF	Tooling & Test Equip. PTLGTS	Rate/Month Tooling RATOOL
	<u>187</u>	<u>C</u>	<u>(1289)</u>	<u>.93</u>	<u>0</u>	<u>0</u>
Additional Data (Mode 10)	Electronic Volume Fraction USEVOL	Structural Weight/ft ³ WSCF	Target Cost TARCST			
	<u> </u>	<u> </u>	<u> </u>			

Notes: $.05 \times 3739 = 187^* \quad 4.61$ CHECK
 $.10 \times 3739 = 374^* \quad 4.66$ P=11 4.719 $1D=1$
 $.85 \times 3739 = 3178^* \quad 4.61$ $E = 3.08 (90\%)$
 $MCPLXS = 3.692$

4.34

Check

.15 Casting % E MCPLXS.3 Mechanism 5% 5 4.58.1 Drilling 45% 5.2 4.67.15 x-Pass 4% 5.3 4.7565.04 1.15 6 5.0431.0 x 1.5 x 5.043 = 7.5645

BASIC MODES

- 1 E/M ITEM
- 2 MECHANICAL ITEM
- 6 MODIFIED ITEM
- 10 DESIGN TO COST

GC 1013 2/79

Note: Inputs in shaded area are optional.

PRICE

TABLE 3.3-2

ARTICULATING V NON MODULAR	V8		V12		V16		V20		V24	
	Qty	WT	Qty	WT	Qty	WT	Qty	WT	Qty	WT
<u>DRIVE ASSEMBLY</u>										
CRANK SHAFT	1	443	1	626	1	810	1	993	1	1176
COUNTER WEIGHTS (51.2)	8	410	12	614	16	819	20	1024	24	1229
CROSS HEAD (51.6)	8	413	12	619	16	826	20	1032	24	1238
MAIN CONNECT ROD (42.0)	4	168	6	252	8	336	10	420	12	504
ARTICULAT CONNECT ROD (36.8)	4	147	6	221	8	294	10	360	12	442
WRIST PIN (15.9)	8	127	12	191	16	254	20	318	24	382
ARTICULAT PIN (6.4)	4	26	6	38	8	51	10	64	12	77
BRG - CRANK SHAFT (2.6)	5	13	7	18	9	23	11	29	13	34
BRG - WRIST PIN (1.7)	8	13	12	20	16	27	20	34	24	40
BAG - CONNECT ROD (2.8)	4	11	6	17	8	22	10	28	12	33
SADDLE - BAG CON ROD (13.4)	4	54	6	80	8	107	10	134	12	161
SADDLE - BAG CRANK SHAFT (20.1)	5	101	7	141	9	181	11	221	13	261
BRG - ARTICULATION PIN (1.1)	4	4	6	7	8	9	10	11	12	13
BALANCE SHAFTS	2	318	2	464	2	610	2	758	2	904
" " DRIVE GEAR	1	1	1	1	1	1	1	1	1	1
" " IDLER	1	1	1	1	1	1	1	1	1	1
" " BELT	1	2	1	2	1	2	1	2	1	2
TOTAL		2252		3312		4373		5430		6498
NOTE: • ALL JOURNAL BEARINGS ARE ASSUMED TO BE BABBITT AT .263 #/IN ³										
• CROSS HEAD IS ALUMINUM.										
• DRIVE BELT OF BALANCE SHAFT IS STEEL BLADE & RUBBER										

TABLE 3.3-3

ENGINE WEIGHT SUMMARY

ENGINE TYPE: NON ARTICULATED V-NON MODULAR

NUMBER OF CYLINDERS ITEM	4	8	12	16	20	24
PISTON ASSEMBLY	210	420	628	839	1,048	1,257
CYLINDER ASSEMBLY	1,649	3,298	4,948	6,597	8,246	9,895
DRIVE ASSEMBLY	1,000	1,848	2,696	3,544	4,394	5,242
BLOCK ASSEMBLY	2,484	3,739	4,994	6,249	7,506	8,761
HEAT TRANSFER ASSEMBLY	1,259	2,499	3,749	4,997	6,248	7,496
SUBTOTAL	6,602	11,804	16,943	22,226	27,442	32,651
ACCESSORIES	545	622	715	810	905	985
TOTAL	7,147	12,426	17,358	23,036	28,347	33,636

TABLE 3.3-4
ENGINE WEIGHT SUMMARY

ENGINE TYPE: NON ARTICULATED V-MODULAR

NUMBER OF CYLINDERS ITEM	4	8	12	16	20	24
PISTON ASSEMBLY		420	628	839	1,048	1,257
CYLINDER ASSEMBLY		3,298	4,948	6,597	8,246	9,895
DRIVE ASSEMBLY		2,205	3,244	4,283	5,325	6,365
BLOCK ASSEMBLY		3,910	5,386	6,862	8,340	9,816
HEAT TRANSFER ASSEMBLY		2,479	3,749	4,997	6,248	7,496
SUBTOTAL		12,332	17,955	23,578	29,207	34,757
ACCESSORIES		622	715	810	905	985
TOTAL		12,954	18,670	24,388	30,112	35,742

ENGINE WEIGHT SUMMARY

ENGINE TYPE:

ARTICULATED V-NON MODULAR

NUMBER OF CYLINDERS	4	8	12	16	20	24
ITEM						
PISTON ASSEMBLY		420	628	839	1,048	1,257
CYLINDER ASSEMBLY		3,298	4,948	6,597	8,246	9,895
DRIVE ASSEMBLY		2,252	3,312	4,373	5,430	6,498
BLOCK ASSEMBLY		3,753	5,040	6,326	7,617	8,903
HEAT TRANSFER ASSEMBLY		2,499	3,749	4,997	6,248	7,496
SUBTOTAL		12,222	17,677	23,132	28,589	34,049
ACCESSORIES						
TOTAL		12,222	17,677	23,132	28,589	34,049

TABLE 3.3-6

ENGINE WEIGHT SUMMARY

ENGINE TYPE: ARTICULATED V-MODULAR

NUMBER OF CYLINDERS ITEM	4	8	12	16	20	24
PISTON ASSEMBLY		420	628	839	1,048	1,257
CYLINDER ASSEMBLY		3,298	4,949	6,597	8,246	9,895
DRIVE ASSEMBLY		2,359	3,501	4,633	5,762	6,902
BLOCK ASSEMBLY		3,924	5,410	6,896	8,384	9,870
HEAT TRANSFER ASSEMBLY		1,499	3,749	4,997	6,248	7,496
SUBTOTAL		12,510	18,237	23,962	29,688	35,420
ACCESSORIES		622	715	810	905	985
TOTAL		13,132	18,952	24,772	30,593	36,405

TABLE 3.3-7

ENGINE WEIGHT SUMMARY

ENGINE TYPE:

RADIAL - MODULAR

NUMBER OF CYLINDERS ITEM	4	8	12	16	20	24
PISTON ASSEMBLY	210	420	628	839	1,048	1,257
CYLINDER ASSEMBLY	1,650	3,298	4,948	6,597	8,246	9,895
DRIVE ASSEMBLY	692	1,371	2,032	2,691	3,351	4,012
BLOCK ASSEMBLY	804	1,228	1,652	2,076	2,500	2,923
HEAT TRANSFER ASSEMBLY	1,250	2,499	3,749	4,997	6,248	7,496
SUBTOTAL	4,606	8,766	13,009	17,200	21,393	25,583
ACCESSORIES		622	715	810	905	985
TOTAL		9,388	13,724	18,010	22,298	26,568

TABLE 3.3-8

ENGINE WEIGHT SUMMARY

ENGINE TYPE:

SWASH PLATE

JOURNAL BRGS

NF BRG

NUMBER OF CYLINDERS ITEM	4	8	12	16	4	3
PISTON ASSEMBLY	211	420			211	420
CYLINDER ASSEMBLY	1,650	3,300			1,650	3,300
DRIVE ASSEMBLY	1,517	2,533			1,660	2,771
BLOCK ASSEMBLY	1,850	3,090			1,850	3,090
HEAT TRANSFER ASSEMBLY	1,250	2,499			1,250	2,499
SUBTOTAL	6,478	11,842			6,621	12,080
ACCESSORIES	568	655			568	655
TOTAL	7,046	12,497			7,189	12,735

FIG 3.3-9

- - - PRICE 84 - - -
MECHANICAL ITEM

DATE 17-JAN-80

TIME 14:21
(790184)

FILENAME: SE009.DAT

V8-NA/NM BLOCK ASSEMBLY 1-4-80

PRODUCTION QUANTITY	3000	UNIT WEIGHT	3739.00	MODE	2
PROTOTYPE QUANTITY	2.0	UNIT VOLUME	25.00	QUANTITY/NHA	1

UNIT PROD COST	84.76	COST PROCESS FACTOR	0	MONTHLY PROD RATE	117.93
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PROGRAM COST (\$ 1000)	DEVELOPMENT	PRODUCTION	TOTAL COST
ENGINEERING			
DRAFTING	135.	0.	135.
DESIGN	375.	0.	375.
SYSTEMS	78.	-	78.
PROJECT MGMT	101.	11398.	11500.
DATA	41.	1791.	1832.
SUBTOTAL (ENG)	731.	13189.	13920.
MANUFACTURING			
PRODUCTION	-	254295.	254295.
PROTOTYPE	685.	-	685.
TOOL-TEST EQ	0.	0.	0.
SUBTOTAL (MFG)	685.	254295.	254980.
TOTAL COST	1416.	267484.	268900.

DESIGN FACTORS	MECHANICAL	PRODUCT DESCRIPTORS	
WEIGHT	3739.000	ENGINEERING COMPLEXITY	1.000
DENSITY	149.560	PROTOTYPE SUPPORT	1.0
MFG. COMPLEXITY	5.043	PROTO SCHEDULE FACTOR	.600
NEW DESIGN	0.600	PLATFORM	1.0
DESIGN REPEAT	0.000	YEAR OF TECHNOLOGY	1980
EQUIPMENT CLASS	106.0+	RELIABILITY FACTOR	1.0
ENGINEERING CHANGES	0.000	MTBF (FIELD)	45424+
INTEGRATION LEVEL	0.6		

SCHEDULE	START		FIRST ITEM		FINISH	
DEVELOPMENT	OCT 80	(15)	DEC 81	(3)	MAR 82+	(18)
PRODUCTION	JAN 87	(15)	NOV 87+	(25)	DEC 89	(36)

SUPPLEMENTAL INFORMATION

YEAR OF ECONOMICS	1980
ESCALATION	0.00
T-1 COST	*****
AMORTIZED UNIT COST	89.16+
DEV COST MULTIPLIER	1.20
PROD COST MULTIPLIER	1.20

TOOLING & PROCESS FACTORS	
DEVELOPMENT TOOLING	0.0
PRODUCTION TOOLING	0.0
RATE TOOLING	.0
PRICE IMPROVEMENT FACTOR	.930
UNIT LEARNING CURVE	*****

COST RANGES	DEVELOPMENT	PRODUCTION	TOTAL COST
FROM	1213.	205969.	207182.
CENTER	1416.	267484.	268900.
TO	1806.	398676.	400482.

FIG 3.3-9 (Con't.)

- - - PRICE 84 - - -
MECHANICAL ITEM

DATE 17-JAN-80

TIME 14:23
(790184)

FILENAME: SE009.DAT

VS-NA/NM CYLINDER ASSEMBLY 1-4-80

PRODUCTION QUANTITY	26400	UNIT WEIGHT	412.30	MODE	2
PROTOTYPE QUANTITY	19.0	UNIT VOLUME	2.00	QUANTITY/NHA	2

UNIT PROD COST.	8.08	COST PROCESS FACTOR	0	MONTHLY PROD RATE	932.68
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PROGRAM COST (\$ 1000)	DEVELOPMENT	PRODUCTION	TOTAL COST
ENGINEERING			
DRAFTING	28.	0.	28.
DESIGN	88.	0.	88.
SYSTEMS	20.	-	20.
PROJECT MGMT	90.	11035.	11125.
DATA	24.	1713.	1737.
SUBTOTAL (ENG)	250.	12748.	12998.
MANUFACTURING			
PRODUCTION	-	213440.	213440.
PROTOTYPE	646.	-	646.
TOOL-TEST EO	53.	888.	941.
SUBTOTAL (MFG)	699.	214328.	215027.
TOTAL COST	949.	227076.	228025.

DESIGN FACTORS	MECHANICAL	PRODUCT DESCRIPTORS	
WEIGHT	412.300	ENGINEERING COMPLEXITY	1.000
DENSITY	206.150	PROTOTYPE SUPPORT	1.0
MFG. COMPLEXITY	5.043	PROTO SCHEDULE FACTOR	.600
NEW DESIGN	0.700	PLATFORM	1.0
DESIGN REPEAT	0.000	YEAR OF TECHNOLOGY	1980
EQUIPMENT CLASS	106.0+	RELIABILITY FACTOR	1.0
ENGINEERING CHANGES	0.000	MTBF (FIELD)	88014+
INTEGRATION LEVEL	0.6		

SCHEDULE	START		FIRST ITEM		FINISH	
DEVELOPMENT	MAY 81	(8)	DEC 81	(15)	MAR 83+	(23)
PRODUCTION	JAN 87	(8)	AUG 87+	(28)	DEC 89	(36)

SUPPLEMENTAL INFORMATION

YEAR OF ECONOMICS	1980
ESCALATION	0.00
T-1 COST	*****
AMORTIZED UNIT COST	8.60+
DEV COST MULTIPLIER	1.20
PROD COST MULTIPLIER	1.20

TOOLING & PROCESS FACTORS

DEVELOPMENT TOOLING	1.6
PRODUCTION TOOLING	1.2
RATE TOOLING	0
PRICE IMPROVEMENT FACTOR	.930
UNIT LEARNING CURVE	*****

COST RANGES	DEVELOPMENT	PRODUCTION	TOTAL COST
FROM	825.	186604.	187429.
CENTER	949.	227076.	228025.
TO	1150.	293476.	294626.

FIG 3.3-9 (Con't.)

- - - PRICE 84 - - -
MECHANICAL ITEM

DATE 17-JAN-80

TIME 14:25
(790184)

FILENAME: SE009.DAT

V8-NA/NM DRIVE ASSEMBLY 1-7-80

PRODUCTION QUANTITY	3000	UNIT WEIGHT	1848.00	MODE	2
PROTOTYPE QUANTITY	2.2	UNIT VOLUME	25.00	QUANTITY/NHA	1

UNIT PROD COST	46.96	COST PROCESS FACTOR	0	MONTHLY PROD RATE	113.40
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PROGRAM COST(\$ 1000)	DEVELOPMENT	PRODUCTION	TOTAL COST
ENGINEERING			
DRAFTING	14.	0.	14.
DESIGN	40.	0.	40.
SYSTEMS	10.	-	10.
PROJECT MGMT	35.	6508.	6543.
DATA	11.	1022.	1033.
SUBTOTAL (ENG)	110.	7530.	7640.
MANUFACTURING			
PRODUCTION	-	140885.	140885.
PROTOTYPE	355.	-	355.
TOOL-TEST EQ	31.	418.	449.
SUBTOTAL (MFG)	386.	141303.	141690.
TOTAL COST	496.	148833.	149329.

DESIGN FACTORS	MECHANICAL	PRODUCT DESCRIPTORS	
WEIGHT	1848.000	ENGINEERING COMPLEXITY	1.000
DENSITY	73.920	PROTOTYPE SUPPORT	1.0
MFG. COMPLEXITY	5.043	PROTO SCHEDULE FACTOR	.600
NEW DESIGN	0.600	PLATFORM	1.0
DESIGN REPEAT	0.800	YEAR OF TECHNOLOGY	1980
EQUIPMENT CLASS	106.0+	RELIABILITY FACTOR	1.0
ENGINEERING CHANGES	0.000	MTBF(FIELD)	56118+
INTEGRATION LEVEL	0.6		

SCHEDULE	START		FIRST ITEM		FINISH	
DEVELOPMENT	FEB 81	(11)	DEC 81	(3)	MAR 82+	(14)
PRODUCTION	JAN 87	(11)	OCT 87+	(26)	DEC 89	(36)

SUPPLEMENTAL INFORMATION		TOOLING & PROCESS FACTORS	
YEAR OF ECONOMICS	1980	DEVELOPMENT TOOLING	.6
ESCALATION	0.00	PRODUCTION TOOLING	.6
T-1 COST	*****	RATE TOOLING	0
AMORTIZED UNIT COST	49.61+	PRICE IMPROVEMENT FACTOR	.930
DEV COST MULTIPLIER	1.20	UNIT LEARNING CURVE	*****
PROD COST MULTIPLIER	1.20		

COST RANGES	DEVELOPMENT	PRODUCTION	TOTAL COST
FROM	422.	116880.	117302.
CENTER	496.	148833.	149329.
TO	629.	212202.	212831.

FIG 3.3-9 (Con't.)

- - - PRICE 84 - - -
MECHANICAL ITEM

DATE 17-JAN-80

TIME 14:27
(790184)

FILENAME: SE009.DAT

V8-NA/MM HEAT TRANSFER ASSEMBLY 1-7-80

PRODUCTION QUANTITY	26400	UNIT WEIGHT	306.00	MODE	2
PROTOTYPE QUANTITY	19.0	UNIT VOLUME	0.70	QUANTITY/NAH	8

UNIT PROD COST	5.93	COST PROCESS FACTOR	0	MONTHLY PROD RATE	913.16
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PROGRAM COST (\$ 1000)	DEVELOPMENT	PRODUCTION	TOTAL COST
ENGINEERING			
DRAFTING	24.	0.	24.
DESIGN	79.	0.	79.
SYSTEMS	23.	-	23.
PROJECT MGMT	96.	7968.	8063.
DATA	26.	1249.	1275.
SUBTOTAL (ENG)	247.	9217.	9464.
MANUFACTURING			
PRODUCTION		156521.	156521.
PROTOTYPE	495.	-	495.
TOOL-TEST EQ	46.	777.	823.
SUBTOTAL (MFG)	541.	157298.	157839.
TOTAL COST	788.	166515.	167303.

DESIGN FACTORS	MECHANICAL	PRODUCT DESCRIPTORS	
WEIGHT	306.000	ENGINEERING COMPLEXITY	1.200
DENSITY	437.143	PROTOTYPE SUPPORT	1.0
MFG. COMPLEXITY	4.927	PROTO SCHEDULE FACTOR	.600
NEW DESIGN	0.800	PLATFORM	1.0
DESIGN REPEAT	0.200	YEAR OF TECHNOLOGY	1980
EQUIPMENT CLASS	105.8*	RELIABILITY FACTOR	1.0
ENGINEERING CHANGES	0.000	MTBF (FIELD)	103691*
INTEGRATION LEVEL	0.6		

SCHEDULE	START		FIRST ITEM		FINISH	
DEVELOPMENT	APR 81	(9)	DEC 81	(15)	MAR 83*	(24)
PRODUCTION	JAN 87	(9)	JUL 87*	(29)	DEC 89	(36)

SUPPLEMENTAL INFORMATION

YEAR OF ECONOMICS	1980
ESCALATION	0.00
T-1 COST	*****
AMORTIZED UNIT COST	6.31*
DEV COST MULTIPLIER	1.20
PROD COST MULTIPLIER	1.20

TOOLING & PROCESS FACTORS	
DEVELOPMENT TOOLING	1.6
PRODUCTION TOOLING	1.4
RATE TOOLING	0
PRICE IMPROVEMENT FACTOR	.930
UNIT LEARNING CURVE	*****

COST RANGES	DEVELOPMENT	PRODUCTION	TOTAL COST
FROM	680.	136739.	137419.
CENTER	788.	166515.	167303.
TO	961.	216644.	217605.

FIG 3.3-9 (Con't.)

== PRICE 84 ==
MECHANICAL ITEM

DATE 17-JAN-80

TIME 14:29
(790184)

FILENAME: SE009.DAT

V8-NR NM PISTON ASSEMBLY 1-7-80

PRODUCTION QUANTITY	26400	UNIT WEIGHT	52.50	MODE	2
PROTOTYPE QUANTITY	19.0	UNIT VOLUME	1.00	QUANTITY/NHA	8
UNIT PROD COST	1.68	COST PROCESS FACTOR	0	MONTHLY PROD RATE	873.04

PROGRAM COST (\$ 1000)	DEVELOPMENT	PRODUCTION	TOTAL COST
ENGINEERING			
DRAFTING	10.	0.	10.
DESIGN	33.	0.	33.
SYSTEMS	10.	-	10.
PROJECT MGMT	31.	2219.	2250.
DATA	9.	349.	358.
SUBTOTAL (ENG)	93.	2568.	2661.
MANUFACTURING			
PRODUCTION	-	44269.	44269.
PROTOTYPE	141.	-	141.
TOOL-TEST EQ	10.	220.	230.
SUBTOTAL (MFG)	151.	44489.	44639.
TOTAL COST	243.	47057.	47300.

DESIGN FACTORS	MECHANICAL	PRODUCT DESCRIPTORS	
WEIGHT	52.500	ENGINEERING COMPLEXITY	1.250
DENSITY	52.500	PROTOTYPE SUPPORT	1.0
MFG. COMPLEXITY	5.043	PROTO SCHEDULE FACTOR	.600
NEW DESIGN	0.800	PLATFORM	1.0
DESIGN REPEAT	0.000	YEAR OF TECHNOLOGY	1980
EQUIPMENT CLASS	106.0+	RELIABILITY FACTOR	1.0
ENGINEERING CHANGES	0.000	MTBF (FIELD)	163331+
INTEGRATION LEVEL	0.6		

SCHEDULE	START	FIRST ITEM	FINISH
DEVELOPMENT	MAR 81 < 10	DEC 81 < 15	MAR 83+ < 25
PRODUCTION	JAN 87 < 10	JUN 87+ < 30	DEC 89 < 36

SUPPLEMENTAL INFORMATION

YEAR OF ECONOMICS	1980	TOOLING & PROCESS FACTORS	
ESCALATION	0.00	DEVELOPMENT TOOLING	1.0
T-1 COST	*****	PRODUCTION TOOLING	1.0
AMORTIZED UNIT COST	1.78+	RATE TOOLING	0
DEV COST MULTIPLIER	1.20	PRICE IMPROVEMENT FACTOR	.930
PROD COST MULTIPLIER	1.20	UNIT LEARNING CURVE	*****

COST RANGES	DEVELOPMENT	PRODUCTION	TOTAL COST
FROM	209.	38288.	38498.
CENTER	243.	47057.	47300.
TO	302.	43586.	43888.

FIG 3.3-9 (Con't.)

- - - PRICE 84 - - -
PURCHASED ITEM

DATE 17-JAN-80

TIME 14:31
(790184)

FILENAME: SE009.DAT

VS-NA/NM ACCESSORIES 1-7-80

PRODUCTION QUANTITY	3000	UNIT WEIGHT	675.0	MODE	3
PROTOTYPE QUANTITY	2.4	UNIT VOLUME	4.0	QUANTITY/NHA	1

PURCHASED ITEM UNIT COST 3.80

PROGRAM COST(\$ 1000)	DEVELOPMENT	PRODUCTION	TOTAL COST
PURCHASED ITEM	11.	13680.	13691.

DESIGN FACTORS	MECHANICAL	PRODUCT DESCRIPTORS	
WEIGHT	675.000♦	PLATFORM	1.0
DENSITY	168.750♦	YEAR OF TECHNOLOGY	1980
MFG. COMPLEXITY	3.482	RELIABILITY FACTOR	1.0
EQUIPMENT CLASS	♦♦♦♦♦	MTBF(FIELD)	248386♦
INTEGRATION LEVEL	0.6		

SUPPLEMENTAL INFORMATION			
YEAR OF ECONOMICS	1980	DEV COST MULTIPLIER	1.20
ESCALATION	0.00	PROD COST MULTIPLIER	1.20

- - - PRICE 84 - - -
INTEGRATION AND TEST

DATE 17-JAN-80

TIME 14:31
(790184)

FILENAME: SE009.DAT

VS-NA/NM STERLING ENGINE(I&T) 1-7-80

PRODUCTION QUANTITY	3000	INT WEIGHT	345.840♦	MODE	
PROTOTYPE QUANTITY	2	INT VOLUME	23.056♦	QUANTITY/NHA	0

UNIT PROD COST. 5.87

FIG 3.3-9 (Con't.)
 - - - PRICE 84 - - -
 INTEGRATION AND TEST

DATE 17-JAN-80

TIME 14:31
 (790184)

FILENAME: SE009.DAT

V8-NA/NM STERLING ENGINE (I&T) 1-7-80

PRODUCTION QUANTITY	3000	INT WEIGHT	345.840♦	MODE	5
PROTOTYPE QUANTITY	2	INT VOLUME	23.056♦	QUANTITY/NHA	0

UNIT PROD COST 5.87

MONTHLY PROD RATE 104.01

PROGRAM COST (\$ 1000)	DEVELOPMENT	PRODUCTION	TOTAL COST
ENGINEERING			
DRAFTING	23.	0.	23.
DESIGN	79.	0.	79.
SYSTEMS	25.	-	25.
PROJECT MGMT	35.	1164.	1200.
DATA	14.	185.	199.
SUBTOTAL (ENG)	176.	1350.	1526.
MANUFACTURING			
PRODUCTION	-	17618.	17618.
PROTOTYPE	74.	-	74.
TOOL-TEST EQ	0.	0.	0.
SUBTOTAL (MFG)	74.	17618.	17692.
TOTAL COST	250.	18967.	19218.

DESIGN FACTORS	MECHANICAL	PRODUCT DESCRIPTORS	
WEIGHT	345.840♦	ENGINEERING COMPLEXITY	1.323♦
DENSITY	15.000♦	PROTOTYPE SUPPORT	1.0
MFG. COMPLEXITY	4.949♦	PROTO SCHEDULE FACTOR	.600
NEW DESIGN	0.500	PLATFORM	1.0
DESIGN REPEAT	0.000	YEAR OF TECHNOLOGY	1980
EQUIPMENT CLASS	♦♦♦♦♦	RELIABILITY FACTOR	1.0
ENGINEERING CHANGES	0.000	MTBF (FIELD)	98565♦
INTEGRATION LEVEL	0.0		

SCHEDULE	START		FIRST ITEM		FINISH	
DEVELOPMENT	AUG 82	(10)	MAY 83	(17)	OCT 84	(27)
PRODUCTION	JAN 87	(7)	JUL 87♦	(29)	DEC 89	(36)

SUPPLEMENTAL INFORMATION		TOOLING & PROCESS FACTORS	
YEAR OF ECONOMICS	1980	DEVELOPMENT TOOLING	0.0
ESCALATION	0.00	PRODUCTION TOOLING	0.0
AMORTIZED UNIT COST	6.32♦		
DEV COST MULTIPLIER	1.20		
PROD COST MULTIPLIER	1.20		

COST RANGES	DEVELOPMENT	PRODUCTION	TOTAL COST
FROM	210.	15161.	15371.
CENTER	250.	18967.	19218.
TO	320.	25876.	26197.

FIG 3.3-9 (Con't.)

- - - PRICE 84 - - -
INTEGRATION AND TEST

DATE 17-JAN-80

TIME 14:33
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V8-NA/NM STERLING ENGINE (I&T) 1-7-80

TOTAL COST, WITH INTEGRATION COST			
PROGRAM COST (\$ 1000)	DEVELOPMENT	PRODUCTION	TOTAL COST
ENGINEERING			
DRAFTING	234.	0.	234.
DESIGN	694.	0.	694.
SYSTEMS	166.	0.	166.
PROJ MGMT	388.	40292.	40680.
DATA	126.	6309.	6435.
SUBTOTAL (ENG)	1607.	46601.	48209.
MANUFACTURING			
PRODUCTION	-	827028.	827028.
PROTOTYPE	2396.	-	2396.
TOOL-TEST EQ	140.	2303.	2443.
PURCH ITEMS	11.	13680.	13691.
SUBTOTAL (MFG)	2547.	843011.	845558.
TOTAL COST	4155.	889612.	893767.
COST RANGES			
FROM	3571.	713320.	716892.
CENTER	4155.	889612.	893767.
TO	5179.	1224139.	1229319.

FIG 3.3-9 (Con't.)

- - - PRICE 84 - - -

DATE 8-JAN-80

TIME 09:46
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FILENAME: SE009.DAT

PROGRAM COST (\$ 1000)	DEVELOPMENT	PRODUCTION	TOTAL COST
V8-NA/NM BLOCK ASSEMBLY 1-4-80			
TOTAL COST	1416.	267484.	268900.
V8-NA/NM CYLINDER ASSEMBLY 1-4-80			
TOTAL COST	949.	227076.	228025.
V8-NA/NM DRIVE ASSEMBLY 1-7-80			
TOTAL COST	496.	148833.	149329.
V8-NA/NM HEAT TRANSFER ASSEMBLY 1-7-80			
TOTAL COST	788.	166515.	167303.
V8-NA/NM PISTON ASSEMBLY 1-7-80			
TOTAL COST	243.	47057.	47300.
V8-NA/NM ACCESSORIES 1-7-80			
TOTAL COST	11.	13680.	13691.
V8-NA/NM STERLING ENGINE (I&T) 1-7-80			
TOTAL COST	250.	18967.	19218.

- - - PRICE 84 - - -

DATE 8-JAN-80

TIME 09:46
(790184)

FILENAME: SE009.DAT

TOTAL COST* WITH INTEGRATION COST			
PROGRAM COST (\$ 1000)	DEVELOPMENT	PRODUCTION	TOTAL COST
ENGINEERING			
DRAFTING	234.	0.	234.
DESIGN	694.	0.	694.
SYSTEMS	166.	0.	166.
PROJ MGMT	388.	40292.	40680.
DATA	126.	6309.	6435.
SUBTOTAL (ENG)	1607.	46601.	48209.
MANUFACTURING			
PRODUCTION	-	827028.	827028.
PROTOTYPE	2396.	-	2396.
TOOL-TEST EQ	140.	2303.	2443.
PURCH ITEMS	11.	13680.	13691.
SUBTOTAL (MFG)	2547.	843011.	845558.
TOTAL COST	4155.	-141-	893767.

FIG 3.3-9 (Con't.)

1000 ENGINES PER YEAR PLUS SPARES

	ENGINEERING			
030	DRAFTING	23.	0.	23.
0390	DESIGN	79.	0.	79.
00	SYSTEMS	25.	-	25.
03410	PROJECT MGMT	35.	1164.	1200.
03420	DATA	14.	185.	199.
03430	SUBTOTAL (ENG)	176.	1350.	1526.
03440				
03450	MANUFACTURING			
03460	PRODUCTION	-	17618.	17618.
03470	PROTOTYPE	74.	-	74.
03480	TOOL-TEST EQ	0.	0.	0.
03490	SUBTOTAL (MFG)	74.	17618.	17692.
03500				
03510	TOTAL COST	250.	18967.	19218.
♦TEXT F-L, UNIT PROD COST				
00210	UNIT PROD COST	84.76	COST PROCESS FACTOR	0 MONTHLY PROD RATE
E 117.93				
00780	UNIT PROD COST	8.08	COST PROCESS FACTOR	0 MONTHLY PROD RATE
E 932.68				
01350	UNIT PROD COST	46.96	COST PROCESS FACTOR	0 MONTHLY PROD RATE
E 113.40				
01920	UNIT PROD COST	5.93	COST PROCESS FACTOR	0 MONTHLY PROD RATE
E 913.16				
02490	UNIT PROD COST	1.68	COST PROCESS FACTOR	0 MONTHLY PROD RATE
E 873.04				
03340	UNIT PROD COST	5.87		MONTHLY PROD RATE
104.01				

TEXT F-L, AMORTIZED UNIT COST			
00590	AMORTIZED UNIT COST	89.16♦	RATE TOOLING
0			
01160	AMORTIZED UNIT COST	8.60♦	RATE TOOLING
0			
01730	AMORTIZED UNIT COST	49.61♦	RATE TOOLING
0			
02300	AMORTIZED UNIT COST	6.31♦	RATE TOOLING
0			
0	AMORTIZED UNIT COST	1.78♦	RATE TOOLING
0			
03210	AMORTIZED UNIT COST	6.32♦	PRODUCTION TOOLING
0.0			

TABLE 3.3-10

COST* AND WEIGHT COMPARISON FOR A 1000 HP UNIT

WEIGHT LB	CONFIGURATION	PRODUCTION RATE, UNITS PER YEAR				RANKING
		10	100	1000	10,000	
12,426	V8 NON-ARTICULATED SINGLE BLOCK & SHAFT	510.3	389.6	296.5	230.8	3
12,222	V8 ARTICULATED SINGLE BLOCK & SHAFT	525.1	400.8	304.9	237.1	5
9,388	RADIAL - 8	399.4	304.5	231.6	180.2	1
12,497	SWASHPLATE - 8 JOURNAL BRGS	510.2	389.3	296.1	230.2	2
12,735	SWASHPLATE - 8 NF - BRGS	511.0	390.4	297.3	231.5	4
7,189	SWASHPLATE - 4 2 UNITS** NF-BRG	587.5	454.9	352.9	280.8	6

* AMORITIZED UNIT COST + SPARES INCLUDING ACCESSORIES W/O HEAT TRANSPORT LOOP

** ADJUSTED FOR DUAL GENERATOR AND SWITCHGEAR ($\Delta = 23.8K$)

FIG 3.3-11

3.4 Levelized Annual Cost of Energy

The results of the calculation were not available at the time of the issue of this report mainly due to the fact that it was hoped that more reliable data for the heat source became available. Therefore, this data will be issued in the final report, including the comparison of a Stirling system with a steam turbine system in the 1 to 10 MW range.

3.5 Candidate Configuration Selection

3.5.1 Engine Cost Comparison

The engine weight and cost data presented in Section 3.3 are summarized in Table 3.5-1 for an 8 cylinder 1000 HP engine*. The cost includes all auxiliaries which are shown in the system schematic (Figure 2.1-1) but not the heat transport system.

Depending on which heat transport system is chosen, \$25/HP for a heat-pipe system or \$62/HP for a pressurized gas transport loop has to be added.

It can be seen that there are clearly 3 weight/cost groupings:

- 1) The radial engine which, due to its' largely reduced crankcase and crankshaft weight, shows a weight saving of more than 20% compared to the closest V-engine. This reduction in weight is clearly reflected in the cost approach for a stationary Stirling engine.

This engine is also ideally suited for a Rina type system since each cylinder row--four cylinders at right angles in one plane--represents a complete 500 HP Rina block with 90° phase angle. Cylinder rows can then be stacked up (up to 6 rows) for engine output ranging from 500 to 3000 HP.

- 2) The second group of engines falling within a few percent in weight and cost are both V-crank engines (the non-articulated version being slightly superior) and the 8 cylinder swashplate engine. For this engine no detailed layout was prepared but drive train weight was estimated by scaling from the layout of the four cylinder swashplate version according to the diameter ratio (cylinder and HX-hardware being the same).
- 3) A surprisingly inferior weight and cost characteristic is exhibited by the 4 cylinder swashplate engine. The inferior cost performance is caused mainly by the additional generator and switchgear hardware ($\approx$ \$23/HP) which is necessitated since modularization with a gear box is even less cost effective ($\approx$ \$32/HP).

As discussed in Section 3.3, the engine costs were calculated for the entire horsepower range for a production rate of 1000 units/year and for a 1000 HP unit as a function of production rate ranging from 10 to 10,000 units per year. These data are shown in Figure 3.5-1 for the non-articulated V-engine (1000 HP). As a comparison, the radial engine data are given for a 1000 HP unit; also shown are GE locomotive Diesel engine data for a 2000 HP unit. It has to be noted that the costs for the Stirling engines were calculated for a first generation engine while the cost of the Diesel engine represents a mature hardware which has increased its output during a 20 year evolutionary development--from 1250 HP to 2000 HP without substantial increase in weight and cost (constant dollars).

can be anticipated that the output per cylinder will be nearly bled considering only the evolution to 200, atmospheres mean pressure and 1800 RPM, which are quite feasible but are not recommended for the first generation hardware. It is interesting to note that the specific weight (lb. per horsepower) is slightly lower for the Stirling engine which is not too surprising since the valving and camshaft of the Diesel, which are not only considerable weight contributors by themselves but also causes a considerable size and weight addition to the block assembly.

3.5.2 Comparison of Qualitative Criteria

Using the qualitative selection criteria and methodology outlined in Section 3.1 and Appendix B, several qualified individuals from the review and design team rated the candidate systems. The individual results were quite close and the combined results are shown in Table 3.5-2 for all engines investigated. In the following paragraphs these ratings will be discussed for each engine. To simplify this discussion it will be assumed that a rating of 8 (ideally meets all requirements) is normal and comments will be limited to deviations from this rating.

3.5.2.1 Non-Articulated V-Engine

This engine ideally meets all requirements and only was judged to be somewhat inferior in vibration to a swashplate or rhombic drive engine and therefore was rated somewhat lower (7) in this category. With an overall rating of 7.95, the engine has the highest rating and therefore is ranked as number one in the qualitative evaluation.

3.5.2.2 Articulated V-Engine

This engine was rated somewhat inferior to the non-articulated engine due to the following considerations:

- 1) The reliability was judged to be slightly lower due to the mechanical complexity of the articulated connection rod design and also due to increased higher harmonic vibrations. These are difficult to compensate and contribute to higher vibration levels, prone to result in higher mechanical loads on crankshaft, bearings and other equipment (seals!) and therefore resulting in a decreased reliability.
- 2) Since the articulated connecting rods are somewhat more difficult to disassemble and affect the opposite rod as well during disassembly, the maintainability was reduced (7) compared to the non-articulated design where a crosshead removal does not affect the opposite piston and therefore results in an easier and less time consuming exchange for crossheads, crosshead pins and connecting rod bearings.
- 3) The crosshead diameter is larger (9 inches vs. 7 inches for the non-articulated design) in order to allow crossheads/connecting rod exchange from the top of the engine. This results in higher mass forces on bearings and crankshaft and also increasing vibration levels.

- 4) Since the slightly decreased engine length (no cylinder offset) does not compensate for these drawbacks the engine was judged to have a slightly reduced viability for a 1985 demo engine, and was ranked in second place.

3.5.2.3 Radial Engine

As discussed in Section 2.4-3 the radial approach is ideally suited for a Rinja type engine, and resulted in the lowest cost and weight design. This engine, however, exhibits some drawbacks which makes this engine less desirable for a 1985 demonstration engine.

- 1) Similar to the articulated V-engine the reliability was assessed lower (7) due to the 3 articulated connecting rods (all connecting rods acting on the same crank), again resulting in higher vibration levels, and increased crosshead diameter.
- 2) Maintainability was rated lower (6) due to the increased difficulty for exchange of the connecting rod bearing and the considerable disassembly work in order to exchange the master connecting rod crosshead where all 3 articulated rods have to be disassembled from the master rod.

The main disadvantage from a maintenance standpoint is the limited access to the cylinders on a multi-block engine (8, 12, 16... cylinder) even if a vertical axis is used since the heat transfers equipments and the connecting lines of the heat transport system significantly impede the access to the lower rows of cylinders. (Note: a cylinder assembly weighs approximately 410 lbs. and requires crane assistance).

- 3) The requirements for a vertical axis (otherwise the lower cylinder are practically inaccessible) reduces the market for this engine since a large number of applications are geared for a horizontal axis.
- 4) Since the radial engines are not commonly used, their market acceptance will certainly be lower due to the lower familiarity with this engine.
- 5) Adaptability for this engine to an advanced version engine was rated high due to the high cost and weight advantage. It is envisioned that this engine could be included later in a program since the altered block and crankcase is a moderate deviation in an existing V-engine production program. Thus, the cost and weight advantages can enhance the competitive position of the Stirling engine where the vertical axis has no adverse consequences. This engine will therefore be investigated as an advanced concept. This engine was rated third overall.

3.5.2.4 Swashplate Engine

The highest uncertainty for a swashplate engine exists in the drive train design since high power swashplate engines have not been reduced to practice in a large scale and experience on experimental units is inaccessible. A sizable effort was made to obtain applicable data but most personnel connected with earlier (50's) developments are retired and very little formal documentation exists. The experience made on the automotive (Ford) swashplate engine was not encouraging and future developments may not use this approach.

According to private communications the only successful large size swashplate engine was developed in France (early 50's) but financial consideration (entirely new production facilities) prevented an introduction into the market. The rating of this engine was considerably lower and it is not considered a viable approach to large stationary Stirling engines at this time.

- 1) Uncertain reliability of the drive train and no available design and hardware experience.
- 2) Significantly reduced maintainability due to limited access to crosshead/swashplate interface bearings.
- 3) Limited shaft access for auxiliary equipment.
- 4) Modularity difficult; separate units required.
- 5) Market acceptability very questionable. Requires new skills in production assembly and maintenance.
- 6) Compactness and good vibration characteristics do not compensate for the inherent drawbacks.
- 7) The journal bearing engine was rated slightly higher due to the increased accessibility and ease of exchange of the shaft bearings
- 8) It appears unadvisable to superimpose a drive train development on top of the specific Stirling component developments for the first generation.

These engines are considered a non-viable approach for first generation hardware and don't exhibit large promise for near term advanced development.

3.5.2.5 Rhombic Drive Engine

Even though this type of engine was eliminated earlier due to an intolerable maintenance problem on the push rod seals, the engine was included in the qualitative rating for completeness and comparison.

3.5.2.6 Heat Transport Loop Rating

Even though not all qualitative rating categories apply to the heat transport system a relative rating was attempted and is included in Table 3.5-2. The totals were adjusted to account for the missing categories.

From this rating it seems that a gas fired version would be preferable for an early test unit. A pressurized gas loop version appears advantageous for early field emplacement.

Due to simplicity and no requirement for auxiliary power the heat pipe indicates excellent possibilities for early advanced versions.

TABLE 3.5-1

COST* AND WEIGHT COMPARISON FOR A 1000 HP UNIT

WEIGHT LB	CONFIGURATION	PRODUCTION RATE, UNITS PER YEAR				RANKING
		10	100	1000	10,000	
12,426	V8 NON-ARTICULATED SINGLE BLOCK & SHAFT	510.3	389.6	296.5	230.8	3
12,222	V8 ARTICULATED SINGLE BLOCK & SHAFT	525.1	400.8	304.9	237.1	5
9,388	RADIAL - 8	399.4	304.5	231.6	180.2	1
12,497	SWASHPLATE - 8 JOURNAL BRGS	510.2	389.3	296.1	230.2	2
12,735	SWASHPLATE - 8 NF - BRGS	511.0	390.4	297.3	231.5	4
7,189	SWASHPLATE - 4 2 UNITS** NF-BRG	587.5	454.9	352.9	280.8	6

* AMORITIZED UNIT COST + SPARES INCLUDING ACCESSORIES W/O HEAT TRANSPORT LOOP

** ADJUSTED FOR DUAL GENERATOR AND SWITCHGEAR ($\Delta = 23.8K$)

TABLE 3.5-2
QUALITATIVE EVALUATION RESULTS

	WEIGHT FACTOR	"V" ARTICULATED	"V" NON-ARTICULATED	SWASHPLATE JOURNAL BRGS	SWASHPLATE A.F. BRGS	RADIAL	OPPOSED RHOMBIC	PRESSURIZED GAS TRANSPORT SYSTEM	HEAT PIPE	DIRECT GAS FIRED
RELIABILITY	.3	7.5 2.	8 2.4	6 1.8	6 1.8	7 2.1	5 1.5	6 1.8	7 2.1	7 2.1
MAINTAINABILITY	.2	7 1.4	8 1.6	5 1.0	4 .8	6 1.2	4 .8	7 1.4	5 1.0	8 1.6
APPLICATION/INTERFACE FLEX.	.1	8 .8	8 .8	6 .6	6 .6	6 .6	5 .5	7 .7	5 .5	8 .8
FUEL FLEXIBILITY	.1	8 .8	8 .8	8 .8	8 .8	8 .8	8 .8	-	-	-
NOISE/VIBRATION	.05	6 .3	7 .35	8 .4	8 .4	6 .3	9 .45	-	-	-
MARKET ACCEPTABILITY	.15	8 1.2	8 1.2	5 .75	5 .75	6 .9	5 .75	6 .9	4 .6	8 1.2
VIABILITY FOR '85 DEMO.	.05	7 .35	8 .4	4 .2	4 .2	6 .3	5 .25	7 .35	6 .3	8 .4
ADAPTABILITY FOR ADV DEVELOP	.05	8 .4	8 .4	5 .25	5 .25	9 .45	5 .25	6 .3	8 .4	-
TOTAL		7.	7.95	5.8	5.6	6.65	5.3	5.45	4.9	6.1
RANKING		2	1	4	5	3	6	-	-	-

RATING: 8 TO 10 MEETS IDEAL STANDARDS (MEETS ALL GOALS)
4 TO 8 ACCEPTABLE (MEETS REQUIREMENTS BUT NOT GOALS)
0 TO 4 UNACCEPTABLE (DOES NOT MEET REQUIREMENTS)

4. System Selection and Recommendations

Based on the results presented in Section 3, the V configuration is selected as the most viable approach for a 1985 demonstration unit for a large stationary Stirling engine program.

The main factors for this selection are:

- Well developed drive train with all analytical and manufacturing techniques available.
- Production facilities for all drive train and auxiliary hardware in place.
- Existing skills for production assembly and maintenance.
- Increased market acceptability due to familiarity with V configuration and components.
- Superior maintainability characteristics due to good cylinder and seal accessibility and easy exchange of crank shaft and connecting rod bearings.
- Good adaptability for many applications.
- Ease of integration with heat source by pressurized gas heat transport system or heat pipe.
- Adaptable to direct gas fired configurations.
- Good modularity.

The chosen design approach stressed reliability and maintainability by using the following design approach.

- Individually exchangeable cylinders and push rod seal cartridges
- Exchangeable HX units with the same removable regenerator and cooler for all heat input systems.
- Use of a low cost cylinder material through water cooled design.
- Use of low temperature seals.

Several areas require further detailed investigation and analysis. These will be further discussed in the final report; some of the more pressing topics needing attention are listed below:

- Reliable seals for this size engine are not state-of-the-art and further design and development testing is needed before a final design can be chosen.

- Reliable heat source cost data are not available making an economic assessment difficult. Economically viable designs for the total energy/cogeneration market have to be established, which fall in the desired output range and address problems such as modularity, turn-down ratio and coal handling equipment for small outputs.

A heat transport loop is a necessary component to achieve good fuel and system flexibility.

Heat pipes of the required size (250 KW thermal) are not state-of-the-art and exhibit certain operational problems which are not sufficiently resolved.

A pressurized gas heat transport loop suffers from the non-availability of suitable high temperature gas circulators at a pressure level of approximately 40 atmospheres. Furthermore, such a system adds approximately 60 \$/HP to the overall engine cost (vs. 25 \$/HP for the heat pipe system) and has an auxiliary power consumption of approximately 5% of engine shaft power.

A considerable design and development effort is required to close the gap to a truly state-of-the-art heat transport equipment.

APPENDIX A

PROGRAM INFORMATION REQUEST / RELEASE

CLASS. LTR.	OPERATION	PROGRAM	SEQUENCE NO.	REV. L.
U -	SSE	79	001	A -
PIR NO.				
*USE "C" FOR CLASSIFIED AND "U" FOR UNCLASSIFIED				

FROM R.R. Laessig		TO Distribution	
DATE SENT REV. A - 11/7/79 11/1/79	DATE INFO. REQUIRED REV. B - 12/12/79	PROJECT AND REQ. NO.	REFERENCE DIR. NO.
SUBJECT Large Stationary Stirling Engine Concept Comparison and Selection			

INFORMATION REQUESTED/RELEASED

1. Configuration Parameters for Candidate Configurations

Before selecting several candidate engine configurations, it is necessary to establish all the various parameters and possible configuration variations viable for a Stirling engine, and then select promising combinations for further detailed evaluation. Table 1 lists all the various parameters and configurations which were considered for the selection of the candidate configurations. Figure 1 gives a simple outline for various configurations.

Before selecting various engine arrangements, a few basic remarks will be made concerning the economics of large Stirling engine in future total energy and cogeneration systems.

1. The introduction of a cogeneration system is only viable for engines which have high reliability (availability factor $>90\%$) and are used a large fraction of the time ($>60\%$). This dictates high reliability combined with a fast and easy replacement of parts, i.e., good accessibility and interchangeability of all components.
2. When oil or gas is used as fuel, diesel engines will most likely out-rank the Stirling engine with respect to the return on investment (or levelized cost of energy) due to the increased cost for the heat source and the higher heat source integration complexity. Therefore, the most viable application for large stationary Stirling engines will be a coal or refuse fired version where the reduction in fuel costs more than offset the higher hardware and installation costs. Therefore, prime consideration for the selection of viable engine configurations will be the integration to a coal fired heat source since this is considered the most viable path for the commercialization of large Stirling engines.

W. Auxer	W. Richards	PAGE NO. 1 OF 21	RETENTION REQUIREMENTS	
W. Chiu	H. Straub		COPIES FOR	MASTERS FOR
M. Concannon	R. Laessig (2)	☐ 1 MO.	☐ 3 MOS.	
L. Dutram	K. Uherka)	☐ 3 MOS.	☐ 6 MOS.	
	R. Holtz) Argonne Labs.	☐ 6 MOS.	☐ 12 MOS.	
		☐ MOS.	☐ MOS.	
		☐	☐ DO NOT DESTROY

OPPOSED

V

IN-LINE

U

RADIAL

W

INVERTED W

RINIA ARRANGEMENT

RHOMBIC

SWASHPLATE

FIGURE 1. DRIVE AND CYLINDER CONFIGURATION

Table I. Configuration Parameters

1.0 TYPE OF DRIVE

- 1.1 Crank
- 1.2 Swashplate
- 1.3 Rhombic
- 1.4 Free Piston

2.0 ARRANGEMENT OF CYLINDERS

- 2.1 In-Line
- 2.2 V-Type
- 2.3 opposed (V-angle 180°)
- 2.4 Radial
- 2.5 U (2 rows parallel, 2 drives)
- 2.6 Circular
- 2.7 W-configuration
- 2.8 Inverted W

3.0 CYLINDER DESIGN

- 3.1 Single Acting
- 3.2 Double Acting

4.0 MODULE ARRANGEMENT

- 4.1 In-Line
- 4.2 Double Row
- 4.3 Radial
- 4.4 Circular

5.0 HEAT TRANSPORT LOOP

- 5.1 Heat Pipe
- 5.2 Reflux Boiler
- 5.3 pumped Liquid Metal Loop
- 5.4 Pressurized Gas
- 5.5 Direct (Gas Fired)
- 5.6 Radiation

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2. Discussion of Configuration Parameters

2.1 Type of Drive

2.1.1 Crank Drive

The crank drive is certainly the most developed drive configuration for reciprocating engines with a well established state-of-the-art in design of bearings, balancing, lubrication, bending and torsional vibrations, and dynamic fatigue behavior. The simplicity, cost effectiveness and high mechanical efficiency is the reason that it is the most universally used drive system for reciprocating engines.

The vast experience available from use in large stationary diesel engines as well as locomotive and shipboard engines will definitely make this type of drive system a viable candidate for large stationary Stirling engines.

It has to be also considered that well established manufacturing procedures and facilities as well as trained maintenance personnel exist for this type of drive, which definitely eases the commercial introduction of a new type of engine. Its accessibility for maintenance and repair and the possibility of using existing designs with a well established reliability further enhances the viability of this type of drive, and minimizes cost and schedule risk.

2.1.2 Swashplate Drive

Swashplate drives have been utilized for many applications ranging from hydraulic pumps to steam engines. The most recent developments of swashplate drives for automotive Stirling engines were undertaken in the design of the Philips/Ford 4-215 engine and a study on the 4-98 engine. Therefore, a swashplate drive will be further evaluated especially for advanced engine concepts. It has to be recognized however, that the state-of-the-art especially for large stationary engines is by far not as well established as the crank drive. Furthermore, these drives have been tried on diesel engines but were either technically or from a production cost standpoint not successful and therefore have never been introduced into commercial reciprocating engines.

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The recent interest in swashplate and wobble plates for automotive Stirling engine drives stems mainly from the fact that the desire to arrive at a cylinder head configuration which allows the integration of all (4) cylinders into one heat source. The circular cylinder arrangement of a swashplate drive lends itself perfectly for a concentrically located burner around the heater heads, and also results in a minimum dead volume considering the connection between cylinders for double acting (Rinia) configurations. A further advantage of this type of drive is that the piston forces for a four cylinder engine are almost perfectly balanced which results in a low vibration engine.

The main drawbacks for this type of drive are:

- a. Lower mechanical efficiency due to increased drive friction (especially at low speeds)
- b. No simple take-off for auxiliary power (water and oil pumps, fans)
- c. Limited access to the drive and cylinder for maintenance and repair
- d. Necessity of gear box or chain drive to connect several modules into one generator.

Therefore, all factors especially cost and reliability have to be very carefully analyzed to establish whether this type of drive lends itself for use in a large stationary Stirling engine.

2.1.3 Rhombic Drive

The rhombic drive is probably the most widely used drive system for Stirling engines to date. Its main advantage is:

- a. Possibility of single cylinder engine
- b. Excellent balancing
- c. Elimination of cross-heads.

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The obvious drawbacks are mechanical complexity resulting in higher cost; due to double drive shaft, double seal (for both rods); the displacer rod seal mounted in the piston drive itself.

Since every cylinder with rhombic drive is a self-contained unit, the possibility exists to use fewer cylinders per module (the double acting Rinia configuration has to use at least 4 cylinders per module) thus counter-acting the increased drive cost. By using an opposed drive, two cylinders can be driven by one Rhombic drive which reduces the cost and complexity disadvantage even further. Therefore, only an opposed cylinder single drive configuration (one drive for both opposed cylinders) will be considered for further evaluation.

The main areas of concern (besides the seal design) for a two piston 500 HP module is the rotational speed variation within one cycle and the integration with the heat source (500 KW thermal/cylinder) which stretches the state-of-the-art for a single heat pipe.

2.1.4 Free Piston Engine

This type of drive configuration lends itself only for small engines and will not be considered for large stationary engines considering the large moving masses, difficulties for power take-off and engine starting.

2.2 Arrangement of Cylinders

The main aspects for cylinder arrangement are the ease of interconnection between cylinders and the integration to the heat source, as well as ease of maintenance and replacement.

2.2.1 In-Line Engine

Engines with "in-line" cylinder arrangements have the main advantage of easy access to both sides of the cylinder and also the vertical cylinder axis allows easy assembly and disassembly (with overhead crane), due to the vertical axis.

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11/1/79

The main disadvantage is the length of the engine and the comparatively high weight of crankshaft, housing and foundation per cylinder. Especially the large length (a 24 cylinder, 3000 HP "in-line" engine would have a total length of approximately 30 feet) would create additional problems for the integration with the heat source and significantly increase the space requirements; also the torsional stability of the drive system (crankshaft) is a major consideration for large stationary engines since long crankshafts can cause dynamic instabilities when coupled with a synchronous generator. The possibility of using two parallel modules requires either an additional gear box or a second generator with the additional switch gear necessary.

Considering these drawbacks and the lower cost effectiveness (compared to a V or opposed arrangement), the "in-line" cylinder arrangement will not be further pursued for this engine.

2.2.2 V-Engine

Large stationary reciprocating engines (except for very large shipboard engines) have favored the V-type engine over a long development period.

This mainly stems from the inherent advantages of the V-engine:

- Reduced length - i.e., short block and crankshaft with good torsional stability
- Low specific weight (#/hp)
- Good accessibility to the drive and cylinder
- Ease of assembly or interchange of cylinders (with overhead crane).

It is interesting to note that in most recent automotive engine development, the small V-type engine (V-6 and V-4) have helped to decrease engine weight and space requirements.

SSE-79-001

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Considering these advantages and the substantial experience and well developed state-of-the-art in design, balance and reliability, this engine type will be considered as a main candidate for the large stationary Stirling engine configuration.

2.2.3 Opposed Cylinder

Opposed cylinder arrangements have basically the same advantage as V-type arrangements. The main drawbacks are:

- The increased connecting length (dead volume) between cylinders for Rinja type Stirling engines
- Increased complexity for cylinder interchange (horizontal cylinders are inherently more difficult to mount in large engines considering connection rods and cross-heads)
- More difficult to integrate to heat source.

If a rhombic drive is used, the opposed cylinder arrangement allows use of a single drive for two cylinders which results in cost and weight reduction. Therefore the opposed cylinder arrangement will only be used in connection with a rhombic drive system since otherwise the V-type arrangement is more advantageous.

2.2.4 Radial Engine

Radial engines have been extensively used for aircraft engines with the use of air cooling.

For large stationary engines, main disadvantages are:

- The limited access to the drive (considering that one end is blocked by the generator and the other by the auxiliaries)
- For horizontal axis engines, the difficulty of interchanging parts or cylinders on the lower cylinders (underneath the engine)
- For vertical axis engines the special foundation requirements and the mounting of generator and/or auxiliaries.

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Since a 4 cylinder radial engine would be well suited for a basic Rinia module and the engine constitutes basically a 90° double V with most of the advantages of the V-configuration, this type of drive is considered a viable alternate and will be selected for further study.

2.2.5 U-Type Engine

U-type engines have been successfully used in Stirling engine design of United Stirling of Sweden for automotive type engines.

Similar to the swashplate drive, this configuration has its main advantage in the close proximity of cylinders which allows the integration of the cylinders into one heat source (concentric burner) and results in short cylinder connections (small dead volume). The double crankshaft also allows ample opportunities for auxiliary power take-off and has good accessibility to the drive. Main drawback of this configuration is the double crank which has a significant cost impact for large engines and the connecting gear drive with load reversals. For large stationary engines, the heat input per cylinder is sufficiently large (≈ 240 KW thermal) that an individual heat input loop is most viable; this is especially true for heat pipes as heat transport loops were 240 KW and is stretching the state-of-the-art somewhat. Thus the proximity of the cylinder heads has no advantage and the cost of a double crankshaft and connecting gear drive with larger engine block is not compensated by any other cost advantages.

Also considering the desirability of separate maintenance for cylinders and heater head heat exchangers for large stationary engines, integration of the heater head heat exchanger is not a preferred solution. It therefore was concluded that there is not incentive to use a "U" configuration for stationary engines and it will not be used for further configuration trade-off studies (it is interesting to note that the MTI trade-off study for automotive engines showed that even considering the advantage in burner design, the V-engine was slightly superior in weight and power performance compared to the U-configuration).

SSE-79-001
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2.2.6 Circular Configuration

This cylinder arrangement is specifically tied to the swashplate drive system where all cylinders, by necessity, are arranged in a circular pattern around the shaft centerline. Thus, this cylinder arrangement will be investigated in connection with the swashplate drive as discussed in Section 2.1.2.

2.2.7 W and Inverted W Configurations

These configurations were pursued by GM (before 1970). It is felt that these configurations exhibit two major problems:

- a. Triple crankshaft for the W-configuration or double crankshaft (with one overhead) and two crank housings for the inverted W-configuration which constitutes significant weight and cost penalties.
- b. Limited accessibility of components especially in the inverted W where the entire overhead crankshaft has to be removed for exchange of seals or cylinder.

Therefore, it is felt that these configurations are unacceptable for cost or ease of maintenance reasons and will not be further evaluated.

2.3 Cylinder Design

Except for the rhombic drive (opposed cylinder configuration) where single acting cylinders are used, double acting cylinders in a Rinia-type configuration will be used for all other configurations in order to maximize engine output per weight and volume.

2.4 Module Arrangement

One of the main considerations of module arrangement is the ease of integration to the generator and heat source. For a coal fired version, the module arrangement influences significantly the design of the heat transport system. Since interchangeability of parts (with a minimum of different parts)

PIR-SSE-79-001
11/1/79

is a major consideration, module arrangement should be such that it results in as many equal units as possible, and the shortest connection to the heat source to minimize heat losses. When heat pipes are used, a connection as straight as possible should be used since bends increase cost and decrease reliability due to extra welds in the pipe. It also has to be considered that present state-of-the-art does not permit a maintenance/assembly joint in heat pipes thus the entire length of heat pipe and the engine heater head heat exchanger has to be one unit; thus, it has to be readily removable in case of major maintenance or cylinder exchange.

This favors a "in-line" module arrangement as can be seen from the sketch below:

An "in-line" module arrangement will therefore be the prime version contemplated.

In case of swashplate engine, the module arrangement using a connecting gear drive is necessarily circular (either horizontal or vertical) due to the fact that there is only one drive shaft access. The sketch shows the consequences for the heat source interface.

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11/1/79

A possible way to avoid this drawback is using a chain drive which results in a "quasi in-line" configuration which eases the heat source integration for heat pipes or using a pressurized gas heat transfer loop which is easier to integrate to this type of configuration.

2.5 Heat Transport Loop

2.5.1 Liquid Metal Heat Transport Loops

Pumped liquid metal loops are too high a risk for the safety of personnel and hardware in case a leak develops. This is true especially for commercial application where stringent safety standards are applicable.

Additionally, the high liquid metal inventory and exotic plumbing and pumping systems make this system very unattractive. Also the necessity for very specially trained maintenance and service personnel is a significant drawback. Considering also a sizable psychological resistance to exotic systems, will certainly considerably reduce the market acceptability.

A reflux boiler as a heat transport system reduces the safety risk somewhat due to the fact that the pre-sure level in the pipe is always below atmospheric for the temperature ranges in question. However, considering that the heat transfer rate in the combustion chamber or fluidized bed is relatively low, the evaporator section is quite long (with a heat input of 240 KW/cylinder, at least 10 feet). Thus the liquid metal inventory in the evaporator is significant and can constitute a sizable safety hazard. Furthermore, the varying position of the liquid/vapor interface in a reflux boiler is accompanied by extremely high thermal stresses which have resulted in cracks at these interface zones. Therefore the only liquid metal heat transport loop which will be considered with respect to the safety hazard during transport installation and operation will be a heat pipe where almost all liquid metal inventory is at all times retained by the wick system and therefore lowers the safety risk to personnel and hardware. It might be advantageous since it simplifies the heat pipe design in the condenser (the heater head), to use a reflux principle in this section of the heat transport loop. Since free liquid inventory is basically limited to the condensate run-off on the heater head tubes, it is small and therefore can be tolerated from a safety aspect.

2.5.2 Pressurized Gas Heat Transport Loop

A pressurized helium loop as heat transport system will be evaluated in order to assess the viability of such a system from a cost standpoint and with respect to auxiliary power requirements and temperature drop between heat source and heater head. The main advantage of a pressurized gas heat transport loop (PGHTL) is its large flexibility from a heat source integration standpoint and also in view of operational consideration such as emergency shut-downs or maintenance/repair when the gas circuit can be easily shut-down or used through a heat dump HX to cool equipment to be maintained or exchanged without requiring a total cool-down of the heat source.

Connection to a second (redundant) heat source is also possible and might be quite desirable for a larger installation.

The vulnerability of a gas (He) leak which would shut-down the entire engine may require multiple loops for redundancy. The availability of a suitable state-of-the-art gas circulator has to be evaluated.

2.5.3 Direct Fired Heater Heads

Direct firing of the heater head by coal and oil combustion systems is unattractive for two reasons:

- The necessity of very close proximity between heat source and engine which makes maintenance and overhaul of engine and heat source very cumbersome.
- Both oil (especially residual oils) and coal combustion systems produce a significant amount of fly ashes and soot which when deposited on the heat transfer surfaces, would greatly impair the heat transfer characteristics on the heater head tubes.

Furthermore, the inherently close spacing required for the heater tubes makes a routine cleaning difficult and could possibly damage the heater head tubes and heat transfer surfaces (fins).

- The decreased heat transfer coefficient in direct fired heater head heat exchangers results in an increase in the number of tubes and increased void volume.

Therefore, only for a gas fired combustion system, a direct heat transfer between combustion products and heater head heat exchangers will be considered.

2.5.4 Heat Transport by Heat Radiation

The heat transport by radiation would be simple, however it suffers from similar drawbacks as direct heat exchange: the necessity of a close proximity to the heat source and therefore resulting in maintenance and access problems. Also the high heat flux on the heater head heat exchanger requires source temperatures well over 2000°F which only are available in luminous flames (oil) or direct exposure to coal beds since fluidized beds are limited to about 1750°F.

Unless a very specific heat source lends itself for this type of heat transfer, it appears that this type of heat transport is of little general interest in this application and will not be further considered.

3. Engine Configuration Matrix

Based on the discussion in Section 2 which is summarized in Table 2, an engine configuration matrix was selected.

The selected engine configurations are shown in Table 3. These configurations will be subjected to further detailed design analysis and evaluation and if preliminary sketches show promise, detailed layouts will be made.

Table 2

CONFIGURATION PARAMETER SUMMARY

1. DRIVE TRAIN

CONFIGURATION	ADVANTAGES	DISADVANTAGES	COMMENTS
<u>CRANK</u>	<ul style="list-style-type: none"> ● Cost Effective ● High Mechanical Efficiency ● Easy Maintenance ● Well Developed ● Existing Production Facilities ● Trained Skills Available 	<ul style="list-style-type: none"> ● Vibration 	Lowest Cost and Schedule Risk. Good market acceptability.
<u>SWASHPLATE</u>	<ul style="list-style-type: none"> ● Compact ● Low Vibration ● Promising Concept for Advanced Design 	<ul style="list-style-type: none"> ● Not state-of-the-art for large engines ● Limited access for maintenance ● Lower mechanical efficiency ● Auxiliary take-offs difficult ● Module arrangement (no through drive) 	Market acceptability uncertain. Difficult to obtain prior experience on large engines.
<u>RHOMBIC</u>	<ul style="list-style-type: none"> ● Reduced No. of Cylinders ● Low Vibration ● No Cross-heads 	<ul style="list-style-type: none"> ● Mechanical complexity ● Double seal ● Seal maintenance difficult 	Maintainability of seal major problem. Smaller modules may require flywheel.
<u>FREE PISTON</u>	NOT CONSIDERED FOR LARGE ENGINES		Not a viable concept

2. CYLINDER ARRANGEMENT

CONFIGURATION	ADVANTAGES	DISADVANTAGES	COMMENTS
<u>IN-LINE</u>	<ul style="list-style-type: none"> ● Good Accessibility ● Easy Maintenance 	<ul style="list-style-type: none"> ● Excessive Length (block and foundation) ● Low torsional stiffness of crankshaft ● Integration to heat source more difficult (due to length) 	<p>Practical only for engines not exceeding 12 cylinders.</p> <p>Less viable for Rinia configuration.</p>
<u>"V"</u>	<ul style="list-style-type: none"> ● Compact ● Good Accessibility ● Low Specific Weight ● Easy Maintenance 	<ul style="list-style-type: none"> ● Cylinder interconnection somewhat longer 	
<u>OPPOSED</u>	<ul style="list-style-type: none"> ● Same as "V" 	<ul style="list-style-type: none"> ● Long cylinder interconnections ● Integration to heat source more difficult 	Heat transfer loop may obstruct crank case access
<u>RADIAL</u>	<ul style="list-style-type: none"> ● Short and Compact ● Low specific weight ● Excellent modularity ● Well suited for Rinia configuration 	<ul style="list-style-type: none"> ● Integration to heat source more difficult ● Limited access to drive 	<p>For ease of maintenance, vertical axis mandatory.</p> <p>Special foundation support structure required.</p> <p>Market acceptability uncertain</p>
<u>"U"</u>	<ul style="list-style-type: none"> ● Ease of Maintenance ● Multiple Auxiliary take-off ● Short Cylinder interconnections ● Balancing benefits 	<ul style="list-style-type: none"> ● Double crankshaft ● Connecting gear drive in reverse loading ● Bulky block 	Advantages only where multiple heater head integration is desirable
<u>CIRCULAR</u>	<ul style="list-style-type: none"> ● Compactness ● Short Cylinder interconnections 	<ul style="list-style-type: none"> ● Limited access ● Auxiliary drive take-off difficult 	Used on swashplate drive only
-170- <u>"W" or</u> <u>INVERTED "W"</u>	<ul style="list-style-type: none"> ● Close Proximity of Cylinders 	<ul style="list-style-type: none"> ● Double or triple crankshaft ● Bulky ● Limited access for maintenance 	Not considered a viable configuration for maintenance reasons.

3. CYLINDER DESIGN

CONFIGURATION	ADVANTAGES	DISADVANTAGES	COMMENTS
<u>SINGLE ACTING</u>		<ul style="list-style-type: none"> • Greater number of seals • Increased engine volume 	<p>Used for rhombic drive only</p> <p>Requires bounce space.</p>
<u>DOUBLE ACTING</u>	<ul style="list-style-type: none"> • Compact Design • Only one rod seal for two cylinder spaces 		

4. MODULE ARRANGEMENT

CONFIGURATION	ADVANTAGES	DISADVANTAGES	COMMENTS
<u>IN-LINE</u>	<ul style="list-style-type: none"> • Ease of Integration with Heat Source and Generator 	<ul style="list-style-type: none"> • Overall Length 	Either bolt-on or integral block.
<u>DOUBLE ROW</u>	<ul style="list-style-type: none"> • Overall Length Reduced 	<ul style="list-style-type: none"> • More difficult to integrate with heat source • Gear box required for single generator 	
<u>RADIAL/ CIRCULAR</u>	<ul style="list-style-type: none"> • Small Foundation 	<ul style="list-style-type: none"> • Requires special gear box for single generator 	Circular arrangement considered for swashplate engines only.

5. HEAT TRANSPORT SYSTEM

CONFIGURATION	ADVANTAGES	DISADVANTAGES	COMMENTS
<u>HEAT PIPE</u>	<ul style="list-style-type: none"> ● Low ΔT ● Small Liquid Metal Inventory ● No auxiliary power required 	<ul style="list-style-type: none"> ● Slight Safety Problem ● Thermal decoupling from heat source difficult ● Exchangeability difficult ● No field repair possible ● Vapor pressure of Na above atmospheric for $T > 1640^\circ\text{F}$ 	Required size not quite state-of-the-art. Gas controlled pipe may be required for ease of maintenance.
<u>REFLUX BOILER</u>	<ul style="list-style-type: none"> ● Low ΔT ● No auxiliary power 	<ul style="list-style-type: none"> ● Safety problem due to increased liquid metal inventory ● Significant thermal stress in evaporator during transients 	Reflux boiler principle would be used in heater head HX only.
<u>PUMPED LIQUID METAL</u>	NOT CONSIDERED DUE TO SAFETY HARDWARE	PROBLEM WITH PERSONNEL AND	Low market acceptability.
<u>PRESSURIZED HELIUM</u>	<ul style="list-style-type: none"> ● Safe operation ● Good temperature control ● Easy to decouple from heat source or engine switch to redundant heat source. ● Good maintainability ● Relative positioning to heat source less critical 	<ul style="list-style-type: none"> ● Auxiliary power required ● Increased ΔT ● High temp. gas circulator required ● Lower heat transfer rates ● Increased He consumption ● Heavy wall piping 	May require multiple circuits per engine for increased reliability (He leak!)
<u>RADIATION</u>	<ul style="list-style-type: none"> ● Simplicity 	<ul style="list-style-type: none"> ● Requires close proximity to heat source ● Low heat flux ● Requires high source temp. 	Only viable for special applications.
<u>DIRECT</u>	<ul style="list-style-type: none"> ● Simplicity 	<ul style="list-style-type: none"> ● Contamination of heater tubes ● Low heat transfer rates ● Close spacing requirements to heat source 	Considered only for gas fired units.

Table 3. Engine Configuration Matrix

SYSTEM	1	2	3	4	5	6
<u>TYPE OF DRIVE</u>						
Crank	X	X	X			X
Swashplate				X		
Rhombic					X	
Free Piston						
<u>ARRANGEMENT OF CYLINDERS</u>						
In-Line						
V-Type	X	X	X			
Opposed (V-angle 180°)					X	
Radial						X
U (2 rows parallel, 2 drives)				X		
Circular						
W-Configuration						
Inverted W						
<u>CYLINDER DESIGN</u>						
Single Acting					X	
Double Acting	X	X	X	X		X
<u>MODULE ARRANGEMENT</u>						
In-Line	X	X	X		X	X
Double Row						
Radial						
Circular				X		
<u>HEAT TRANSPORT LOOP</u>						
Heat Pipe	X					
Reflux Boiler						
Pumped Liquid Metal Loop						
Pressurized Gas		X		X	X	X
Direct (Gas Fired)			X			
Radiation						

APPENDIX B

GENERAL ELECTRIC

SPACE DIVISION
PHILADELPHIA

PROGRAM INFORMATION REQUEST/RELEASE

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U	SSE	79	002	
*USE "C" FOR CLASSIFIED AND "U" FOR UNCLASSIFIED				

R. Laessig		TO Distribution	
DATE SENT	DATE INFO. REQUIRED	PROJECT AND REQ. NO.	REFERENCE DIR. NO.

SUBJECT
Evaluation Criteria & Method of Economic Analysis for LSSE

INFORMATION REQUESTED/RELEASED

After conceptual layouts of candidate systems had been developed, an evaluation of these concepts was made to identify those which best satisfy the program requirements and goals. The principles and techniques which were employed to evaluate the concepts are discussed in the following sections.

Evaluation Process and Selection Procedure

The evaluation criteria which have been identified for the LSSE are presented in Table 1; they are divided into three categories - performance, costs, and qualitative assessment criteria. The contributing factors which are considered for the qualitative evaluation criteria are listed in Table 2.

1. Concept Evaluation Methodology

The methodology employed for effecting an objective ranking of the various design concepts is outlined below.

1.1. Since engine performance directly enters into the operating costs and thus is reflected in the levelized annual energy cost (see para. 2) performance is not used as a separate rating parameter.

1.2. All systems are ranked separately: a) on levelized annual energy cost and b) on the qualitative assessment criteria as listed in Table 2.

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Overall system ranking will be achieved by comparing individual ratings of levelized annual energy cost and the qualitative assessment criteria and a subjective judgment for the overall most viable system will be made from the design and review teams.

1.3. Ranking on cost: lowest energy cost= highest ranking.

1.4. For the preliminary concept screening (Task 1A-2) calculating levelized annual energy cost for all systems in the screening matrix would have exceeded the scope of this contract. Therefore, the following method is used.

- a) Cost categories are compared by a relative rating to an arbitrarily chosen reference system which is set at 1.0 (e.g. if a system has a development cost estimated 60% higher than the reference system, its relative rating in this cost category (development) is 1.6).
- b) The different cost categories are weighted according to contribution to a levelized annual energy cost, which were determined by a representative sample calculation.
- c) Multiplying the relative rating numbers with the weight factors results in a relative cost factor.
- d) Summing the cost factors for all cost categories results in a relative cost factor which is proportional to the annual levelized cost of energy.
- e) Ranking: lowest cost factor = highest ranking.

The results for the preliminary concept screening are shown in Table 3.

1.5. For the qualitative assessment criteria, the assigned weighting factors are primarily subjective. Several qualified individuals were approached to establish weighting factors based on their assessment:

1. W. Auxer Mgr. Energy Systems Engineering
2. W. Chiu Mgr. Systems Analysis & Evaluation
3. S. Eckard Technical Staff Engineer
4. R. Laessig Technical Consultant/Program Manager
5. Dr. R. Meijer President, Stirling Technology, Inc.
6. W. Richard Mgr. Heat Engine Project Engineering
7. H. Straub Project Engineer

The result is shown in Table 4. Based on these results, the weighting factors selected are shown in the last column of Table 4.

Table 4. Weighing Factors for Qualitative Assessment Criteria

	Individuals							Average	Selected Weight Factor
	1	2	3	4	5	6	7		
Reliability	.25	.31	.3	.31	.22	.25	.19	.27	.3
Maintainability	.2	.24	.2	.18	.13	.20	.22	.2	.2
Application/Interface Flexibility	.1	.12	.05	.10	.04	.15	.15	.11	.1
Fuel Flexiblity	.15	.06	.1	.13	.13	.1	.1	.11	.1
Noise/Vibration	.05	.06	.05	.05	.13	.15	.12	.09	.05
Market Acceptability	.2	.15	.25	.13	.22	.05	.12	.16	.15
Viability for 85 Demo	.05	.06	.05	.1	---	.1	.1	.06	.05
Adaptability for Av. Development					.13				.05

1.6. Rating in these qualitative categories are based on the following method:

<u>Attribute</u>	<u>Rating</u>
Meets Ideal Standard (all goals)	8 to 10
Acceptable (meets requirement but not goals)	4 to 8
Unacceptable (does not meet requirements)	1 to 4

These subjective ratings are multiplied by the weighting factors and summed for all categories in one system. Highest numbers = highest ranking.

2.0 Levelized Annual Cost

The Levelized Annual Cost (sometimes termed annualized cost) is the levelized dollar amount required during each year of the life of the system to:

1. Pay operating and maintenance expenses
2. Pay taxes, including corporation (where applicable)
3. Pay prorated interest on debt
4. Pay prorated return to stockholders
5. Build a capital fund that will allow for periodic component replacement or overhaul and retirement of debt.

Computation of Levelized Annual Costs (LAC) begins with calculation of year-by-year system costs. The present value (at the start year) of all system costs is then computed using after-tax cost of capital (r) as the appropriate discount rate. The present value is analogous to that amount which, if deposited in an account earning interest, r , would permit withdrawals to pay all yearly costs and diminish to zero at the end of system life. There is a related amount which, if deposited annually in the same account would permit the same withdrawals.

This is the Levelized Annual Cost. It is derived from the present value by using a capital recover factor (CRF) of the form:

$$CRF = \frac{r (1 + r)^N}{(1 + r)^N - 1}$$

where r = after-tax discount rate

N = system life

and $LAX = CRF \times \text{PRESENT VALUE}$

The Levelized Annual Cost is analogous to the annual mortgage payment (at interest r) on a loan equal to the present value of future costs.

To calculate the levelized annual energy cost for the candidate system, a computer program is used which was developed for the recently completed Cogeneration Technology Alternative Study (CTAS) Program.

The economic groundrules used in this program are summarized in Table 6.

Further inputs required are:

1. Performance data tabulation of process heat/fuel ratio and power/fuel ratio versus process temperature. A typical input is shown in Figures 1 and 2, and is discussed in detail in Chapter 2.2.
2. Capital cost of the basic combustion subsystem equipment and for the combined equipment cost of the engine, generator, engine/process heat hydronic loop and controls. Listed in Table 7.
3. Cost scaling exponent, X , for use in the equation:

$$\frac{\text{cost}_2}{\text{cost}_1} = \left(\frac{\text{size}_2}{\text{size}_1} \right)^X$$

2.1. Representative Applications

Presently 60 industrial applications are contained in the computer bank of the CTAS computer program. Several representative applications were selected for the LSSE conceptual design which are listed in Table 8.

The commercial cogeneration system used for energy cost performance comparison, is a scaled down version (1:4) of the Fox Valley Center & Village used in the TETAS study. (Reference 1).

2.2. Performance of Stirling System

Engine indicated power and efficiency is calculated employing the Philips thermodynamic computer simulation program. These performance parameters include wall conduction, appendix, and ambient losses. The mechanical losses including piston ring, seal friction, bearing and gearing losses, can be calculated once the thermodynamic gas forces, the inertial forces and the resulting torques over the cycle are known. The slider/wiper seal friction losses, are based upon available test data for the appropriate speeds, strokes and rod diameter combinations. Gearing losses were considered when significant. The bearing friction and pumping losses were estimated based on empirical data.

Some auxiliary equipment is driven directly from the output shaft and therefore will be subtracted from the net shaft output.

Electrically driven auxiliaries and other parasitic electrical losses such as exciterpower etc. are subtracted from the gross generator output. Generator peak and part load performance are obtained according to system size from commercially available hardware data.

System performance then is defined in terms of energy input to the heat transport system, electrical power output available for thermal processes and/or space conditioning, and is presented as shown in Figure 1.

2.3. Output Of Economic Calculation

A typical output is shown in table 9 to 12 containing the following informat

Table 9 (Report 5.1) Fuel energy saved

Table 10 (Report 5.3) Capital costs

Table 11 (Report 5.4 Economic sensitivity report

vary Capital Cost of Cogen ECS -20% to +60% in 20%Δ

vary Cogen Fuel Cost -20% to +60% in 20%Δ

vary Electric Power Cost -20% to +60% in 20%Δ

Table 12 (Report 6.1) Fuel, emissions, and utility savings.

In order to obtain a comparison with other cogeneration/total energy systems the following systems will be computed:

<u>System</u>	<u>Data Source</u>
1. Coal fired AFB Stirling	LSSE
2. Residual Fired Stirling	LSSE
3. Steam Turbine/AFB STM Coal AF	CTAS #1
4. Gas Turbine Residual CTAC12	CTAS #15
5. Diesel Residual DEADV	CTAS #25

Figure 1 Typical Energy Conversion System Characteristic
For Stirling Engine

FIGURE 2

Typical Energy Flow for Cycle Cogeneration Plant

Table 1

PERFORMANCE	COSTS	QUALITATIVE ASSESSMENT CRITERIA
Peak and Part Load Efficiency	Levelized Annual Energy Cost* Development Production Maintenance Operation Installation	Reliability Maintainability Application/Interface Flexibility Fuel Flexibility Noise/Vibration Market Acceptability Viability for 1985 Demonstration Adaptability for Advanced Development

* Only for Final Candidate Systems

Table 2

CRITERIA	CONTRIBUTING FACTORS
1) Reliability	No. of moving parts No. of wear related parts No. of brazed/welded joints - total base area Control complexity Maturity of hardware Factor of safety in design Uniformity of mass flow and heat transfer Experience on selected materials Number size and temperature of seals
2) Maintainability	Component accessibility Component exchangeability (no. of operations, down time) Special tooling required Consumables Estimated routine maintenance intervals No. of spare parts required Exchange of seals
3) Application/Interface Flexibility	Range of power/heat ratio Adaptability to load requirements and load profile Adaptability to heat source

Table 2 (cont'd)

CRITERIA	CONTRIBUTING FACTORS
4) Fuel Flexibility	Capability to burn alternate fuels Degree of difficulty to implement change-over
5) Noise/Vibration	Comparative noise estimate based on RPM Vibration levels based on achievable mass balance
6) Market Acceptability	Ease of installation Modularity Compliance with existing codes Required training/experience level of maintenance personnel Power quality
7) Viability for 1985 Demonstration	Technology risk Schedule risk
8) Adaptability to Advanced Development	Ease of integrating advanced technologies in an existing design

Table 3. Preliminary, Relative Cost Rating of Candidate Configurations

COST	WEIGHT FACTOR	V-ARTICULATED	V-NON-ARTICULATED	SWASHPLATE-JOURNAL BRY.	SWASHPLATE-N.F. BRY.	RADIAL	OPPOSED RHOMBIC	He-TRANSPORT LOOP	HEAT PIPE
DEVELOPMENT	.01	1 .01	1 .01	1.7 .017	1.6 .016	1.1 .011	1.5 .015	1 .01	1.5 .015
PRODUCTION	.24	1 .24	.98 .235	1.2 .288	1.3 .312	9 5 .24	1.3 .312	1 .24	1.2 .288
OPERATION	.61	1 .61	1 .61	1.05 .64	1.05 .64	1 .61	1 .61		
MAINTENANCE	.09	1 .09	.95 .0855	1.4 .126	1.5 .135	1.1 .099	1.6 .144	1 .09	1.0 .09
INSTALLATION	.05	1 .05	1 .05	1.2 .06	1.2 .06	1.2 .06	1.2 .06	1 .05	.8 .04
		1.0	.990	1.131	1.163	1.068	1.141	.39	.433
RANKING		2	1	4	6	3	5		

LEGEND:

RELATIVE RATING	COST FACTOR
--------------------	----------------

TABLE 6

CTAS ECONOMIC ANALYSIS GROUND RULES
(All Costs are in 1978 Constant Dollars)

<u>Factor</u>	<u>Symbol</u>	<u>Value</u>
Annual Inflation Rate	i	0
Cost of Debt (before taxes) above inflation	i_d	3%
Fraction of debt in capital	f_d	30%
Cost of Preferred equity above inflation	i_p	-
Fraction of preferred equity in capital	f_p	0
Cost of common equity above inflation	i_c	7%
Fraction of common equity in capital	f_c	70%
Federal & State Income Tax rate	t	50%
Tax depreciation method	SYD	Sum of Years Digits
Tax depreciation life	n_t	15 Years
Salvage Value	v	0
Investment tax credit	c	10%
Local real estate taxes & insurance	p	3%
Useful life of investment	n	30 Years
First Full Year of Operation	N*	1990
Capital cost escalation rate above inflation	e'_k	0
<u>Cost of Fuels, Power & Expendables for 1985 in 1978 \$'s</u>		
Coal	P_C	$\$1.80/10^6$ Btu
Distillate Oil (Petroleum or Coal Derived)	P_D	$\$3.80/10^6$ Btu
Residual Oil (Petroleum or Coal Derived)	P_R	$\$3.10/10^6$ Btu
Natural Gas	P_G	$\$2.40/10^6$ Btu
Purchased Power	P_{EP}	$\$0.033/kWh$
Limestone	P_L	$\$10.00/Ton$
Dolomite	P_B	$\$12.50/Ton$

TABLE 6
(Cont'd)

ECONOMIC ANALYSIS GROUNDRULES

<u>Factor</u>	<u>Symbol</u>	<u>Value</u>
<u>Escalation of Fuels & Power Above Inflation</u>		
Coal	e'_{p_C}	1%
Distillate Oil (Petroleum or Coal Derived)	e'_{p_D}	1%
Residual Oil (Petroleum or Coal Derived)	e'_{p_R}	1%
Natural Gas	e'_{p_G}	4.6% (1985-2000) 1.0% (2000-)
Purchased & Exported Power	e'_{p_E}	1%
Limestone	e_{p_L}	0
Dolomite	e_{p_B}	0

Price of surplus power exported to utility = .6 x purchased power rate =
.6 x .033 = \$0.0198/kWh

TABLE 7
REQUIRED CAPITAL COST INPUT DATA

Combustion System

1. Combustor & Structure
2. Coal Preparation & Feeder System (or oil supply system)
3. Air Supply System & Preheater HX
4. Exhaust/Stack & Economizer
5. Ignition System, Controls & Safety
6. Piping & Ducting
7. SO₂ - Removal System
8. Fly ASM Separator

Note: Not including coal storage and installation

Heat Transport Loop

- A. Heat Pipe with heat source heat exchange portion (not including engine)
- B. He-heat Transport Loop
 - 1) piping and seals
 - 2) gas circulator

Engine/Generator Set

1. Engine with starter system, cooling pump and lub oil system (excluding heater changes)
2. Generator

TABLE 7 (cont'd)

Auxiliaries/Hydronic Loop

1. Pump (if not included in engine)
2. Expansion Tank
3. Valving
4. Process HX
5. Auxiliary HX--Heat dump
6. Oil Cooler
7. Piping and Fittings
8. Gas Reservoir and Compressor
9. Seal Pumping Systems and Oil Separation

Note: Not including installation

Controls and Instrumentation

1. Control Console and Computer MDW
2. Instrumentation and Wiring
3. Safety and Alarm System

Switch Gear and Electrical Equipment

1. Switch gear
2. Cabling
3. Transformer

Table 8

SELECTED INDUSTRY PROCESSES & SUMMARY OF ENERGY REQUIREMENTS

SELECTED INDUSTRY PROCESSES & SUMMARY OF ENERGY REQUIREMENTS												By-Product or			
SIC Code	Process No.	Description	Process Electric Power		Process Steam MBtu/hr	% Hot Water	Temperature		Power /Heat Ratio	Load Factor hrs/yr	Primary Fuel	Waste Fuel Avail MBtu/hr	National Energy Consumption Utility		
			MW e	MBtu/hr			Of Peak	Of Avg.					+ Site 10 ¹² Btu/yr	1978	1985
20 FOOD AND KINDRED PRODUCTS															
2011	1	Meat-Packing	1.940	6.625	24	40	250	250	0.28	2100	Gas		71	96	168
2026	1	Fluid Milk	1.310	4.474	11	50	250	250	0.41	2100	Gas		71	80	101
2046	1	Wet Corn Milling	28.500	97.327	659		250	250	0.15	6600	Gas		104	141	159
2063	1	Beet Sugar Refining	4.700	16.050	301		250	250	0.05	2800	Gas	76.47	100	118	162
2082	1	Malt Beverage	6.040	20.627	86	60	250	250	0.24	6600	Gas		75	120	190
22 TEXTILE MILL PRODUCTS															
2260	1	Textile Finishing	6.200	21.173	158		341	331	0.13	6240	Coal		75	75	75
24 LUMBER AND WOOD PRODUCTS															
2421	1	Soft Wood-Lumber Sawmill	1.500	5.123	30		353	353	0.17	4000	Dark-Sawdust	41.2	237	300	400
2436	1	Soft Wood-Plywood/Veneer	3.000	10.245	75		406	406	0.14	6000	Bark	100.0	100	150	275
2442	1	Particle Board	5.000	17.075	37		406	406	0.46	8000	Natural Gas	41.2	32	100	172
26 PAPER & ALLIED PRODUCTS															
2621	2	Bleached Kraft	50.000	170.750	780		366	340	0.22	8400	Coal	353	416	454	784
2621	4	Unbleached Kraft	29.000	99.035	610		366	328	0.16	8400	Coal	259	405	441	950
2621	6	Neutral Sulfide Semichemical	20.000	68.300	307		366	345	0.22	8400	Coal		63	69	128
2621	7	Thermo-Mechanical Pulping	31.300	106.889	227		366	355	0.47	8400	Coal		102	110	205
2621	8	Waste Paper	15.000	51.225	224		366	355	0.21	8400	Coal		176	191	419
28 CHEMICAL & ALLIED PRODUCTS															
2800	1	Small Integrated Power Plant	32.500		1100			366	0.101	8760					
2800	2	Medium Integrated Power Plant	77.200		1054			366	0.25	8760					
2800	3	Large Integrated Power Plant	97.200		947			366	0.35	9760					
2812	1	Chlorine - Caustic Soda	120.000	409.800	265		338	311	1.55	3500	Any		180	240	300
2813	1	Cryogenic Oxygen	34.000	116.110	0		0	0	999.99	3400	Electric		22	33	66
2819	1	Alumina	30.290	103.440	980		495	434	0.11	8136	Coal-Oil		53	76	135
2821	2	Vinyl Chloride	4.000	13.660	207		422	373	0.07	8300	Gas		50	110	160
2821	3	Low Density Polyethylene Resin	55.000	187.825	16		448	448	11.74	7900	Any		18	38	60

Table B cont'd
SELECTED INDUSTRY PROCESSES & SUMMARY OF ENERGY REQUIREMENTS

SELECTED INDUSTRIAL PROCESSES & SUMMARY OF ENERGY REQUIREMENTS																
SIC Code	Process No.	Description	Process Electric Power		Process Steam MBtu/hr	% Hot Water	Temperature		Power /Heat Ratio	Load Factor hrs/yr	Primary Fuel	By-Product or Waste Fuel Avail MBtu/hr	National Energy Consumption Utility			
			MW _e	MBtu/hr			°F Peak	°F Avg.					+ Site	10 ¹² 1978	1985	Btu/yr 2000
28	CHEMICAL & ALLIED PRODUCTS (Cont'd)															
2822	1	Styrene-Butadiene Rubber	7.500	25.612	35		338	338	0.73	7900	Any		7	9	13	
2824	1	Polyester Fibre	32.000	109.280	30		406	406	3.64	7900	Gas-Oil		30	55	75	
2824	2	Nylon Fibre	11.000	37.565	23		274	274	1.63	8760	Any		14	20	25	
2865	2	Cumene-Benzene	0.600	2.049	0		0	0	999.99	8400	Gas-Oil		6.5	10	15	
2865	3	Phenol/Acetone	6.000	20.490	300		489	398	0.07	8200	Any		20	45	60	
2865	4	Ethylbenzene	0.700	2.390	220		489	489	0.01	7900	Oil-Gas		22	45	65	
2869	1	Methanol Synthesis	1.500	5.123	133		574	538	0.04	7880	Feedstock	352.9	0	0	0	
2869	4	Ethanol	3.300	11.270	400		460	460	0.03	7900	Gas-Oil	70.6	18	24	30	
2873	1	Ammonia Synthesis	3.500	11.952	640		598	598	0.02	8400	Gas-Oil		200	250	305	
2874	1	Phosphoric Acid	4.000	13.660	92		353	292	0.15	7900	Gas-Oil		35	48	60	
2895	1	Carbon Black	4.000	13.660	20		298	298	0.68	7900	Oil-Gas		18	20	24	
29	PETROLEUM REFINING															
2911	1	Small Refinery	14.000	47.810	375		470	389	0.13	8760	Oil-Der		560	580	630	
2911	2	Medium Refinery	52.000	177.500	1333		470	395	0.13	8760	Oil-Der		850	870	950	
2911	3	Large Refinery	126.000	430.290	3042		470	385	0.14	8760	Oil		1220	1250	1280	
32	STONE, CLAY AND GLASS															
3211	1	Flat-Glass	5.600	19.124	0		0	0	999.99	7500	Nat-Gas					
3221	1	Glass Containers	5.100	17.416	0		0	0	999.99	7500	Nat-Gas					
3229	1	Press-Blown Glass	1.100	3.756	0		0	0	999.99	7500	Nat-Gas					
3241	1	Cement	20.316	69.379	0		0	0	999.99	7920	Coal					
33	PRIMARY METALS															
3312	1	Specialty Steel	60.000	204.900	93		448	446	2.20	6700	Nat-Gas		560	643	835	
3325	1	Integrated Steel	280.000	956.200	912		448	445	1.05	8400	Cok-Coal	529.4	3080	3539	4596	
3325	4	Mini-Steel	40.000	136.600	91		448	446	1.50	6700	Nat-Gas		360	612	1070	
3331	1	Copper-Fire Smelted	24.800	84.692	0		0	0	999.99	8400	Oil		4.6	5.8	9.3	
3331	4	Copper-Anode Smelted	10.100	34.491	40		364	364	0.86	7620	Oil		12.2	15.5	24.8	
3334	1	Aluminum	756.000	2581.740	0		0	0	999.99	8760	Oil		31.8	49.2	86.4	

TABLE 9

ATE 06/06/79

GENERAL ELECTRIC COMPANY
COGENERATION TECHNOLOGY ALTERNATIVES STUDY
REPORT 9.1

PAGE 1

FACILITY ADV DESIGN ENG

FUEL ENERGY SAVED BY PROCESS AND ECS

INDUSTRY 10101 HV 10.00 PROCESS MILLIONS BTU/HR 137.0 PROCESS TEMP(F) 300. PRODUCT NASA-CLDR#1 HOURS PER YEAR 8000.

UTILITY FUEL COAL

POWER TO HEAT RATIO 0.249

WASTE FUEL EQV BTU=10**6= 110. HOT WATER BTU=10**6= 0.

WASTE FUEL USED 10**6 BTU/HR	FUEL SAVED= NO-NET 10**6 BTU/HR	COGEN FUEL USED 10**6 BTU/HR	COGEN PROCESS HEAT 10**6 BTU/HR	COGEN PROCESS POWER 10**6 BTU/HR	COGEN HV ELECT 10**6 BTU/HR	AUX PROCESS BOILER 10**6 BTU/HR	UTILIT FUEL USED 10**6 BTU/HR	TOTAL FUEL SITE 10**6 BTU/HR	SITE FUEL USED 10**6 BTU/HR	NET= TOTAL+ UTILIT 10**6 BTU/HR	FAIL	FESR	POWER HEAT FACTOR	HEAT FACTOR
0 DMICOH N O C O O O H	110.	0.	0.	0.	0.	161.	107.	161.	COAL-FED	260.	0	0.	0.13	0.61
1 ST1141 STM-TURB-1 POWER	110.	63.	203.	130.	34.	10.	-2.	203.	RESIDUAL	203.	0	0.43	0.17	0.60
1 ST1141 STM-TURB-1 HEAT	110.	66.	201.	137.	34.	10.	0.	201.	RESIDUAL	202.	0	0.44	0.17	0.60

Does not apply

Does not apply

Fuel energy saved ratio = $\frac{Q_{\text{Cogen}} - Q_{\text{No Cogen}}}{Q_{\text{No Cogen}}}$

Does not apply

Total Q = Q_{onsite} + Q_{utility}Total Fuel on site (Q_{onsite})Utility equivalent (Q_T - 32% incl. distrib.)

Electr. Power MW

Electr. Power (Btu/Hr)

On site fuel used

No cogen - Q_{cogen} (No cogen = Q_{onsite} + Q_{utility})

Power = power match, Heat = heatmatch

TABLE 10

DATE 03/31/79
1 GE-PEO ADV. DES. ENSRG.

GENERAL ELECTRIC COMPANY
COGENERATION TECHNOLOGY ALTERNATIVES STUDY
REPORT 5.3
CAPITAL COSTS BY ISLAND FOR SELECTED PROCESS-ECS MATCHES

PAGE 10

PROCESS 20021

ECS STIRL PROCESS MEGAWATTS 6.04 PROCESS TEMP. 250. PROCESS HEAT(BTU*10**6) 86.
STIRLING-1472F SITE FUEL= RESIDUAL COGEN FUEL BTU*10**6= 65. KW FUEL= 19161.

		*****COSTS - MILLIONS 1978\$*****						
ISLAND DESCRIPTION	COMPONENT DESCRIPTION	MAJOR EQUIPMENT	INSTALL MAT'L	INSTALL LABOR	INDRCT FLD CST	TOTAL INSTALLED	TOTAL	\$PER-KW FUEL
1. FUEL-HANDLING	1. FUEL-OIL-UNLOADING-S	0.070	0.014	0.084	0.076	0.174	0.244	12.742
	ISLAND TOTAL	0.070	0.014	0.084	0.076	0.174	0.244	12.742
2. FUEL-UTILIZATION-CLE	29. STIRLING-ENGINE-COMB	0.077	0.009	0.009	0.008	0.027	0.104	5.405
	29. STIRLING-ENGINE-I	1.404	0.178	0.178	0.160	0.516	2.000	104.377
B →	21. OIL-FIRED-BOILER	0.190	0.394	0.601	0.541	1.636	1.726	90.086
	ISLAND TOTAL	1.751	0.581	0.788	0.710	2.079	3.830	199.869
8. BALANCE-OF-PLANT	84. POWER-PLANT-STRUCTUR	0.	0.179	0.156	0.141	0.475	0.475	24.008
	80. MASTER-CONTROL	0.143	0.022	0.036	0.032	0.090	0.233	12.167
	01. ELECTRIC-SWITCHGEAR-	0.	0.036	0.036	0.032	0.103	0.103	5.391
	02. INTERCONNECTING-PIPI	0.	0.068	0.068	0.062	0.198	0.198	10.350
	03. STRUCTURES-MISCELLAN	0.	0.150	0.140	0.126	0.425	0.425	22.165
	ISLAND TOTAL	0.143	0.463	0.436	0.393	1.291	1.435	74.030
TOTAL THIS CASE		1.064	1.050	1.300	1.170	3.544	5.509	61.470
INDIRECT COSTS	SPARES 2% of major equipment						0.039	
	START UP 1% of major equipment and installation mat'l. and labor						0.043	
	SPARES+STARTUP						0.083	
	CONTINGENCY 15% of total cost						0.039	
	ENGINEERING SERVICES 6% of total cost						0.335	
	A-E FEE 5% of total cost						0.280	
GRAND TOTAL							7.045	

Does

NOT APPLY

TABLE 11

06/07/79
ELO-ADM ENERGY-SYSGENERAL ELECTRIC COMPANY
COGENERATION TECHNOLOGY ALTERNATIVES STUDY
REPORT 5.4

PAGE 1

ECONOMIC SENSITIVITY REPORT FOR SELECTED PROCESS-ECS MATCHES

SENSITIVITY OF CAPITAL COST

PERCENT OF ORIGINAL COST 100

*****LEVELIZED ANNUAL ENERGY COSTS(\$ MILLIONS)*****

PGY CONV	SITE- FUEL	POWER REQ'D MW	POWER GEN/ REQ'D	FESPOWER /HEAT COST RATIO *10**6	CAPITAL COST *10**6	TAXES + INSNC	CANDM	FUEL	PURCHD ELEC	REVENUE	TOTAL	NORML	PRESENT WORTH 15%	ROI %	GROSS PAY BACK
101 DMCCO COAL-FG	10.	0.	0.	0.25	12.3	0.93	0.40	0.83	0.74	3.00	0.	5.97	1.000	0.	0
101 ST1141 RESIDUA	10.	0.99	0.439	0.25	9.3	0.63	0.27	0.57	2.42	0.03	0.	3.93	0.658	8.	999
101 ST1141 COAL-FG	10.	0.99	0.439	0.25	16.2	1.23	0.52	1.00	1.41	0.03	0.	4.27	0.715	3.	20
101 ST1141 COAL-AF	10.	0.99	0.439	0.25	12.5	0.95	0.40	0.96	1.41	0.03	0.	3.74	0.626	7.	999
101 ST1000 RESIDUA	10.	0.75	0.333	0.25	7.4	0.56	0.24	0.54	2.15	0.76	0.	4.26	0.713	8.	999

Not applicable

Return on investment (ROI)

Not applicable

$$\frac{\text{Cogen Levelled Cost}}{\text{No Cogen Levelized Cost}}$$
Total levelized annual energy cost 10⁶\$

Revenue of exported power

Cost of purchased power

Fuel cost

O & M

Taxes and insurance = 3% of capital cost

Yearly cost of capital = interest and derecation

Total capital cost 10⁶\$

Power to heat ratio of process

$$\frac{Q_{\text{cogen}} - Q_{\text{cogen}}}{Q_{\text{No cogen}}}$$

$$\frac{P_{\text{EI cogen}}}{P_{\text{EI process req'd.}}}$$

Process elect. power req'd.

TABLE 12

DATE 08/12/79
SE PEO AES

GENERAL ELECTRIC COMPANY
COGENERATION TECHNOLOGY ALTERNATIVES STUDY
REPORT 6.1 FUEL AND EMISSIONS SAVINGS
TIME 1990 LEVEL ALL

PAGE 1

FUEL UNITS		REPORT 6.1		FUEL AND EMISSIONS SAVINGS		(SAVINGS ARE POSITIVE)								
EMISSION UNITS		TIME 1990		LEVEL ALL										
COST		*\$=10**9				TYPE MATCH=POWER								
*****FUEL SAVINGS*****		-----EMISSIONS SAVINGS-----		CAPITL--ELECTRIC POWER---										
ROCS	EC9	ECS	*****DIRECT*****	TOTAL	FESR	-----DIRECT-----	*****TOTAL*****	EMSR	SAVING	TOTAL	COST	LAEC		
		FUEL	OIL+GAS	COAL	OIL+GAS	COAL	NOX	SOX	PART	NOX	SOX	PART	EXPORT	SAVED
													MWH	
0111	STM141	RESIDU	0.	-0.002	0.	0.003	0.26	-1.	-1.	-0.	1.	2.	0. 0.28	-0.
0111	STM141	RESIDU	0.	-0.002	0.	0.003	0.28	-1.	-1.	-0.	1.	2.	0. 0.29	0.
0111	STM141	COAL-F	0.	-0.002	0.	0.003	0.26	-1.	-2.	-0.	1.	0.	0. 0.18	-1.
0111	STM141	COAL-F	0.	-0.002	0.	0.003	0.28	-1.	-2.	-0.	1.	1.	0. 0.20	-0.

Levelized energy cost savings ($E_{\text{Cogen}} - E_{\text{Noncogen}}$)Net Cost of electric energy $\frac{\text{MILS}}{\text{KWL}}$

mpbs/yr exported

Capital cost saved including utility

Emissions savings ratio $\frac{E_{\text{Cogen}} - E_{\text{Noncogen}}}{E_{\text{Noncogen}}}$

Particulate emission

Sox emission

Nox emission

Savings including utilities 10^{-3} tons/hr.

Particulate emission

Sox emission

Nox emission

Savings on site 10^{-3} tons/hr.

$$\frac{Q_{\text{noncogen}} - Q_{\text{cogen}}}{Q_{\text{noncogen}}}$$
Include Q_{utility} Include Q_{utility} Does not include Q_{utility}

APPENDIX C

GENERAL ELECTRIC

SPACE DIVISION
PHILADELPHIA

PROGRAM INFORMATION REQUEST / RELEASE

CLASS. LTR.	OPERATION	PROGRAM	SEQUENCE NO.	REV. L.
U	SSE	79	003	
PIR NO. — SSE — 79 — 003				
*USE "C" FOR CLASSIFIED AND "U" FOR UNCLASSIFIED				

H. Sayell		TO Distribution	
DATE SENT 11/21/79	DATE INFO. REQUIRED	PROJECT AND REQ. NO.	REFERENCE DIR. NO.
SUBJECT Materials for Large Stationary Stirling Engine (LSSE)			

INFORMATION REQUESTED/RELEASED

This PIR discusses materials considered for the construction of an LSSE and related hardware. Candidate materials for fabrication of various engine components are listed in Table I; selections are made on the basis of information presented below. Specific material assignments await design finalization and sorting of candidate material systems by means of proof testing in representative environments.

A. GENERAL DISCUSSION (PRIORITIES USED FOR MATERIAL SELECTION)

High priority considerations narrowing the field of candidate materials are base material cost, estimated fabrication costs, material availability, materials compatibility and joinability, and thermal stability of the alloy in its intended environment. The prototype LSSE engine will be constructed out of the most available, low cost, joinable, and thermally stable materials that reliably meet design and environmental needs.

Cost/Availability/Fabricability

The attractiveness of an alloy diminishes as its elemental complexity increases. Alloys containing higher percentages of stratigic (government stock-pile) materials like Cr, Mo, Co, W, etc., are considered less desirable than those that meet environmental needs but are comprised of more available elements.

Compatibility

Material compatibility with adjoining (and close proximity) components is considered as important an acceptance criteria as is individual stability in a given hostile environment. Materials behaving acceptably in adverse environments are rejected for service recommendation if their union is mutually degrading, or destructive to either member of the pair. Considerations such as thermal expansion mismatch, galvanic cell combination, alloy dilution (or enrichment) by possible elemental diffusion, contamination by solutioning, evaporation, etc. guide the selection of alloys used in any given assembly.

W. Auxer	W. Richards	PAGE NO. 1 of 37	RETENTION REQUIREMENTS	
W. Chiu	H. Sayell (3)		COPIES FOR	MASTERS FOR
R. Emmer	H. Straub		☐ 1 MO.	☐ 3 MOS.
R. Laessig (2)			☐ 3 MOS.	☐ 6 MOS.
			☐ 6 MOS.	☐ 12 MOS.
			☐ MOS.	☐ MOS.
			☐	☐ DO NOT DESTROY

Thermal Stability

Alloy preference is given to material systems that do not change crystal structure or thermally age when operated in the 800°F to 1500°F temperature range, or during thermal cycling between ambient and elevated temperatures. In general, ferritic and martensitic stainless steels do not behave as well as austenitic stainless steels in the temperature ranges cited above. Low carbon and stabilized grades of austenitic stainless steels are preferred over those having carbon contents greater than 0.03% without addition of Cb and/or Ti. Heat treatable (age-hardenable) stainless steels and superalloys are discounted for greater than 8000 Hr. service in moderate temperature (1200° to 1500°F) environments containing sulfur, O₂, H₂, CO₂, Na, methane, water-vapor, etc. because overaging (softening and accumulation of carbides and/or brittle intermetallic compounds in grain boundaries) accelerates corrosion and oxidation.

Joinability

Alloys that can be welded, and/or brazed, without need of post-weld heat treatment are preferred. Alloys containing greater than 0.5% Ti and/or Al are not attractive where brazing is required because Ni plating and/or special brazing atmospheres are necessary to effect the union.

B. DESIGN CONSIDERATIONS

The Stirling engine is attractive because available, low-cost, high sulfur bearing coal can be used to supply the energy required to heat the gas powering the piston. When this type fuel is burned in excess air, in the proximity of dolomite (fluidized bed converter), sulfur, is converted into relatively non-hostile compounds. If a thermally efficient Na bearing heat-pipe is introduced into the system, the design becomes even more attractive. The selection of low cost materials that can perform reliably in these environments is a challenge that must be met. In effect, there are 3 different general environments one must consider when assigning specific materials for construction of such a system:

- (1) Coal Converter; high temperature, low pressure, combustion product (gas and solids) external, and solid and liquid sodium internal, environment.
- (2) Cylinder (Hot-side); high temperature, high pressure system having external air and internal helium environments.
- (3) Cold Side; components operating at less than 800°F in air, ethylene glycol solutions, or helium.

Of the three general environments listed above, the atmospheres provided by the coal conversion part of the system is most challenging to the long term stability and life of the engine.

C. COAL CONVERTER

The fluidized bed coal converter will probably be a low carbon steel, refractory oxide lined vessel that houses critical metal components used to convey liquid sodium to the engine heat exchanger (condenser). These metallic parts, and temperature monitoring instrumentation, will be subjected to low pressures (<10 psi) but high temperature ($\sim 1500^\circ\text{F}$) operational and 1700°F max. short term excursions. The major constituents external to the metallic components will be hydrogen, water-vapor, carbon monoxide, carbon dioxide, methane, sulfurous compounds, air (O_2 and N) and hot char deposits (ash constituents). Other possible contributors to the external deterioration of the metal components are possible liquid metals (eutectic formations between metals and sulfides), erosion by particulates in the gas stream, and possible hot-spots created by char build-up on metal wall surfaces. The internal surfaces of the tubing used to convey the heat transfer media must be compatible with solid and liquid sodium. Some information concerning the compatibility of metals in these environments follows.

Oxidation, (Oxygen, water-vapor, CO_2)

Carbon and low alloy steels do not have sufficient chromium to make them serviceable in $>800^\circ\text{F}$ oxidizing environments. Most stainless steels have adequate oxidation resistance to 1500°F , Figure 1. The most oxidation resistant of the stainless steels are 310, 330, and 446 which survive in dry air to $\sim 1900^\circ\text{F}$ (intermittent service). If the air is moist, oxidation resistance is reduced significantly. The rate of oxidation by CO_2 is dependent upon its partial pressure; greatest attack between 20 and 100 torr.

Incolloys are more oxidation resistant than the best stainless steels and the Inconels are superior to the Incolloys, Figure 2. It is important to note (Table II) that the cost and delivery time increases significantly as one moves away from stainless steels in the direction of the Inconels.

Gaseous Environments

Carburization resistance of the stainless steels, Incolloys, and Inconels follow the same general pattern discussed for oxidation resistance above, Figure 3.

The data on Table III indicates that sulfurous environments degrade the stainless steels more severely than do oxidizing or carburizing environments. H_2S is probably the most severe of all sulfur environments; natural gas produces significantly less damage to stainless steels than does coke oven gases. Of the stainless steels, 310 and 314 appear to be most resistant to attack by most sulfur bearing atmospheres.

Data from coupons exposed to an operating fluidized bed atmosphere (low sulfur content) for 782 hrs. at 800° to 1550°F (Tables IV & V) indicate that most high Cr iron-base alloys performed better than Ni base alloys and that coupons placed in vapor zones suffer greater degradation than those placed within the fluidized bed.

In addition, plant gassifier equipment fabricated out of Incoloy 800 has experienced no failures except in welds to stainless steel. Apparently Incoloy 800 forms carbides as a result of welding and stress corrosion cracking follows. A solution used is replacement of Incoloy 800 with Incoloy 825 in locations where welding is required.

IIT performs experiments in simulated coal gasification atmospheres and follows the behavior of coupons placed in some of the different operating fluidized bed systems operating around the country. Some of the alloy systems being followed are listed in Table VI. Results of tests performed in simulated coal gasification atmospheres are shown in Table VII. A comparison of results obtained in simulated gasification environments and those in operation plant environments are shown in Table VIII. Some of the conclusions drawn from coupons run at ~1500°F are:

- (1) Fe base alloys perform better than Ni base alloys.
- (2) Fe base alloys require $\geq 20\%$ Cr to perform well in gasifier environments; Ni base alloys need $\sim 50\%$ Cr.
- (3) Aluminized coatings help protect stainless steels and Incolays in a simulated environment but fail to protect in actual fluidized bed atmospheres.
- (4) Plant environments attack specimens more aggressively than simulated environments.
- (5) H_2S gas concentrations ranging between 0.1 and 1.0 vol % do not significantly affect corrosion rate of high Cr bearing Fe base alloys.

Some of the conclusions drawn as a result of studying the 1800°F coupon experiments are important to this program since the LSSE fluidized bed may rise to 1700°-1750°F (short term):

- (1) Corrosion rates of both Fe and Ni base alloys change in relation to the volume percent H_2S present, Figures 4 through 6. Cobalt base alloys exhibit excellent corrosion resistance over the zero to 1.0 vol. % H_2S range (1000 hrs.).
- (2) Ni base and high Ni alloys have higher corrosion rates when H_2S Vol. % is $\geq 0.5\%$.
- (3) High Cr iron base alloys demonstrate higher corrosion rates when H_2S Vol. % is $\geq 0.5\%$.
- (4) Corrosion rates are significantly higher at 1800°F than at 1500°F.
- (5) Short term (1000 hrs.) data should not be extrapolated especially if an alloy's Cr content is $< 25\%$.

- (6) Cobalt base alloys should be used for parts required to function long term at 1800°F; Ni and Fe base alloys are probably not acceptable.

Data tabulated in Table IX indicates that Ni base (NOT Fe base) alloys exhibit molten corrosion products as a result of exposure to gasifier atmospheres containing 0.5 to 1.0 Vol. % H₂S.

Hydrogen and Sodium Environments

Age hardenable alloys appear to be more vulnerable to degradation by hydrogen than solid solution strengthened alloys; vulnerability increases as alloy strength increases. Thermal cycling may accelerate damage to a material system since hydrogen absorption is reversible in some elements contained in most age-hardenable alloys. Thermal gradients within a given metal structure will tend to concentrate hydrogen in the cooler parts of the system. Additive gases, CO, CO₂, Air, O₂, etc. (of the type that are present in the external atmosphere of a fluidized bed gasifier) dilute the hydrogen and reduce its activity. The hydrogen that may be generated by reactions of liquid sodium with metal tubing, may be gettered with Zr and/or Ti, etc. if determined necessary. The literature, however, indicates that most high Cr stainless steels and Ni based alloys are compatible with hydrogen and sodium at 1500°F, see Figures 7 through 10. As is the case with hydrogen, the higher Cr grades of solution strengthened Fe and Ni base alloys tend to be most compatible with Na; the more thermally stable the alloy the less the interaction. The information offered in Figures 11 through 13 show little difference in hydrogen permeability, diffusivity, and solubility (3000 psi; 1200° to 1500°F) in A286, 19-9DL, N155 and Incoloy 800. The curves in Figure 14 indicate that the cobalt base alloy 6B shows lower permeability and less temperature effect than iron-base alloys. IIT (NASA sponsored) has shown that pack aluminide coatings increase resistance to hydrogen permeation but Si₃N₄, glass, and Al₂O₃ do not. Stress rupture data presented in Table X shows that hydrogen does not significantly effect the life Inconel 700 and S-816 at 1500°F.

D. CYLINDER (HOT-SIDE)

As previously discussed, the materials required for containment of helium, on the hot side of the LSSE, are highly stressed (~6000 psi) at temperature in the 1300° to 1500°F temperature range; resistance to long term stress and oxidation are the main concerns in this part of the overall design.

Strength

The data in Tables XI and XII show that the most creep resistant of the austenitic stainless steels at elevated temperatures is 316. None of the austenitic stainless steels have adequate long term (>10000 Hr.) strength at temperatures greater than 1250°F. Table XIII indicates that there are many available Fe, Ni, and Co base alloys that have good creep resistance at 1400°F but many of these are discounted for LSSE on the basis of high cost, poor joinability, and thermal instability. The more joinable and thermally stable alloys having elevated temperature strength advantage over stainless steels are shown in Figures 15 and 16 and Tables XIV through XVI. The higher carbon in Incoloy 800H, over that in Incoloy 800

(Table II), makes this alloy considerably more creep resistant at elevated temperature; max. temperature for long term use is around 1400°F. Of the alloys considered, Inconel 617 and Haynes 188 are the most creep resistant and have highest fatigue strength at 1500°F, and higher.

Incolloys 800, 800H, and 802 as well as Inconel 625 and Haynes 188, all have elevated temperature ductility troughs between 1200° and 1400°F. Inconel 617 has continuously increasing ductility to 1600°F, see Figure 17. Except for Incoloy 802, all the alloy ductilities between 1200° and 1400°F are 30% or higher (unaged) and fatigue life of all alloys, at 1300°F to 1400°F, is excellent. However, all these alloys (except Inconel 617) sensitize (form CrC and/or brittle phases) with time at 1200° to 1500°F. The elevated ductility trough shifts to lower temperature and deepens (decreased ductility) as shown in Figure 18. These data suggest that the low temperature fatigue life of aged Incoloy 800, 800H, and 802, Inconel 625, and Haynes 188 can be significantly less than that shown in literature for optimized base metal structure. High thermal and/or mechanical stress of aged material may be more critical at room to 600°F, for example, than at 1200 to 1500°F. Since these same materials respond about the same way to welding (structure in heat affected metal is somewhat relatable to that produced during aging) it is not uncommon to find cracking in stressed welds (especially in a hostile environment).

Oxidation and Corrosion Resistance

As has already been discussed, all of the candidate alloys shown in Table II, have adequate oxidation resistance to 1500°F. Of the alloys, Haynes 188 probably can be expected to perform the best in all environments discussed; 316(L) stainless not as resistant to most environments as the others.

Thermal Expansion and Conductivity

Thermal expansion and thermal conductivity of candidate alloys are listed in Table XVII. The austenitic stainless steels have the highest thermal expansion and conductivity of alloys considered. As one expects, joining of stainless steels to Incolloys produce more stress free joints than those made between stainless steels and Inconels. However, successful joints are commonly made between stainless steels and Inconels when joints are designed properly and good welding techniques are utilized.

E. COLD SIDE COMPONENTS (<800°F)

There are many low alloy steel products (1/2 to 1% Cr and 1/2 to 3% Mo) that can be used adequately for the construction of cold side components. In many instances, low carbon steels may be found acceptable where painting is not prohibitive. No discussion of these materials is considered necessary.

RECOMMENDATIONS

Coal Gas and Fluidized Bed Environment

On the basis of cost, availability, and overall performance, 310(S) stainless

steel shapes and seamless tubing are recommended for service in a 1400° to 1500°F range coal gas/fluidized bed environment. Incoloy 800 and Incoloy 825 are considered acceptable back-up materials (Incoloy 825 is a better choice than Incoloy 800 where welding is required). Both Incoloy alloys require highly specialized handling (Ni plating and high vacuum for joining by brazing); 310(S) will sensitize as readily as Incoloy 800.

The best all around material is probably Haynes 188 but high cost and low probability of obtaining tubular shapes causes one to discount the alloy.

Selection of any one of the three alloys for coal bed service is done at some risk since performance is highly dependent upon the type atmosphere actually developed in the gasifier part of the prototype LSSE. Testing in simulated coal gas environments and accelerating testing by increasing temperature are not recommended. Coatings and progress of Inconel 671 (50 Ni/50 Cr) should be followed by reference to other programs.

Hot Side/Single Wall Cylinder and Heat Exchanger Tubing

Where good high temperature creep resistance is required (1400°/1500°F range), Inconel 625 is recommended. If cost is prohibitive, Incoloy 802 may be considered as a substitute. Both Inconel 617 and Haynes 188 are higher strength materials that should perform better than either Inconel 625, or Incoloy 802, but cost and delivery causes one to look more favorably on the first two mentioned alloys.

Hot Side/Dual Wall Cylinder (H₂O Cooled) & Engine Piston

Where creep resistance is not a critical factor and temperature does not exceed 1300°F 316(L) stainless steel is the most attractive construction material.

Fins

Where fin temperature can reach 1700° to 1800°F (intermittent operation), 310(S) stainless steel and 601 Inconel are suggested. Inconel 601 is more oxidation resistant than 310 stainless but 601 requires plating and hard vacuum atmosphere to form acceptable brazed joints. Inconel 625 can be brazed in relatively poor vacuum (10⁻⁴ torr) but is more expensive and less oxidation resistant than Inconel 601.

TABLE I. COMPARISON OF CANDIDATE MATERIALS (1)

ATTRIBUTE	STAINLESS STEEL		INCOLOYS				INCONELS			HAYNES
	316(L)	310(S)	800	800H	802	825	601	625	617	182
BASE MAT'L COST (X=\$1.50/lb.)	X	1.2X	2X	2X	2.3X	2.5X	3.1X	5.6X	6.3X	8X
MILL DELIVERY (MONTHS)	3	4	6	6	8	10	10	17	10	10
AVAILABLE FORMS	ALL	ALL	ALL	ALL	FEW	FEW	ALL	ALL	FEW	FEW
THERMAL STABILITY (2)	2	4	4	5	5	3	5	4	1	2
RESISTANCE TO COAL GAS (2)	3	2	2	3	4	3	7	6	5	1
RESISTANCE TO H ₂ Permeation (2)	3	5	3	5	5	4	3	4	2	1
RESISTANCE TO Na (2)	2	3	2	3	4	3	3	3	2	1
RESISTANCE TO 0.1 VOL.% H ₂ S (2)	3	2	3	4	4	3	5	7	6	1
RESISTANCE TO 1.0 VOL.% H ₂ S (2)	5	2	2	3	4	3	6	8	7	1
RESISTANCE TO OXIDATION (2)	6	3	4	4	5	5	1	2	2	1
EASE OF WELDING (2)	1	6	6	6	5	4	3	2	4	4
EASE OF BRAZING (2)	1	2	5	5	3	5	6	2	5	4
10,000 HR.@1500°F RUPTURE STR. (3) x 10 ⁻³	3.05	2.8	4.0	5.2	8.2	-	4.0	9.5	12.0	10.0
RESISTANCE TO RUPTURE (1500°F) (3)	6	7	5	4	3	5	5	2	1	2
FATIGUE STR. (1400°F) (4)x10 ⁻³	-	-	22	-	37	-	-	55	-	-
RESISTANCE TO FATIGUE (2)	6	7	5	4	3	5	5	1	2	2
THERMAL CONDUCTIVITY (5)	113	98	80	80	82	76	78	68	94	100
THERMAL EXPANSION (6)	8.9	8.0	7.9	7.9	7.8	7.8	7.6	7.1	6.4	6.6
RECOMMENDATIONS FOR LSSE SERVICE:										
COAL GAS/FLUIDIZED BED		1	3			2				
HOT SIDE CYL/SINGLE WALL AND/OR HEAT EXCHANGER TUBING					2			1	3	4
HOT SIDE CYL/DUAL WALL	1	2			3					
FINS (AIR @ ~1750°F)			1				2	3		

(1) Comparisons made at 1500°F, unless otherwise specified.

(2) Rating: (1) Best to (9) Worst.

(3) Rupture Strength x 10³ PSI.

(4) Rotating Beam, A = ∞, R = -1, x 10³ PSI.

(5) Thermal Conductivity = BTU/FT²/IN./HR./°F

(6) Thermal Expansion = IN./IN./°F x 10⁻⁶

TABLE II. NOMINAL CHEMISTRY (WT %), COST AND DELIVERY OF
CANDIDATE ALLOYS

ALLOY	Fe	Ni	Cr	C	Co	Mo	Cb	Al	Ti	W	La	BASE MATERIAL COST \$/LB.	DELIVERY MONTHS
316(L) S.S.	Bal.	12	17	.03	+-	2.5	--	--	--	-	--	1.50	1 to 3
310(S) S.S.	Bal.	20	25	.08	--	-	--	--	--	-	--	1.75	2 to 4
(1) INCOLOY 800 800H	Bal.	33	21	.03 1.0	--	-	--	.3	.3	-	--	2.90	4 to 8
INCOLOY 802	Bal.	33	22	.40	--	-	--	--	--	-	--	3.44	6 to 10
INCOLOY 825 (2)	30	Bal.	21	.05	.10	3.0	--	0.1	0.9	-	--	3.75	8 to 12
INCONEL 601	16	Bal.	23	.07	--	-	--	1.3	1.3	-	--	4.77	8 to 12
INCONEL 625	2.5	Bal.	22	.05	--	9.0	3.6	.02	.02	-	--	8.66	16 to 18
INCONEL 617	-	Bal.	22	.07	12.5	9.0	--	--	1.0	-	--	9.52	8 to 12
HAYNES 188	3.0	22	22	.10	Bal.	-	--	--	--	14	.10	~12.00	8 to 12

NOTES: (1) Incoloy 800 and 804 have same chemistry except "C" is .07 for Incoloy 800H.

(2) 2.25% Cu included in Incoloy 825 chemistry.

Corrosion of Stainless Steels in Sulfur Vapor at 1060°F (571°C)

Material type	Corrosion rate* mils/yr (mm/yr)
314	16.9 (0.43)
310	18.9 (0.48)
309	22.3 (0.57)
304	27.0 (0.69)
302B	29.8 (0.76)
316	31.1 (0.79)
321	51.5 (1.30)

*Corrosion rates based on 1295-h tests, International Nickel Company data.

Corrosion Rates of Chromium and Chromium-Nickel Steels in 100% H₂S at Atmospheric Pressure (120-h Test)⁴³

Material	Corrosion rate, mils/yr (mm/yr)	
	400°C	500°C
5% Cr steel	240 (6.1)	1000 (25.4)
9% Cr steel	200 (5.08)	700 (17.8)
410	130 (3.31)	400 (10.2)
430	90 (2.29)	200 (5.08)
446		100 (2.54)
301	80 (2.03)	200 (5.08)
310	60 (1.52)	100 (2.54)

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1977.

Sulfidation Rates of Austenitic Stainless Steels in 50 vol % H₂-50 vol % H₂S at Atmospheric Pressure (150-h Tests)⁴⁴

Material type	Corrosion rate, mils/yr (mm/yr)			
	500°C (932°F)	550°C (1022°F)	600°C (1112°F)	700°C (1292°F)
304	44 (1.13)	62 (1.55)	116 (2.93)	400 (10.16)
310	36 (0.91)	57 (1.45)	110 (2.79)	352 (8.94)
316	59 (1.5)	96 (2.44)	175 (4.45)	424 (10.77)

ALUMINUM DIFFUSION-COATED SAMPLES

Material type	304	310	316
		7 (0.18)	33 (0.84)
		3 (0.06)	27 (0.69)
		4 (0.10)	29 (0.74)

Recommended Maximum Service Temperatures in Sulfur Dioxide for Selected Stainless Steels

Material type	Temperature	
	°C	°F
304	800	1472
321	800	1472
347	800	1472
310	1030	1922
410	700	1292
430	800	1472
446	1025	1875

Oxidation of Type 302 Stainless Steel in Sulfur Dioxide at 1650°F (898°C) for 24 h⁴⁴

Condition	Weight gain, mg/cm ²
Atmosphere	46
Atmosphere + 2% SO ₂	113
Atmosphere + 5% SO ₂ + 5% H ₂ O	353

Corrosion Rate of Stainless Steels in Flue Gases (Exposure 3 months)⁴

Material type	Corrosion rate, mils/yr (mm/yr)		
	Coke oven gas (1500°F (815°C))	Coke oven gas (1800°F (981°C))	Natural gas (1500°F (815°C))
430	91 (2.31)	236 (5.99)	12 (0.30)
446 (26 Cr)	30 (0.76)	40 (1.02)	4 (0.10)
446 (28 Cr)	27 (0.69)	14 (0.36)	3 (0.07)
302B	104 (2.64)	223 (5.72)	
301S	37 (0.94)*	43 (1.11)	3 (0.07)
310S	38 (0.97)*	25 (0.64)	3 (0.07)
314	23 (0.58)*	94 (2.39)	3 (0.07)

Table III. Effect of Various Gaseous Environments and Temperatures on Stainless Steels

Table IV. MPC/IITRI Corrosion Coupon Data
from Fluidized Bed of Gasifier

Alloy Material	Scaling Loss (mils)	Rate of Scaling Loss		Depth of Penetration (mils)	Total Sound Metal Loss ⁽¹⁾	
		(mpy)	(mm/y)		(mpy)	(mm/y)
304 Stainless	0.2	2	0.06	0.2	5	0.13
309 Stainless	0.2	2	0.06	0.2	4	0.11
Incoloy 800	0.2	2	0.06	0.2	4	0.11
Incoloy 825	0.2	2	0.06	0.2	4	0.11
Hastelloy X	0.2	2	0.06	0.3	5	0.13
RA-333	0.2	2	0.06	0.2	4	0.10
316 Stainless	0.2	2	0.06	2.0	25	0.63
321 Stainless	0.2	2	0.06	1.3	17	0.43
21Cr-6Ni-9Mn	0.2	2	0.06	0.5	7	0.19
22Cr-13Ni-5Mn	0.2	2	0.06	0.3	6	0.14
430 Stainless	0.2	2	0.06	0.7	10	0.24
Inconel 600	43.9	492	12.49	34.4	878	22.30
Inconel 601	0.2	2	0.06	1.4	18	0.46

(1) Based on linear extrapolation of the sound metal loss measured after 781.1 hours under operating conditions. Coal feed: Montan Rosebud (low sulfur, noncaking coal); temperature range: 427 to 843 C (800 to 1550 F); average temperature: 779 C (1434 F); and pressure: 4.2 MPa (600 psig).

Table V. MPC/IITRI Corrosion Data from
Vapor Zone of Gasifier

Alloy Material	Scaling Loss (mils)	Rate of Scaling Loss		Depth of Penetration (mils)	Total Sound Metal Loss ⁽¹⁾	
		(mpy)	(mm/y)		(mpy)	(mm/y)
304 Stainless	0.2	2	0.06	0.8	11	0.29
309 Stainless	0.2	2	0.06	0.6	9	0.23
Incoloy 800	0.2	2	0.06	0.5	7	0.19
Incoloy 825	0.2	2	0.06	0.2	4	0.11
Hastelloy X	0.2	2	0.06	0.3	6	0.14
RA-333	0.2	2	0.06	0.6	2	0.04
316 Stainless	0.2	2	0.06	1.5	20	0.50
321 Stainless	0.4	4	0.10	1.3	18	0.46
21Cr-6Cr-9Mn	0.2	2	0.06	0.4	7	0.17
22Cr-13Ni-5Mn	0.2	2	0.06	0.4	7	0.17
430 Stainless	0.2	2	0.06	0.4	6	0.16
Inconel 600	36.8	412	10.47	15.1	582	14.77
Inconel 601	0.5	5	0.13	0.5	11	0.27

(1) Based on linear extrapolation of the sound metal loss measured after 781.8 hours under operating conditions. Coal feed: Montana Rosebud (low sulfur, noncaking coal); temperature range: 427 to 843 C (800 to 1500 F); average temperature: 696 C (1284 F); and pressure: 4.2 MPa (600 psig).

Materials Performance at the Synthane Coal Gasification Plant*

MATERIALS PERFORMANCE, NACE, Oct. 1979.

B. N. GREENE

C-E Lummus Company, Bloomfield, New Jersey

Table VI. Composition of Alloys Selected for Testing in Phase I

Alloy	Composition (Wt%)									
	C	Mn	Si	Fe	Ni	Cr	Co	Al	Mo	Other
Series 1										
AISI 302	0.05	0.64	0.55	71.02	9.24	18.18	—	—		
AISI 304	0.05	1.45	0.54	70.04	9.10	18.76	—	—		
AISI 316	0.05	1.65	0.43	65.28	13.88	17.14	—	—		
AISI 309	0.11	0.54	0.74	60.46	14.70	22.97	—	—		
AISI 314	0.06	1.90	2.21	51.64	20.00	24.00	—	—		
AISI 310	0.06	1.71	0.68	52.16	20.20	25.00	—	—		
AISI 310(Al)										
AISI 310(Cr)										
AISI 446	0.10	0.45	0.38	74.55	0.37	24.00				
IN 600	0.05	0.15	0.20	6.99	76.48	15.82	0.03	—		
IN 601	0.04	0.24	0.19	15.86	59.52	22.50	0.03	1.19		
IN 800	0.03	0.80	0.33	47.08	30.84	20.60	0.10	0.32		0.42 Ti
IN 800(Al)										
IN 800(Cr)										
IN 793	0.03	0.72	0.40	43.22	32.22	21.40	0.02	1.76		
IN 671 (50/50)	0.03	0.03	0.03	0.30	48.40	50.00	0.01	—		0.35 Ti
Series 2										
HC 250	3.03	0.6	0.7	68.1	0.1	27.5	—	—	0.1	
HD 45	0.48	0.7	1.5	62.1	5.2	29.9	—	—	0.1	
HL 40	0.47	0.6	1.4	47.1	19.4	30.9	—	—	0.1	
HL 40-3Si	0.42	0.7	2.4	45.8	19.3	31.4	—	—	0.1	
TA-333	0.05	1.5	1.4	15.5	47.5	26.2	3.0	—	3.8	2.7 W
Crutemp 25	0.07	1.5	0.6	47.2	24.8	25.4	—	—	0.4	
Multimet N155	0.11	1.4	0.7	29.1	19.8	21.8	19.5	—	3.0	1.1 Cb+Ta, 3.9W
Haynes 150	0.06	0.5	0.2	16.5	1.7	27.9	49.6	—	0.1	
Haynes 188	0.08	0.7	0.4	1.4	23.3	23.4	35.7	0.22	0.6	0.51 La, 14.6W
Stellite 6B	1.0	1.4	0.6	2.0	2.4	28.5	56.4	—	1.1	6.5 W
VE 441	0.03	0.1	0.1	1.5	0.01	0.01	—	15.1	3.2	0.04 Zr
N1	0.004	0.2	2.3	0.20	90.0	2.8	—	4.4	0.1	
Armco 21-6-9	0.04	8.2	0.7	62.9	6.8	20.7	—	—	0.22	
Series 3a										
AISI 312	0.15	2.0	0.5	Bal	9	30	—	—	—	
AISI 329	0.05	0.5	0.4	Bal	4.3	27.1	—	—	1.4	
AL 29-4-4	0.005	—	—	Bal	4.0	29	—	—	4	
AL EX-20	1.0	20	—	Bal	3	5	—	10	—	
Co-Cr-W No. 1	2.5	—	—	—	—	30	Bal	—	—	12 W
Thermalloy 63WC	0.4	—	—	Bal	35	26	15	—	—	5 W
Wiscalloy 30/50N	0.55	—	—	Bal	47	27	—	—	—	4 W
Armco 18SR	0.05	—	1.0	Bal	—	18.0	—	2.0	—	0.40 Ti
Armco 22-13-5	0.06	5.0	—	Bal	12.5	22.0	—	—	2.0	0.20 Cb, 0.20 V, 0.3 N
Hastelloy X	0.15	1.0	1.0	18.5	Bal	21.8	2.5	—	9.0	0.6 W
IN 625	0.05	0.25	0.25	2.5	Bal	21.5	—	0.2	9.0	0.2 Ti, 3.65 Cb
Sanicro 32X	0.08	—	—	Bal	31	21	—	0.35	—	0.35 Ti, 3W
IN 825	0.03	0.50	0.25	30	41	21.5	—	0.10	3.0	2.25 Cu, 0.9 Ti
Series 3b										
FSX-414	0.25	1.0	1.0	2.0	10.5	29.5	Bal	—	—	7.0 W
HK 40	0.4	2.0	2.0	Bal	20	28	—	—	0.5	
HK 40-3Si	0.35/ 0.45	2.0	3.0	Bal	18/22	24/28	—	—	0.5	
Thermalloy 63	0.4	—	—	Bal	35	26	—	—	—	
Thermalloy 63W	0.40	—	—	Bal	35	26	—	—	—	5 W
RA-330	0.05	1.5	1.25	43	35.0	19.0	—	—	—	
IN 657	—	—	—	—	50	48	—	—	—	1.5 Cb
IN 738	0.17	0.2	0.3	0.5	Bal	16.0	8.5	3.4	1.75	3.4 Ti, 0.1 Zr, 2.6 W, 0.9 Cb
Haynes 556	0.1	1.5	0.4	Bal	20	22	20	0.3	3.0	2.5 W, 0.02 La, 1.0 Cb+Ta
IN 617	0.07	—	—	—	54	22	12.5	1.0	9	

Table VII. Summary of Linearly Extrapolated Corrosion Rates of Phase 1 Alloys in Coal Gasification Atmospheres ⁽¹⁾

Temp.	Materials Grouping by Corrosion Rate (mpy) ⁽²⁾											
	0 H ₂ S			0.1 H ₂ S			0.5 H ₂ S			1.0 H ₂ S		
	<20	20-40	>40	<20	20-40	>40	<20	20-40	>40	<20	20-40	>40
Gasifier Gas Composition ⁽³⁾ - 1000 psi												
815 C												
1500 F	-	-	-	302	800(Cr) ⁽⁴⁾		800	316	600	314	304	800(Cr)
				316	310(Cr) ⁽⁴⁾		310		601	800		310(Cr)
				600	800		309		304	310		302
				304	601		446			309		316
				314			800(Al) ⁽⁵⁾			800(Al)		
				309			793			310(Al)		
				446			50/50			793		
				800(Al)						50/50		
				310(Al)								
				793								
				50/50								
980 C												
1800 F	310(Cr)	800(Cr)	302	800(Cr)	601	302	800(Al)	800	316	800(Al)	314	800(Cr)
	601	800	316	310(Cr)	600	316	671	310	600	671	800	310(Cr)
	310	309	304	800(Al)	800	304	HD 45	309	309	HC 250	446	302
	800(Al)	793	314	310(Al)	793	314	HL 40	446	304	HL 40-3Si	310(Al)	316
	310(Al)	600	446	HL 40	HD 45	310	RA 333	310(Al)	793	Cru 25	HD 45	600
	671	310		HL 40-3Si	Cru 25	309	Cru 25	HC 250	601	N155	HL 40	601
				RA 333		446	N155	HL 40-3Si	21-6-9	150	RA 333	304
				N155		21-6-9	188	150		188		314
				150		HC 250	68			68		800
				188		VE 441						310
				68								309
				671								446
												793
												21-6-9
												VE 441
Regenerator Gas Composition ⁽⁶⁾ - 150 psi												
1010 C												
1850 F	-	-	-	314	800(Cr)	302						
				310	310(Cr)	316						
				800(Al)	309	600						
				310(Al)	671	601						
						304						
						800						
						446						
						793						

(1) Based on linear extrapolation of total corrosion (scaling + internal corrosion) in 1000 hours.

(2) 1 mpy = 0.0254 mm/y.

(3) Inlet gas composition: 24 H₂, 18 CO, 12 CO₂, 5 CH₄, 1 NH₃, 0.1 H₂S, bal. H₂O (Vol%).

(4) (Cr) - chromized.

(5) (Al) - aluminized.

(6) Inlet gas composition: 67 N₂, 30 CO₂, 1 CO, 1 H₂, 0.13 SO₂ (Vol%).

Metallic Corrosion in Coal Gasification Pilot Plants*

MATERIALS PERFORMANCE, NACE, Oct. 1978.

V. L. HILL and M. A. H. HOWES
IIT Research Institute, Chicago, Illinois

Table VIII. Comparison of Laboratory and Pilot Plant Corrosion Tests of Conoco Regenerator Environment *

Alloy	Phase I Laboratory Test ⁽¹⁾		Phase II Conoco Pilot Plant				Conoco Tests ⁽⁶⁾
	Total Corrosion ⁽⁴⁾ (mils)	Corrosion Rate ⁽⁵⁾ (mpy)	Exposure 1 ⁽²⁾		Exposure 2 ⁽³⁾		
			Total Corrosion (mils)	Corrosion Rate (mpy)	Total Corrosion (mils)	Corrosion Rate (mpy)	
AISI 304	35.7	313	53.9	645	(9)	(9)	(7)
AISI 309	2.4	21	11.1	122	74.6	408	> 298
AISI 310	1.5	13	5.8	64	75.2	412	38
21-6-9	—	—	7.1	78	36.3	199	—
22-13-5	—	—	7.8	85	100.6	551	32
AISI 446	(9)	(9)	2.8	31	Not tested		> 364 ⁽⁸⁾
IN 800	8.1	71	19.9	218	(9)	(9)	145
IN 793	9.1	79	13.6	149	57.9	317	177
IN 825	—	—	68.8	753	(9)	(9)	—
Hastelloy X	—	—	40.9	438	117.2	642	(7)
IN 671	2.8	25	(9)	(9)	(9)	(9)	(7)
AISI 310(AI)	1.5	13	5.9	65	14.6	80	92
IN 800(AI)	1.5	13	5.9	65	14.5	79	121

(1) 1850 F, 150 psi, 1000 hours, gas composition from Table 2.

(2) 1850 F, 150 psi, ~ 800 hours above 1700 F.

(3) 1850 F, 150 psi, ~ 1600 hours above 1700 F.

(4) 1 mil = 25.4 μ m.

(5) 1 mpy = 0.0254 mm/y.

(6) Reproduced from Reference 5, 1127 hours above 1700 F, 2319 hours total exposure.

(7) Not tested, extensive corrosion in previous shorter time test.

(8) Data for similar alloy E Brite (Fe-26Cr-1Mo).

(9) Completely destroyed.

Metallic Corrosion in Coal Gasification Pilot Plants*

MATERIALS PERFORMANCE, NACE, Oct. 1978.

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UT Research Institute, Chicago, Illinois

Table IX. Eutectic Temperatures of Selected Metal-Metal Sulfide Systems (*)

Metal	Eutectic	MP of Sulfide		Eutectic Temperature		Sulfur Concentration at Eutectic	
		C	F	C	F	At%	Wt%
Nickel	Ni-Ni ₃ S ₂	810 ⁽²⁾	1490	645	1193	33.4	21.5
Cobalt	Co-Co ₄ S ₃	932 ⁽²⁾	1710	877	1611	40	26.6
Iron	Fe-FeS	1190	2174	988	1810	44	31
Chromium	Cr-CrS	1565	2849	1350	2462	43.9	32.5

(*) Reference above.

(2) Formed peritectically.

Table X

Stress-Rupture Lives of Three High Temperature Alloys*

ALLOY	INTERNAL GAS	STRESS, psi	TEMPERATURE, °F	LIFE, hr	AVERAGE LIFE, hr	STANDARD DEVIATION, hr	CONFIDENCE LEVEL, percentile
A-286	Air	57,000	1200	230.6 253.3 254.2 264.3 3.6.8	264	± 32	83
	Hydrogen	57,000	1200	109.7 125.0 188.9 246.6 280.0 340.0			
Inconel 700	Air	40,000	1500	167.3 188.3 196.7 282.0	209	± 51	20
	Hydrogen	40,000	1500	154.6 198.3 206.3 207.3 266.2			
S-816	Air	19,000	1500	166.6 228.3 232.8 232.8	215	± 32	98
	Hydrogen	19,000	1500	120.5 130.6 140.4 161.0 200.2			

* NASA-D-1458

Table XI. Average Stress, ksi (MN/m²), to Produce Rupture in 10,000 h for Several Austenitic Stainless Steels*

Temperature		Grades					
°F	°C	304	304L	316	316L	321	347
1000	538	36 (248.2)	25 (172.4)	43 (296.5)	39 (268.9)	40 (275.8)	48 (331)
1050	566	28 (193.1)	20 (137.9)	34 (234.4)	30.5 (210.3)	31 (213.7)	36 (248.2)
1100	593	22.2 (153.1)	15.6 (107.6)	26.5 (182.7)	23.5 (162)	23.5 (162)	27.5 (189.6)
1150	621	17.3 (119.3)	12.2 (84.1)	20.8 (143.4)	18.2 (125.5)	17.3 (119.3)	20.3 (141.3)
1200	649	13.8 (95.2)	9.7 (66.9)	16.2 (111.7)	14.2 (97.9)	12.9 (88.9)	15.6 (107.6)
1250	677	10.8 (74.5)	7.6 (52.4)	12.7 (87.6)	11 (75.8)	9.7 (66.9)	11.9 (82.1)
1300	704	8.5 (58.6)	6 (41.4)	9.9 (68.3)	8.5 (58.6)	7.2 (49.6)	9 (62.1)
1350	732	6.7 (46.2)	4.7 (32.4)	7.7 (53.1)	6.6 (45.5)	5.4 (37.2)	6.8 (46.9)
1400	760	5.3 (36.5)	3.7 (25.5)	6 (41.4)	5.1 (35.2)	4 (27.6)	5.1 (35.2)
1450	788	4.15 (28.6)	2.9 (20)	4.7 (32.4)	3.95 (27.2)	3.05 (21)	3.85 (26.5)
1500	816	3.25 (22.4)	2.3 (15.9)	3.7 (25.5)	3.05 (21)	2.28 (15.7)	2.9 (20)

Table XII. Average Stress, ksi (MN/m²), to Produce Rupture in 100,000 h for Several Austenitic Stainless Steels*

Temperature		Grades					
°F	°C	304	304L	316	316L	321	347
1000	538	25.8 (177.9)	19.5 (134.5)	37 (25.5)	34.5 (237.9)	29 (200)	37.5 (258.6)
1050	566	20 (137.9)	15 (103.4)	28 (193.1)	25 (172.4)	23 (158.6)	28 (193.1)
1100	593	15.8 (108.9)	11.6 (80)	20.8 (143.4)	18.5 (127.6)	16.5 (113.8)	20.8 (143.4)
1150	621	12.5 (86.2)	8.9 (61.1)	15.1 (104.1)	13.7 (94.5)	12 (82.7)	15.3 (105.5)
1200	649	9.8 (67.6)	6.9 (47.6)	11.4 (78.6)	10.1 (69.6)	8.7 (60)	11.5 (79.1)
1250	677	7.6 (52.4)	5.3 (36.5)	8.4 (57.9)	7.4 (51)	6.3 (43.4)	8.6 (59.3)
1300	704	6 (41.4)	4.1 (28.3)	6.3 (43.4)	5.5 (37.9)	4.6 (31.7)	6.4 (44.1)
1350	732	4.7 (32.4)	3.15 (21.7)	4.7 (32.4)	4 (27.6)	3.3 (22.8)	4.7 (32.4)
1400	760	3.7 (25.5)	2.4 (16.5)	3.45 (23.8)	3 (20.7)	2.45 (16.9)	3.55 (24.5)
1450	788	2.9 (20)	1.87 (12.9)	2.6 (17.9)	2.2 (15.2)	1.75 (12.1)	2.64 (18.7)
1500	816	2.3 (15.9)	1.45 (10)	1.95 (13.4)	1.6 (11)	1.27 (8.8)	1.95 (13.4)

* Stainless Steel Handbook, 1967

Table XIII. Wrought Alloys

		100 HOUR RUPTURE STRENGTH, 1000 psi MN/m ²																							
Ref. No.	Alloy Designation	Form	1200F	849C	1130F	704C	1140F	702C	1150F	816C	1160F	871C	1170F	927C	1180F	882C	1190F	1034C	7000F	1033C	1210F	1149C	2200F	121	
Nickel Base Alloys																									
1	Astroloy	Bar	125	860	105	725	78	540	58	405	37	255	25	175	15	105	—	—	—	—	—	—	—	—	—
2	D-979	Bar	90	620	71	490	55	380	33	230	20	140	—	—	—	—	—	—	—	—	—	—	—	—	—
3	HASTELLOY [®] alloy X	Sheet	42	290	31	215	21	145	14	97	90	62	60	41	38	26	22	15	12	8	—	—	—	—	—
4	HASTELLOY [®] alloy S	Bar	57	395	38	260	24	165	15	105	95	68	—	—	—	—	—	—	—	—	—	—	—	—	—
5	INCO [®] alloy 600	Bar	—	—	—	—	—	—	80	55	53	37	—	—	28	19	—	—	14	10	11	8	—	—	—
6	INCO [®] alloy 601	Bar	38	260	21	145	13	90	98	68	70	48	—	—	34	23	—	—	18	11	12	8	—	—	—
7	INCO [®] alloy 617	Bar	68	470	44	305	30	205	20	140	13	90	95	66	60	41	42	29	27	19	—	—	—	—	—
8	INCO [®] alloy 625	Bar	64	440	46	315	32	220	18	125	11	76	—	—	46	32	—	—	18	12	—	—	—	—	—
9	INCO [®] alloy 690	Bar	25	170	18	110	94	65	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
10	INCO [®] alloy 706	Bar	100	690	74	510	43	295	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
11	INCO [®] alloy 718	Bar	105	725	74	510	44	305	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
12	INCO [®] alloy X750	Bar	80	550	—	—	—	—	26	180	12	83	82	57	32	22	—	—	—	—	—	—	—	—	—
13	INCO [®] alloy MA 754	Bar	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	19	131	—	—	—	—	—
14	IN-102	Bar	67	460	37	255	20	140	12	83	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
15	IN-587	Bar	—	—	79	545	58	420	40	275	23	160	—	—	—	—	—	—	—	—	—	—	—	—	—
16	IN-597	Bar	—	—	88	605	67	460	48	330	30	205	17	115	15	66	—	—	—	—	—	—	—	—	—
17	M-252	Bar	98	675	73	505	52	360	35	240	23	160	—	—	—	—	—	—	—	—	—	—	—	—	—
18	NIMONIC [®] alloy 75	Bar	37	255	20	140	11	76	57	39	38	25	21	14	14	10	0.8	6	—	—	—	—	—	—	—
19	NIMONIC [®] alloy 80A	Bar	86	595	64	440	45	310	23	195	14	97	70	48	20	14	—	—	—	—	—	—	—	—	—
20	NIMONIC [®] alloy 91	Bar	70	435	54	370	35	240	22	140	10	69	—	—	—	—	—	—	—	—	—	—	—	—	—
21	NIMONIC [®] alloy 80	Bar	86	555	66	455	49	340	32	220	18.5	130	8.5	59	25	17	—	—	—	—	—	—	—	—	—
22	NIMONIC [®] alloy 105	Bar	115	815	90	620	70	485	46	320	29	200	18	125	59	61	44	30	—	—	—	—	—	—	—
23	NIMONIC [®] alloy 115	Bar	—	—	96	660	76	525	58	400	41	285	27	185	16	110	90	62	44	30	20	14	—	—	—
24	NIMONIC [®] alloy 263	Sheet	73	505	54	370	38	260	24	165	13	90	60	41	24	17	—	—	—	—	—	—	—	—	—
25	NIMONIC [®] alloy 942	Bar	93	640	73	500	51	350	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
26	NIMONIC [®] alloy PE11	Bar	65	445	47	325	31	215	17.5	120	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
27	NIMONIC [®] alloy PE16	Bar	68	470	47	325	28	195	16	110	7.3	50	—	—	—	—	—	—	—	—	—	—	—	—	—
28	NIMONIC [®] alloy PK33	Sheet	102	705	80	550	61	420	43	300	25	175	13	90	54	37	—	—	—	—	—	—	—	—	—
29	PYROMET [®] 860	Bar	98	675	74	510	52	360	35	240	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
30	RENÉ 41 [®]	Bar	110	760	55	665	68	470	45	310	28	195	18	125	10	69	—	—	—	—	—	—	—	—	—
31	RENÉ 41 [®]	Sheet	92	635	75	515	55	360	40	275	26	180	18	125	90	62	—	—	—	—	—	—	—	—	—
32	RENÉ 95	Bar	150	1035	105	725	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
33	RGT [®] 4	Bar	85	585	66	455	45	310	28	195	15	135	—	—	—	—	—	—	—	—	—	—	—	—	—
34	RGT [®] 13	Bar	85	585	69	475	50	345	35	240	22	150	—	—	—	—	—	—	—	—	—	—	—	—	—
35	TD Nickel	Bar	28	195	26	180	23	160	21	145	19	130	17	115	15	105	13	90	11	76	90	62	80	—	—
36	TD Nickel	Sheet	22	150	20	140	18	125	16	110	14	97	13	90	12	83	10	69	86	55	65	45	50	—	—
37	TD NiCr	Sheet	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
38	UDIMET [®] 400	Bar	110	760	85	585	62	430	43	295	26	180	14	97	—	—	—	—	—	—	—	—	—	—	—
39	UDIMET [®] 500	Bar	135	930	99	680	66	455	44	305	29	200	20	140	12	83	—	—	—	—	—	—	—	—	—
40	UDIMET [®] 520	Bar	—	—	81	560	63	450	47	325	32	220	21	145	—	—	—	—	—	—	—	—	—	—	—
41	UDIMET [®] 630	Bar	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
42	UDIMET [®] 700	Bar	—	—	98	675	78	540	58	400	43	295	27	185	16	110	80	55	—	—	—	—	—	—	—
43	UDIMET [®] 710	Bar	140	980	93	780	89	600	61	440	45	310	30	205	19	130	—	—	—	—	—	—	—	—	—
44	UNITEMP [®] AF2-10A	Bar	150	1035	120	830	95	655	70	485	52	360	36	250	22	150	—	—	—	—	—	—	—	—	—
45	UNITEMP [®] AF2-10A	Sheet	145	1000	115	815	90	620	70	485	52	360	36	250	18	125	—	—	—	—	—	—	—	—	—
46	WASPALLOY [®]	Bar	110	760	85	585	60	415	40	275	25	175	14	97	65	45	—	—	—	—	—	—	—	—	—
47	WASPALLOY [®]	Sheet	110	760	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
Cobalt Base Alloys																									
48	HAYNES [®] alloy No. 188	Sheet	—	—	47	325	33	230	23	160	15	105	10	69	60	41	38	26	22	15	—	—	—	—	—
49	L-605	Sheet	48	330	39	270	29	195	22	150	15	105	10	69	70	46	48	33	28	19	—	—	—	—	—
Iron-Nickel Alloys																									
50	Alloy 901	Bar	92	635	70	485	44	305	24	165	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
51	A-286	Bar	61	420	44	305	27	185	13	90	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
52	DISCALOY [®]	Bar	49	340	30	220	18	125	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
53	HAYNES [®] alloy 556	Sheet	53	365	37	255	26	180	17	115	11.5	80	7.5	52	47	32	28	19	16	11	—	—	—	—	—
54	INCOLOY [®] alloy 800	Bar	33	230	22	150	13	90	92	63	60	41	43	30	31	21	—	—	—	—	—	—	—	—	—
55	INCOLOY [®] alloy 801	Bar	37	255	—	—	—	—	92	63	—	—	—	—	24	16	—	—	—	—	—	—	—	—	—
56	INCOLOY [®] alloy 802	Bar	36	250	—	—	21	145	—	—	14	97	9.5	66	7	48	—	—	3	21	2	14	—	—	—
57	INCOLOY [®] alloy 807	Bar	—	—	31	215	21	145	14	97	10	69	6.5	45	45										

Table XIV. Stress (PSI) to Produce Rupture at Indicated Temperature.

ALLOY	1300°F		1400°F		1500°F	
	10,000 HR	100,000 HR	10,000 HR	100,000 HR	10,000 HR	100,000 HR
316 STAINLESS	9,900	6,300	6,000	3,400	3,700	1,950
INCOLOY 800H	11,000	8,000	7,300	5,300	5,200	3,700
INCOLOY 802	14,000	10,000	12,000	8,000	8,200	5,500
INCONEL 625	25,000	----	16,000	---	9,500	6,000
INCONEL 617	26,000	----	16,000	---	12,000	---

Table XV. Stress (PSI) to Produce 1% Creep
of Candidate Alloys at Elevated
Temperatures

ALLOY	1300°F		1400°F		1500°F	
	10,000 HR	100,000 HR	10,000 HR	100,000 HR	10,000 HR	100,000 HR
316 STAINLESS	7,500	---	---	---	2,400	---
INCOLOY 800H	9,000	---	6,000	---	5,000	---
INCOLOY 802	15,000	---	13,000	---	7,500	4,500
INCONEL 625	16,000	---	9,000	---	4,500	---
INCONEL 617	22,000	---	12,500	10,000	7,500	---

Table XVI. Fatigue Strength (1000 PSI) of Some Alloys (1)

MATERIALS	STRENGTH (1000 PSI)				TEST TEMPERATURE
	CYCLES				
	10 ⁵	10 ⁶	10 ⁷	10 ⁸	
INCONEL 625	-	64	60	55	1400°F
INCONEL 718	83	69	54	40	1400°F
INCOLOY 802	42	40	38	37	1400°F
INCONELX 750	-	-	44	38	1350°F
NIMONIC 80A	-	-	36	29	1400°F
INCOLOY 801	39	38	33	26	1350°F
INCOLOY 800	-	2	25	22	1400°F
INCONEL 600	-	-	-	15	1400°F
INCONEL 625	80	77	72	70	1200°F
MULTIMET (N155)	-	-	-	66	1200°F
D979	-	-	-	60	1200°F
19-9 DL	-	-	-	52	1200°F
UDIMET 700	-	-	49	-	1200°F
WASPALLOY	-	-	40	-	1200°F
S590	-	-	-	33	1500°F
MULTIMET (N155)	-	-	-	33	1500°F
INCOLOY 801	-	21	16	11	1500°F
INCONEL 617	-	-	27	22	1600°F
INCOLOY 802	25	22	19	17	1600°F
INCOLOY 800	-	20	16	14	1600°F
INCONEL 625	-	38	27	-	1600°F
INCONEL 600	-	-	-	10	1600°F

(1) Rotating Beam, Smooth Bar Specimen

A = ∞ , R = -1

Huntington and Cabot Brochures plus Aerospace Structural Metals Handbook

Table XVII. Some Physical Properties of Candidate Alloys

ALLOY	THERMAL EXPANSION IN./IN./°F x 10 ⁻⁶ (1)	THERMAL CONDUCTIVITY BTU/FT ² /IN./HR./°F	MELTING RANGE °F	DEGREES DELTA (MELTING)
304 STAINLESS	9.6 = 1.10X	113 = X	2550-2650	+50°
347 STAINLESS	9.3 = 1.05X	112 = X	2550-2600	+50°
321 STAINLESS	9.3 = 1.05X	112 = X	2500-2600	0°
A286 STAINLESS	9.2 = 1.03X	88 = 0.8X	2500-2550	0°
316(L) STAINLESS	8.9 = X	113 = X	2500-2550	0°
INCOLOY 800	7.9 = 0.9X	80 = 0.7X	2475-2525	-25°
INCOLOY 802	7.8 = 0.9X	82 = 0.7X	2450-2500	-50°
INCONEL 625	7.1 = 0.8X	68 = 0.6X	2350-2460	-150°
INCONEL 718	6.8 = 0.7X	79 = 0.7X	2300-2435	-200°
INCONEL 617	6.4 = 0.7X	94 = 0.83X	2430-2510	-70°

(1) (Room to 212°F)

Recommended Maximum Service Temperatures in Air for Wrought Stainless Steels

Material type	Intermittent service		Continuous service	
	°C	°F	°C	°F
201	815	1500	915	1650
202	815	1500	845	1550
301	840	1550	900	1650
302	870	1600	925	1700
304	870	1600	925	1700
308	925	1700	980	1800
309	950	1750	1095	2000
310	1035	1900	1150	2100
316	870	1600	925	1700
317	870	1600	925	1700
321	870	1600	925	1700
330	1035	1900	1150	2100
347	870	1600	925	1700
406	815	1500	1035	1900
410	815	1500	705	1300
416	760	1400	675	1250
420	735	1350	620	1150
440	815	1500	760	1400
405	815	1500	705	1300
430	870	1600	815	1500
412	1035	1900	950	1800
446	1175	2130	1095	2000

Resistance of austenitic stainless steels to cyclic conditions at 1800°F (980°C) (cycle consists of 15 min at temperature and 5 min in air).¹⁴

Oxidation of Types 302 (upper) and 330 (lower) in wet and dry air.¹²

HDBK. of STAINLESS STEELS,
PECKNER & BERNSTEIN,
1977

Effect of carbon dioxide pressure on the carburization and oxidation of Type 304

Figure 1. Oxidation of Stainless Steels

Figure 2. Cyclic Oxidation Resistance at 1800°F. Each Cycle was 15 Minutes Heating and 5 Minutes Cooling in Air

* HUNTINGTON BROCHURE

(2) Huntington alloys.

Figure 3. Carburization Resistance at 1800°F and 2000°F (1)

FIGURE 4 - Effect of H₂S concentration on 1000 hour total corrosion of Series 1 and 2 Fe base alloys at 980 C (1800 F).

Metallic Corrosion in Coal Gasification Pilot Plants*

MATERIALS PERFORMANCE, NACE, Oct. 1979.

V. L. HILL and M. A. H. HOWES

IIT Research Institute, Chicago, Illinois

FIGURE 5 - Effect of H₂S concentration on 1000 hour total corrosion of Series 1 and 2 high Ni alloys at 980 C (1800 F).

FIGURE 6 - Effect of H₂S concentration on 1000 hour total corrosion of Series 1 and 2 Co containing alloys at 980 C (1800 F).

(1)
Fig. 7 Safe operating limits for carbon and alloy steels in contact with hydrogen at high temperatures and pressures (after G. A. Nelson, "Metals for High Pressure Hydrogenation Plants", Trans ASME, 73, 205 to 213, 1951; see also the 1958 revision in Oil and Gas Journal, May 25, 1959)

Test bars in reactor furnaces, 1000F, 3500 psi

(1)
Fig. 8 Effect of exposure to high-pressure hydrogenation on elongation of test bars inserted in codimer reactor furnaces at temperatures up to 1000 F and a pressure of 3500 psi and tested at room temperature soon after removal from the furnaces.

(1) METALS
HANDBOOK, Vol. 1
1977.

*
ANL-7849
Reactor Technology

Figure 9. Corrosion of Stainless Steel Foils
in Low Velocity Na at 650°C *

Alloy (AISI)	Yield strength, MPa	Loss* in ultimate tensile strength, %		Loss* in reduction of area (RA), %		Ref.
		Unnotched	Notched	Unnotched	Notched	
304L	200	11	13	59	52	1,2
305	350†	3	11	4	11	1
309S	220	0	2	0	0	5
316	215	0	0	0	0	1,2
321	220	0	12	10	64	6
347	460†	0	9	0	9	7

Alloy	Yield strength, MPa	Loss* in ultimate tensile strength, %		Loss* in reduction of area (RA), %		Ref.
		Unnotched	Notched	Unnotched	Notched	
Type 410	1325	21	78	80	73	1
Type 440C	1625	60	50	~100	~100	1
17-7PH	1035	8	77	94	60	1
AFC 77	~1400	~75	.			3

* HANDBOOK OF STAINLESS STEELS, PECKER AND BERNSTEIN, 1977

(10,000 PSI H₂ Exposures)
TESTING AT ROOM TEMPERATURE.

Figure 10. Effect of High Pressure Hydrogen on Martensitic Precipitation Hardening, and Austenitic Stainless Steels *

* HYDROGEN PERMEABILITY IN UNCOATED
AND COATED METALS

V. L. Hill and Edward J. Vesely, Jr.

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Figure 11.

Permeability versus Reciprocal of Absolute Temperature
for the Four Uncoated Iron-Base Superalloys *

* HYDROGEN PERMEABILITY IN UNCOATED
AND COATED METALS

V. L. HILL AND EDWARD J. VESELY, JR.

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Figure 12

Diffusivity versus Reciprocal of Absolute Temperature
for the Four Uncoated Iron-Base Superalloys *

* NATIONAL AERONAUTICS & SPACE ADMINISTRATION

ALLOY	Fe	Ni	Cr	Co	W	Ti	Mo
19-9 DL	BAL.	10	20	—	1.5	0.3	—
6B	2.0	2.0	29	BAL.	3.8	—	0.9

Figure 14

Permeability versus Reciprocal of Absolute Temperature*

* See Fig. 13.

* METALS HDBK., VOL. E, 1977

Figure 15. Creep and Creep-Rupture Comparisons *

FIGURE 16. COMPARATIVE CREEP RUPTURE LIFE (1500°F) OF CANDIDATE ALLOYS

High-temperature tensile properties of INCOLOY alloy 800H.

High-temperature tensile properties of cold-rolled annealed sheet, INCONEL 625.

High-temperature tensile properties of annealed INCOLOY alloy 802.

High-temperature tensile properties of solution-treated, cold-rolled sheet, INCONEL 617.

* HUNTINGTON BROCHURES

Figure 17. Comparison of the High Temperature Physical Properties of Some Alloys *

Figure 18. Comparisons Between Effects of Solution Annealed and Aged Conditions on the Yield Strength, Ultimate Tensile Strength, and Total Elongation of Inconel 625.
(ORNL-5774)

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APPENDIX D

TASK 1A.1 REPORT

STIRLING ENGINE HEAT SOURCE

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January 1980

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ABSTRACT

The integration and adaptability of pulverized-coal, cyclone, fluidized bed and oil-fired combustors as a heat source for the Stirling Engine were evaluated during Task 1A.1. Evaluation of these combustion systems and their fuel firing capabilities included:

- Limitation in the fuel specifications and operating conditions imposed by the impurities in the fuel
- Corrosion potential and material selections
- Erosion potential
- Fuel preparation requirements
- Cycle arrangement - utilization of available energy at high temperature and its impact on combustion arrangement and cycle efficiency
- Heat-transfer coefficient
- Preliminary assessment of coal.

This report presents the results of the Task 1A.1 evaluation.

TABLE OF CONTENTS

<u>Section</u>		<u>Page</u>
1	FUEL SELECTION	1-1
	COAL	1-1
	Slagging	1-1
	Fouling	1-4
	Conditions to Which Metallic Components Will Be Exposed	1-4
	High-Temperature Corrosion of Convective Heat-Transfer	
	Surfaces Caused by Impurities in Combustion Gases	1-5
	Hot Corrosion in Gas Turbines	1-18
	Material Considerations	1-21
	Erosion	1-23
	Fuel Preparation	1-24
	FUEL OIL	1-25
	REFUSE	1-33
	WOOD	1-33
2	CYCLE ARRANGEMENT	2-1
	UTILIZATION OF AVAILABLE ENERGY AT HIGH TEMPERATURES	2-1
	STRAIGHT COMBUSTION--NO COGENERATION	2-10
	No Air Preheat	2-10
	Low-Temperature Air Preheat	2-11
	High-Temperature Air Preheat	2-12
	Very High-Temperature Air Preheat	2-13
	COGENERATION	2-14
	INFLUENCE OF SIZE ON FURNACE DESIGN AND MODULAR CONCEPT	2-16
3	HEAT-TRANSFER COEFFICIENTS	3-1
4	COST	4-1
5	SUMMARY	5-1
6	REFERENCES	6-1

LIST OF ILLUSTRATIONS

<u>Figure</u>		<u>Page</u>
1	Influence of Percentage of Basic Constituents in the Ash on Ash Softening Temperatures Under Reducing Conditions for Different Ranks of Coal	1-2
2	Relative Corrosion Rates of Steam Generator Steel as a Function of Temperature	1-7
3	Melting Points of Alkali-Sulfur Compounds as a Function of Sulfur Content of the Fuel	1-8
4	Melting Points of Alkali-Sulfur Compounds as a Function of SO ₂ Concentration in the Flue Gas	1-9
5	Effect of SO ₂ Levels on the SO ₂ Concentration Above Fe ₂ O ₃ Surfaces at 1100°F	1-11
6	SO ₂ Profiles Above Mixtures Containing Fe ₂ O ₃ at 1100°F in 6-Percent SO ₂	1-12
7	Effect of Flow Velocity on SO ₂ Gradient Above Fe ₂ O ₃	1-13
8	Reaction Sequences of Iron Oxides and Alkalies With Sulfur Oxides	1-15
9	Alkali Sulfate Corrosion	1-16
10	Coal- and Solids-Handling Systems for a Fluidized Bed Steam Generator	1-26
11	Coal- and Solids-Handling Systems for Stoker-Fired and Pulverized-Coal-Fired Steam Generators	1-27
12	Flow Diagram--Plant Flow System for Georgetown University	1-28
13	Flow Diagram--Solids Removal System for Georgetown University	1-29
14	Coal- and Solids-Handling Systems for a Gas Turbine Coal-Fired Heat Exchanger Modified for a Stirling Engine	1-30
15	Schematic of Fluidized Bed Gasifier Applied to a Steam Generator as a Retrofit	1-32

LIST OF ILLUSTRATIONS (Cont)

<u>Figure</u>		<u>Page</u>
16	T-H Diagram for Pulverized-Coal Combustor and Stirling Engine Convective Heat Exchanger, Utilizing Air Preheat and Requiring 250-Percent Excess Air	2-3
17	T-H Diagram for Pulverized-Coal Combustor and Stirling Engine Convective Heat Exchanger, Utilizing Air Preheat and High-Temperature Cogeneration With 15-Percent Excess Air	2-4
18	T-H Diagram for Fluidized Bed Combustor and Stirling Engine Heat Exchanger, Utilizing Air Preheat and Cogeneration at 15-Percent Excess Air	2-5
19	T-H Diagram for Fluidized Bed Combustor and Stirling Engine Heat Exchanger, Utilizing Air Preheat at 15-Percent Excess Air	2-6
20	T-H Diagram for a Conventional Steam Generator Firing Wood or Coal	2-7
21	T-H Diagram for a Fluidized Bed Steam Generator Firing Wood or Coal	2-8
22	T-H Diagram for a Fluidized Bed Steam Generator With Air Preheat Firing Dried Wood	2-9
23	Once-Through Combustor--No Air Preheat	2-10
24	Once-Through Combustor--Preheat Limited to 500 to 600°F	2-11
25	Once-Through Combustor--Preheat Restricted to 1000°F	2-12
26	Once-Through Combustor--Preheat to 1300°F	2-13
27	Cogeneration Using an Economizer	2-14
28	Alternative Cogeneration Application	2-15
29	Approximate Cost of Combustion/kW	4-7
30	Approximate Distribution of Cost of Combustion System/kW as a Function of Capacity	4-8

LIST OF TABLES

<u>Table</u>		<u>Page</u>
1	Furnace Capacity and Its Influence on Furnace Configuration	2-17
2	Cost Data for Combustor Systems	4-3
3	Relative Cost of Steam Generator Components	4-5
4	Cost Comparison of Tube Materials	4-6
5	Combustion Systems Summary Chart	5-2

Section 1
FUEL SELECTION

COAL

The factors influencing coal selection and operating conditions of the Stirling Engine heat source are:

- Slagging
- Fouling
- Corrosion
- Erosion
- Fuel preparation.

SlaggingPulverized-Coal-Fired Dry Bottom Combustor

The softening temperature (at least) and preferably the initial deformation temperature of the ash should be greater than the flue gas exit temperature to ensure that dry ash enters the convective heat-transfer passes. Thus the maximum flue gas temperature should be between 1950 and 2400°F.

Lignites. The maximum flue gas temperature should be about 1900°F.

Subbituminous. Flue gas temperature should fall between 1900 and 2300°F, depending upon the sodium, potassium, calcium, and magnesium present. The lower temperature is limited by high-alkali coals and coals with a percentage of basic constituents between 30 and 70 (Figure 1).^{1*}

*Numbers designate references appearing in Section 6.

Figure 1 Influence of Percentage of Basic Constituents in the Ash on Ash-Softening Temperatures Under Reducing Conditions for Different Ranks of Coal

Bituminous. Fuel selection depends upon iron concentration and therefore sulfur levels. At 40 percent iron in the ash, the flue gas temperature should be limited to 1900°F. At 20 percent iron, the flue gas temperature should be limited to about 2250°F, unless pyrites are present as pure specie separate from other coals. At 10 percent iron, the flue gas temperature may approach 2400 to 2500°F unless the pyrite is present as a pure specie.

Anthracite. This fuel has limited economic value. The same rules as bituminous apply, however.

Cyclone Furnace

Lignites, subbituminous, and bituminous ash fusion temperatures must not exceed 2400°F; otherwise, slag will not flow freely through the tap.

Fluidized Bed Combustor

The maximum bed temperature is limited by sulfur capture to 1750°F if emissions are a governing criterion. At 1750°F and above, slight surface melts may occur as a result of impurities in the ash, causing agglomeration and, ultimately, clinkering.

Stoker Combustor

High-sulfur washable coals can be prone to clinker formation. The same rules regarding mineral content of pulverized coal apply to coal prepared for stoker firing.

FoulingStokers, Cyclones, and Pulverized-Coal Combustors

Lignites and Subbituminous Lignite. The alkali concentration of the ash should not exceed 7 percent. Between 7 and 4 percent, the fuel can be troublesome and cause moderate maintenance problems. Below 4 percent the fuel should be trouble-free.

Bituminous. Alkali concentration should be limited to less than 0.4 percent ($\text{Na}_2\text{O} + \text{K}_2\text{O}$) on a dry-coal basis. Chlorine should be less than 0.3 percent on a dry-coal basis.²

Fluidized Bed Combustors

Theoretically, fluidized bed combustors should be trouble-free, with the possible exception of coals containing large quantities of sodium chloride (certain Illinois bituminous). Operation with high-alkali fuels has not yet been tried.

Conditions to Which Metallic Components Will Be Exposed

In cyclone, pulverized-coal, or stoker-fired combustors, the metal surfaces will be exposed to an oxidizing flue gas laden with impurities at gas temperatures of 350 to 2300°F. The different regimes that describe the operating temperatures to which the metal surfaces will be exposed include:

- Stirling Engine heat-transfer loop--either helium loop or heat pipe--metal temperatures of 1400 to 1550°F
- Low-temperature air preheater--350 to 750°F

- Intermediate-temperature air preheater--750 to 1050°F
- High-temperature air preheater for high-efficiency operation--1050 to 1300°F.

We assume that cogeneration will not be employed; therefore, all furnace walls, ducting, etc., will be refractory lined rather than water cooled.

In the case of the fluidized bed application, a portion of the heat-transfer surface will be immersed in a bed of agitated limestone at 1700°F. Deposit formation will be minimized by the action of the bed against the tube surface. The local environment on a microscale may be oxidizing, reducing, or cyclic because of the presence of carbon in the bed (about 5 percent of the bed material is coal at all times). At temperatures from 1700 to 350°F, the surfaces will be subjected to gases laden with innocuous ash. Presumably, the troublesome impurities will be contained by the ash or limestone retained in the bed. Several regimes of metal operation exist:

- Immersed heat transfer surface--extended helium loop or heat pipe at 1400 to 1550°F
- Tube bundle support material at 1700°F
- Distributor plate at 600, 1000, or 1300°F, depending on the air preheater design and cycle efficiency selected.
- Air preheater surface at 350 to 600°F, 350 to 1000°F, or 350 to 1300°F, depending upon the cycle efficiency selected.

High-Temperature Corrosion of Convective Heat-Transfer Surfaces Caused by Impurities in Combustion Gases

High-temperature fireside chemical attack in conventional coal-fired steam generators may be caused by gas-phase oxidation or liquid-phase corrosion. Gas-phase oxidation is seldom a problem if tube and support materials are selected

with due regard for their oxidation resistance at the operating temperatures be encountered and if the protective oxide layer that builds up on the surface is not periodically removed by spalling, flaking, or chemical reaction with its environment. Liquid-phase corrosion, on the other hand, usually results in accelerated attack and rapid metal wastage. This type of corrosion is generally accepted as being caused by the presence of a liquid phase on the surface of the metal beneath an overlying ash deposit. If deposits do not form, there can be no liquid-phase attack.

Experience and extensive research have established that the metal temperature and the gas temperature in the area adjacent to the metal are interrelated in establishing a corrosion rate when corrosive agents are present. For most conventional applications, this relationship can be corrected to the average gas temperature.

Alkali-Sulfur Compounds

All liquid-phase attack thus far observed in coal-fired steam generators has been associated with the presence of compounds of alkalies and sulfur. Temperature ranges in which accelerated corrosion takes place can be related to the melting temperatures of these various compounds. Thus the rate of corrosion resulting from alkali-sulfur compounds as a function of temperature is typically as shown in Figure 2. The melting points of the alkali-sulfur compounds as a function of sulfur content appear in Figures 3 and 4.

Pyrosulfates. In the temperature range from about 550 to 900°F, corrosion might be attributed to the pyrosulfates, $\text{Na}_2\text{S}_2\text{O}_7$, and $\text{K}_2\text{S}_2\text{O}_7$. The specific temperature range for each pyrosulfate depends on whether oxidizing or reducing

Figure 2 Relative Corrosion Rates of Steam Generator Steel as a Function of Temperature

Figure 3 Melting Points of Alkali-Sulfur Compounds as a Function of Sulfur Content of the Fuel

Figure 4 Melting Points of Alkali-Sulfur Compounds as a Function of SO₃ Concentration in the Flue Gas

conditions prevail. The existence of pyrosulfates also depends upon any SO_2 proximal to the alkali sulfates from which they form. The first investigators found that at least 250 ppm SO_2 was necessary for the formation of the pyrosulfates. The SO_2 level leaving the combustion zone is 2000 to 3000 ppm for most coals and fuel oil, and the SO_3 level is about 30 to 40 ppm. Under these conditions, the formation of pyrosulfates should not occur.³

Subsequent studies with a microprobe at Battelle Columbus Laboratories revealed that the SO_3 level at the tube surface may be 30 times that in the bulk gas stream. These researchers found that the concentration of SO_3 increased considerably with a rise in the level of SO_2 (Figure 5). At the higher levels, the rate of change in concentration with distance from the tube or concentration gradient increased.⁴

Figure 6 shows that a mixture of equal percentages of Fe_2O_3 and Fe_3O_4 is less efficient as a catalyst than Fe_2O_3 alone, while an oxidized iron surface is about comparable to the mixture. When Na_2SO_3 is present, the SO_3 near the surface is low because the SO_3 is being captured to form trisulfate, as is the case when NaOH is available.

An increase in gas velocity from 1.2 to 12 ft/s decreases the SO_3 concentration by 90 percent, showing that the microprobe gas samples were measuring SO_3 levels in the boundary layer (Figure 7).

Recent data from Coats, Dear, and Penfold clearly show that the minimum SO_3 level for the formation of $\text{K}_2\text{S}_2\text{O}_7$ is 150 ppm and for $\text{Na}_2\text{S}_2\text{O}_7$ is 2000 ppm,

Figure 5 Effect of SO_2 Levels on the SO_3 Concentration Above Fe_2O_3 Surfaces at 1100°F

Figure 6 SO_3 Profiles Above Mixtures Containing Fe_2O_3 at 1100°F in 6-Percent SO_2

Figure 7 Effect of Flow Velocity on SO_2 Gradient Above Fe_2O_3

at 765 and 745°F respectively (Figure 4).⁵ Since the SO_3 level increases as the temperature rises, there is an upper limit to the formation of the pyrosulfates of about 900°F.

Control of pyrosulfate formation can be achieved by ensuring an oxidizing environment, avoiding flame impingement on furnace walls, and limiting alkali content in the fuel.

Alkali Iron Trisulfates. The alkali iron trisulfates $\text{Na}_2\text{Fe}(\text{SO}_4)_3$, have been mainly responsible for the severe loss of metal in superheaters and reheaters where the metal temperature is in the 1100 to 1300°F range. The following sequence of events is generally accepted as representing the actions occurring during the corrosion process (Figure 8). The block diagram (Figure 9) shows that the process is continuous because of its cyclic behavior.

- Sodium and potassium are volatilized, in part, from the mineral matter in the high-temperature flame, forming Na_2 and K_2O .
- During combustion, pyrites are dissociated thermally and, with the organic sulfur in the coal, react with oxygen, forming mostly SO_2 and some SO_3 .
- The Na_2O and K_2O then react with SO_3 , either in the gas stream or on the tube, to form sodium or potassium sulfate, a low-melting material that attracts other ash particles, eventually building up a moderately thick deposit on the tube.
- SO_2 present in the flue gas is catalytically oxidized to SO_3 on iron-oxide surfaces present as tube scale. SO_3 reaches near equilibrium concentrations in localized areas near the tube (≈ 1000 ppm).
- The alkali sulfates, iron oxides, and SO_3 then react to form the complex sulfates $2(\text{Na}_2 \text{ or } \text{K}_2) \text{Fe}(\text{SO}_4)_3$, which are molten at about 1100°F in a high- SO_3 atmosphere (≈ 1000 ppm).

Figure 8 Reaction Sequences of Iron Oxides and Alkalies With Sulfur Oxides

Figure 9 Alkali-Sulfate Corrosion

A difference of opinion exists about the source of Fe_2O_3 in the alkali-iron trisulfate and, therefore, the exact mechanism of the tube wastage. Some investigators believe the source of Fe_2O_3 to be the coal ash and have shown that migration of the complex sulfate does occur, resulting in tube wastage according to the following reaction:⁶

Others indicate that tube scale is a more likely form of Fe_2O_3 , resulting in a reaction where the complex sulfate is a product of the wastage reaction as follows:⁷

Alkali Sulfates. The third and highest temperature range is associated with the alkali sulfates Na_2SO_4 and K_2SO_4 , which are highly corrosive when molten. The temperature range extends from 1530°F--the minimum melting point for the system $\text{Na}_2\text{SO}_4 \cdot \text{K}_2\text{SO}_4$ --to about 1950°F--the melting point for pure K_2SO_4 . Alkali sulfate corrosion has not been a problem in steam generators because the temperatures at which it would occur are not reached by metal parts in steam generators.

Chlorine

The role of chlorine in the corrosive attack is still uncertain. Brown, DeLong, and Auld show that elemental chlorine attacks carbon steel rapidly above 400°F, while the HCl attack increases markedly above 895°F.⁸

On the other hand, Cutler, et al., indicated that chlorides will not condense as an independent phase in the deposit and that the HCl will have little effect on any metal surface it contacts.⁹ Furthermore, the absorption of gaseous chloride by the deposits will not enhance their long-term corrosiveness. They do recognize, however, that chlorides may deposit on tube surfaces because of a short residence time or poor combustion.

The presence of chlorides as an independent phase on a surface, by whatever mechanism it is introduced, may be taken as an indication of enhanced rates of attack on the metal. If the general atmosphere permeating the deposit is oxidizing, the chloride can play an active role in the corrosive process. If it is reducing, the atmosphere may cause enhanced rates of corrosion by destruction of any protective oxide layer, sulfidation of the metal, etc. The chloride itself may play little or no direct part.

Chlorine rarely exceeds 0.3 percent on a dry-coal basis in American fuels. It thus presents little problem in coal. Its contribution to corrosion while firing refuse is not as clear.

Hot Corrosion in Gas Turbines

Since the air preheater and Stirling Engine heat exchange surface in all but the fluidized bed applications will be subjected to environmental conditions similar to gas turbine operation (high metal temperatures and short residence time), it is worth noting hot corrosion in gas turbines.

Hot corrosion of superalloys in gas turbines has been reviewed by Stringer:¹⁰

Alkali sulfates deposit on the metal surface in the absence of a covering layer of ash. The protective oxide layer is removed by one of a number of mechanisms: a particularly interesting mechanism is the "salt-fluxing" model. In this, the sulfate at the protective oxide surface is displaced from stoichiometry, either in the more basic direction (by removal of sulfur) or in the more acid direction (by removal of the alkali oxide). The sulfur can be removed by reacting with the underlying metal; the sodium by reacting with (for example) molybdenum trioxide, to form the stable species Na_2MoO_4 .

The acid or basic salt is then capable of dissolving the protective oxide: in the former case the metal goes into solution as the cation, and in the latter as an anion complex (e.g., NiO_2). The iron migrates toward the outer surface of the molten sulfate and as it moves into regions where the sulfate is stoichiometric it becomes unstable, precipitating as the oxide. These precipitated particles do not form a protective scale. Again, this is only one mechanism, and several others are possible: in particular, sulfidation of the more reactive constituents of the alloys is of considerable importance.

As with the corrosion in boilers, this corrosion shows a "bell-shaped" dependence on temperature, but the threshold temperatures depend very much on the particular alloy, and indeed on the exact conditions. Typical values are 1450 to 1700°F (788 to 927°C) for the upper and lower limits.

Fluid Bed Combustion Corrosion

The chemical situation within the fluidized bed combustor is not easy to define. The bed is relatively cool, so that the release of volatile alkalis should be less; the acceptor reduces the concentration of sulfur oxides to 300 ppm or so. The alkali content is nevertheless in the range 1-10 Na, 0.1-1 ppm K, which is high by gas turbine standards, and would correspond to a dew point of at least 1900°F (1038°C), in terms of sodium sulfate condensation. Furthermore, the difference between the bed temperature and the metal temperature, while not as great as in the pf furnace, is nevertheless sufficient to cause salt condensation. Because the effective radiating surface is cooler, there will be a reduced tendency for the outer surface of the deposit to become sticky and capture fly ash, so the mechanism outlined above is unlikely to occur in the freeboard tubes in the fbc. Indeed, no ash build-up is normally observed.

The exhaust gases would appear to be likely to produce hot corrosion in gas-turbine-type superalloys: the likely used temperature falls within the range of maximum susceptibility, and the alkali content is certainly high. However, it is not altogether clear what form the alkali is in: on the basis of volatilization of constituents from the bed into the gas stream it cannot be present in the vapor phase, at least as sulfate. If the metal temperature is equal to the gas temperature, little condensation would be expected; cooled components would collect a deposit....The chlorine content of the combustion gases is in the range 10-100 ppm and this is very high ...[in relation to the sulfur present].

The situation of the...tubes within the bed is much more complex. The accumulation of alkali containing deposits on the tubes does not seem likely, since the mild abrasion of the bed should remove them continuously; there may be some condensation of alkali sulfates on to the relatively cool metal surface, although again it may be removed by abrasion. The accumulation of ash and the generation of the reactions proposed...[earlier] seem to be altogether impossible, however. It is thus unreasonable to compare the corrosion behavior with that of conventional superheaters.

The danger--if indeed there is a danger--will lie in the relatively low oxygen activity and high carbon activity in the bed. The bed is obviously not at equilibrium: the coal particles burn relatively slowly, although the overall carbon content of the bed at any time is quite small. There is an overall air excess on the order of 15% passing through the bed, but nevertheless when coal is charged the volatiles are driven off and some burn above the bed, apparently passing through the bed largely uncombusted. There is thus a reducing-to-oxidizing microenvironment around each burning coal particle, and a reducing-to oxidizing macroenvironment pulse associated with each charge of coal as the volatiles pass through the bed. There "reducing" means "relatively low oxygen activity." In addition, there are variations in chemical character within the bed. In spite of the rapid mixing, on average the region near the coal-charge points will be relatively reducing. Relatively stationary caps of ash will form on the upper surface of the tubes, [and at the roots of fins on extended heat transfer surface] giving a more reducing environment; bubbles will impinge on the lower side giving relatively oxidizing conditions. There is thus a possibility of accelerated attack of the carburization-sulfurization-oxidation type, and this may be enhanced by the presence of impurities in the bed.

It is important to emphasize that it cannot be concluded that the attack will take place, merely that if there is a high-temperature corrosion risk, it will be of this type rather than the more usual superheater fireside corrosion encountered in pf boilers.

The conclusions of this analysis are thus:

- There is little risk of fireside corrosion of freeboard tubes with metal surface temperatures below 1000°F.
- There is an as-yet unexplored possibility of corrosion of in-bed superheater tubes and support members, but this will not be of the conventional superheater fireside corrosion type.

Material Considerations

Material selection for the Stirling engine heat source is based on strength, cost, and ability to withstand fireside corrosion. The heat exchange surface was selected on the basis of continuous operation at 1550°F with occasional excursions to 1650°F in upset conditions. Both of these temperatures exceed the 1500°F maximum allowable stress in the ASME Code, Section VIII-Division 1. The heat source combustor is not presently under the jurisdiction of the ASME Code; thus a Code Stamp is not required. If, however, a Code Stamp is desired, a Code Case will have to be submitted during the design phase.

We must also point out that materials are being recommended for a regime of operation previously described on a conceptual level. The selection is subject to change as details in design are developed.

Type 310 stainless steel is recommended for the finned tube heat exchanger with sodium as a coolant at 8 lb/in²a and immersion in the fluidized bed; 800H is recommended for the 350 lb/in² helium loop finned or bare tube bundle immersed in the fluidized bed.

Carbon and low-alloy steels as well as the stainless steels were dismissed for this application because of inadequate corrosion resistance and strength.

The austenitic stainless steels are worthy of consideration at the lower pressure because they have adequate strength and creep-rupture properties. To date, Type 310 stainless steel in particular has demonstrated adequate air oxidation at temperatures in related industries and in the fluidized bed steam generator application. The iron-nickel-based alloys also have adequate strength and air oxidation resistance for the generating conditions but will experience sulfidation, depending on the percentage of nickel present in the alloy. The optimum alloy for high-pressure application would be 800H because of its high strength, good creep-rupture properties, and relatively low nickel level of 32 percent. The sulfidation is not nearly as great as the nickel-based alloys with typical nickel concentrations of 60 to 70 percent.

The cobalt alloys have adequate strength and air oxidation resistance. However, their very high cost and the shortage of cobalt has ruled them out as candidate materials.

The two most favorable, from a cost and fabrication standpoint, are Type 310 stainless steel and 800H. These alloys have the required strength and creep-rupture properties. Type 310 stainless steel has slightly better corrosion resistance than 800H, but 800H has better strength and creep-rupture properties.

The same material selection would apply to the convective heat-transfer surface immersed in the convective passes of the stoker, pulverized-coal, or cyclone combustor. For these applications the heat exchanger appears to be operating in a corrosion-free window between the trisulfate molten salt attack and the sulfate molten salt attack (See Figure 2). If corrosion is encountered neither Type 310

stainless steel or 800H material will withstand the corrosive attack, nor will any other material selected withstand the trisulfate attack. The best defense against the high-temperature molten salt attack is fuel selection. This is accomplished by restricting sulfur, chlorine, and alkali concentrations.

Providing the heat exchanger is not subjected to alkali-trisulfate attack, Type 310 stainless steel would be adequate for the air preheater in the 1000 to 1300°F range.

In the 750 to 1000°F range, 1-1/4%Cr-1/2%Mo or 2-1/4%Cr-1%Mo would be recommended for the air preheater, depending on the specific design conditions. Below 750°F carbon steel could be used for tubular materials of construction.

Carbon steel may be used as a material of construction for distribution plates operating below 600°F air preheat. Between 600 and 1000°F Type 304 stainless steel would most likely be used. Above 1000°F the distribution plates must be constructed of refractory.

Erosion

The gas velocities in the convective passes are generally limited to 70 ft/s for low-ash fuels and 30 ft/s for high-ash coals. For conceptual design purposes, a gas velocity of 50 ft/s between tubes may be considered as a reasonable value.

Fuel Preparation

Illustrated below are the fuel feed sizes required for the various combustion systems.

Combustion System	Fuel Feed Size
Cyclone*	1-1/2 in. x 0 in.
Stoker*	1-1/4 in. - 3/4 in. x 0 in. Maximum 40% <1/4 in.
Pulverized coal†	70% through 200 mesh
Fluidized bed combustion‡	1/4 in. x 0 in.

*No moisture limitation.

†Drying takes place in pulverizer.

‡Moisture limited to 3 percent surface moisture. Over-bed feeding, no limitation. The latter is presently being developed.

Pulverized coal requires the use of a mill. In the 500- to 3000-hp range, mills must be custom built or an oversized unit must be selected. The system is very expensive for the smallest size range. The fines output from the latter is questionable at part load. Furthermore, a bin system is required for storage and transport of the pulverized coal. When the combustor requires only one burner, the fuel preparation and firing systems for the small size range require a close examination to determine the availability and cost of a pulverizer and feeders. Crushers, although apparently available for remaining combustor technologies, should be investigated to be certain of their availability.

There is no doubt that the coal preparation facility (i.e., crusher, storage, hopper, feeder, and transport lines) will be a major cost item in the overall system, particularly at the lower size range. Costs and state-of-the-art technology should be developed by means of a conceptual design study. Typical coal-receiving, -storage, and -preparation systems appear in Figures 10 through 14. Figures 10 and 11 illustrate the conceptual coal- and solids-handling systems for a 150,000 lb/h of steam stoker, pulverized-coal, or fluidized bed steam generator. The cost of the system is summarized in Section 4.

Figures 12 and 13 are the coal-handling and solids-flow systems process flow diagrams for the Georgetown fluidized bed boiler installed at Georgetown University. The cost of the system is shown in Section 4. The data at this point are incomplete.

Figure 14 is the conceptual process flow diagram for a 200-kW gas turbine coal-fired heater modified to suit a Stirling Engine. Budget costs also appear in Section 4.

FUEL OIL (Residual Oil)

Modes of combustion include suspension combustion, fluidized bed combustion, and gasification. Suspension combustion is presently available. Fluidized bed combustion has received little attention. Fluidized bed gasification is in the developmental stage.

Operation of tube surface at 1500°F places severe restrictions on the vanadium and sodium concentration in the oil. If allowed to happen, corrosion by

100 Car Shaker W/Hoist
101 Car Positioner
102 Dust Suppression Sys.
103 Receiving Hopper W/Feeder
104 Sump Pump
105 Belt Conveyor
106 Belt Scale
107 Belt Magnet
108 Metal Detector
109 Radial Stacker

110 Bulldozer
111 Reclaim Hopper W/Feeder
112 Belt Conveyor
113 Belt Scale
114 Belt Magnet
115 Metal Detector
116 Sump Pump
117 Crusher
118 Screen
119 Pneumatic Transport Sys.

120 30-Day Cap. Prepared Coal Silo
121 Weigh Belt Feeder
122 Recirculating Conveyor
123 Prepared Coal Silo
124 Screw Feeder
125 Pneumatic Transport Sys.
126 Pneumatic Unloading Sys.
127 Limestone Storage Silo
128 Screw Feeder

129 Pneumatic Transport Sys.
130 FBC Package
131 Pneumatic Ash Transport Sys.
132 Baghouse Dust Collector
133 Ash Silo
134 Ash Conditioner and Loadout Sys.

Figure 10 Coal- and Solids-Handling Systems for a Fluidized Bed Steam Generator

100 Car Shaker W/Hoist	111 Reclaim Hopper W/Feeder	121 Weigh Belt Feeder
101 Car Positioner	112 Belt Conveyor	122 Recirculating Conveyor
102 Dust Suppression Sys.	113 Belt Scale	123 Boiler
103 Receiving Hopper W/Feeder	114 Belt Magnet	124 Pneumatic Ash Transport Sys.
104 Sump Pump	115 Metal Detector	125 Coal Bunker
105 Belt Conveyor	116 Sump Pump	126 Double Alkali Flue Gas Cleanup Sys.
106 Belt Scale	117 Crusher	127 Ash Silo
107 Belt Magnet	118 Screen	128 Ash Conditioner and Loadout Sys.
108 Metal Detector	119 Pneumatic Transport Sys.	
109 Radial Stacker	120 30-Day Cap. Prepared Coal Silo	
110 Bulldozer		

Figure 11 Coal- and Solids-Handling Systems for Stoker-Fired and Pulverized-Coal-Fired Steam Generators

Figure 12 Flow Diagram--Plant Flow System for Georgetown University

Figure 13 Flow Diagram--Solids Removal System for Georgetown University

Figure 14 Coal- and Solids-Handling Systems for a Gas Turbine Coal-Fired Heat Exchanger Modified for a Stirling Engine

vanadium is catastrophic. No metallurgical solution has been found to satisfactorily resist attack. The fuel concentrations of sodium, lead, vanadium, and calcium must be restricted as follows:

<u>Concentrations</u>	<u>Maximum ppm</u>
Sodium plus potassium	1
Lead	1
Vanadium--untreated	15
Vanadium--treated (3:1 weight ratio of Mg/V)	500
Calcium	10

Since residual oils are frequently contaminated with sodium as a result of the transport of the crude by sea, the oil would have to be washed to remove the sodium. Fuels high in vanadium require the use of an additive.

Fluidized bed gasification has been used to control sulfur emissions. Under reducing conditions about 80 percent of the vanadium has been found to deposit on the bed surface material. The oxide state of the remaining vanadium is not certain. Lower oxides of vanadium (i.e., V_2O_3 and V_2O_4) have very high melting points in excess of 3000°F and thus do not cause corrosion or fouling. V_2O_5 , on the other hand, melts at 1240°F and forms eutectics with Na_2O and Na_2SO_4 at temperatures as low as 950°F . The behavior of vanadium in a fluidized bed combustor is uncertain. Figure 15 is a schematic of the application of the fluidized bed gasifier to a steam generator as a retrofit. If applied to

Figure 15 Schematic of Fluidized Bed Gasifier Applied to a Steam Generator as a Retrofit

the Stirling cycle, the steam generator would be replaced with a gas-fired combustor and helium cycle heat-exchange loop.

REFUSE

Municipal refuse can be burned in a fluidized bed or on a grate. It frequently contains zincs, leads, and chlorides in addition to the impurities normally found in coal. These impurities have been known to lower the threshold of corrosion to as low as 750°F. The mechanism and contaminants responsible for the corrosion are not clearly understood. Refuse-derived fuels may be void of these problems, but this is not presently certain. Potentially, refuse may contribute approximately 10 percent to the energy market. Therefore, refuse would appear to be a specialized market somewhat farther in the future than coal or residual oil.

WOOD

For the purpose of combustion, wood may be treated as a refuse fuel. The bulk of the wood, about 90 to 95 percent, is processed into a consumable product. About 5 to 10 percent must be disposed of by mills and paper plants as waste. Under these circumstances wood is a refuse material creating disposal problems. Wood refuse deteriorates slowly, causes spontaneous combustion at landfills, and consumes vast quantities of landfill real estate. It has good heating value--about 7000 Btu/h (dry). It is low in nitrogen and ash and can be burned in a suspension, on a grate, in cyclones, and in fluidized beds. Fuel preparation has been one of the main sources of problems when wood is treated as a fuel.

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Wood contains no corroding impurities. Its slagging and fouling characteristics are uncertain. Wood refuse, apparently a specialty market for the wood and paper industry, requires process steam, electricity, and a means of disposing of wood waste.

Section 2

CYCLE ARRANGEMENT

Several process arrangements may be used for the Stirling Engine heat source. Each arrangement appears to apply to a different type of combustion process (e.g., stoker, pulverized-coal).

UTILIZATION OF AVAILABLE ENERGY AT HIGH TEMPERATURES

Examination of the various cycle options begins by subdividing thermal energy use into two categories: thermal-electrical production and cogeneration. In the first case, those processes that are inefficient simply discharge unavailable energy to the atmosphere as waste heat. In the second case, an attempt is made to use this energy as process heat at the expense of some additional capital investments. Inefficient processes have high process heat to electrical production rates; efficient processes have low process heat utilization to electrical production.

In Section 1 we established that the furnace exit temperature of all combustion processes (excluding the fluidized bed combustor) was limited by the impurities in the coal to between 1950 and 2300°F. This limitation results in a need to dissipate the available sensible heat in the flue gas between the combustor furnace exit temperature and the adiabatic flame temperature at stoichiometric conditions through cogeneration heat sink or by dilution with excess air. In the latter case, a portion of this sensible heat is made available to the Stirling Engine heat exchanger, while the remaining portion is lost up the

stack, as illustrated on the Temperature-Enthalpy (T-H) diagram in Figures 16 and 17. The fluidized bed is a well-stirred reactor. Consequently, all the sensible heat at stoichiometric conditions and above bed temperature is available to the Stirling Engine. Since no additional air is required for dilution, the process is somewhat more efficient. This efficiency is illustrated in the T-H diagrams in Figures 18 and 19. Figures 20, 21, and 22, T-H diagrams for conventional steam generators, fluidized bed steam generators, and cycles using air preheat, have been included for comparison.

Construction of the combustor may also have an impact on the use of the highest grade source of available energy. The furnace may be constructed as an adiabatic or nonadiabatic combustor. The former requires the use of refractory for insulation and is slightly larger to accommodate the excess air required for temperature control. The latter uses air or water-cooled furnace walls and minimal excess air. The adiabatic combustor generally requires at least partial field construction to avoid destruction of the refractory during shipment. The nonadiabatic combustor may be completely shop assembled, thereby realizing a reduction in cost and an improvement in quality control. The additional cost of a pump to move the coolant and piping, etc., to contain and process the coolant may negate some of the advantage of shop fabrication. Size of the unit also has a bearing on whether to choose shop fabrication or field construction. There are some trade-offs of fuel cost, energy dissipation, and capital investment that are dependent on size and design and can be evaluated only by going through

Figure 16 T-H Diagram for Pulverized-Coal Combustor and Stirling Engine Convective Heat Exchanger, Utilizing Air Preheat and Requiring 250-Percent Excess Air

Figure 17 T-H Diagram for Pulverized-Coal Combustor and Stirling Engine Convective Heat Exchanger. Utilizing Air Preheat and High-Temperature Cogeneration With 15-Percent Excess Air

Figure 18 T-H Diagram for Fluidized Bed Combustor and Stirling Engine Heat Exchanger, Utilizing Air Preheat and Cogeneration at 15-Percent Excess Air

Figure 19 T-H Diagram for Fluidized Bed Combustor and Stirling Engine Heat Exchanger, Utilizing Air Preheat at 15-Percent Excess Air.

Figure 20 T-H Diagram for a Conventional Steam Generator Firing Wood or Coal

Figure 21 T-H Diagram for a Fluidized Bed Steam Generator Firing Wood or Coal

Figure 22 T-H Diagram for a Fluidized Bed Steam Generator With Air Preheat Firing Dried Wood

an exercise involving detailed design and cost evaluation. Tentatively, we recommend using refractory construction based on the following criteria:

- More of the units will probably be in the large range; therefore, at least partial field construction will be in order.
- Highest efficiency will be achieved.
- Wall penetration of heat-transfer surface may be the simplest.
- Expansion problems may be simplified.
- The additional prime movers and auxiliary circuitry for the coolant are eliminated.

STRAIGHT COMBUSTION--NO COGENERATION

Excluding the possibility of cogeneration, there are several possible cycle arrangements, all dependent upon disposing of the available energy in the flue gas below the operating temperature of the Stirling Engine heat exchanger.

No Air Preheat

The simplest case is a once-through combustor using no air preheat.

Figure 23 Once-Through Combustor--No Air Preheat

The flue gas is discharged at about 1750°F. Slightly lower temperatures may be achieved with combustors other than the fluidized bed combustor. Cycle efficiency is poor ($\eta = 43$ percent excluding nonadiabatic furnace losses). The capital cost of the combustor is the lowest of all. The only surface exposed to severe operating conditions in the Stirling Engine is the heat exchanger surface. This cycle arrangement applies to the stoker, where preheated air is undesirable. Stoker manufacturers prefer no air preheat. If air preheat must be used, the temperature is limited to about 200°F to avoid clinkers and damage to the grates.

Low-Temperature Air Preheat

The second case is one in which the air preheat is limited to 500 to 600°F. The flue gas without the benefit of further cooling by cogeneration drops to about 1300°F. Cycle efficiency is still poor, about $\eta = 62$ percent.

Figure 24 Once-Through Combustor--Preheat Limited to 500 to 600°F

Carbon steel can be used for the materials of construction for a fluidized bed distributor plate. Unfortunately, without the benefit of a cogenerator cooler at point (1) upstream of the air preheater, the flue gas temperature

becomes excessively high and, consequently, at least a portion of the air preheater metal temperature approaches 1000°F. Normally, the air preheater is designed for a maximum metal temperature of 750°F. At temperatures above 750°F, carbon steel can no longer be used, and one must resort to stainless alloys with a corresponding incremental jump in cost. One alternative would be to introduce air upstream of the air preheater as a means of reducing its temperature to a more desirable level. This would require an additional prime mover and a slight compromise in efficiency. The trade-offs, once again, require a more detailed evaluation.

This cycle arrangement applies equally well to all processes except the stoker.

High-Temperature Air Preheat

A third case restricts the air preheat to 1000°F, which is the upper limit for the use of metals as materials of construction in the fluidized bed combustor. The flue gas exit temperature, not making use of cogeneration or of flue gas-air dilution, would be about 1000°F. Cycle efficiency would be $\eta = 64$ percent.

Figure 25 Once-Through Combustor--Preheat Restricted to 1000°F

The air preheater metal surface temperature at the flue gas inlet would be about 1375°F. This temperature far exceeds the present state of the art. Superalloys would have to be used, and there would have to be insurance against the formation of alkali trisulfate from coal or vanadium pentoxide condensation from oil. This cycle arrangement is applicable to fluidized bed combustion. Pulverized-coal and cyclone combustors would have to be modified somewhat with regard to the introduction of secondary air.

Very High-Temperature Air Preheat

The fourth cycle arrangement would permit air preheat to reach 1300°F. The flue gas exit temperature would approach 643°F, which far exceeds the lower limit of 350°F established by the dewpoint of HCl or H₂SO₄. The cycle efficiency is 74.4 percent, neglecting heat losses caused by temperature limitations imposed on the furnace.

Figure 26 Once-Through Combustor--Preheat to 1300°F

The upper limit for air passing through refractory constructed distributor plates in catalytic crackers is 1300°F. At 1300°F air preheat temperatures, the tube material temperature of the air preheater approaches 1528°F. The materials of construction of the hottest surface would be the same as those of the Stirling Engine heat exchanger. This cycle arrangement may be restricted to fluidized bed operation, depending upon the designer's capability of dealing with extremely hot secondary air in the other applications.

COGENERATION

Cogeneration can be applied to any one of the previously mentioned means of recovering low-grade heat by mounting an economizer (convection heat-transfer surface) between the combustor and the air preheater. By so doing the flue gas exit temperature can be reduced to 350°F, which is the lowest limit still capable of avoiding dewpoint condensation of HCl or H₂SO₄. This arrangement gives the maximum cycle efficiency of 85 percent.

Figure 27 Cogeneration Using an Economizer

An alternative application exists with cogeneration, which is dependent upon the demands for process steam and electrical energy. For a case in which the demands for process heat far exceed the electrical requirements, the Stirling Engine heat exchanger loop might be considered an appendage to a process heater.

Figure 28 Alternative Cogeneration Application

In comparing the various alternatives available, it should be noted that cogeneration will require about 40 percent more fuel input/kW output than the most efficient air preheat cycle so that the additional process heat required to make the cogeneration process efficient can be generated. The most efficient air preheat cycle is advanced state of the art, requiring the use of superalloys under untried and severe operating conditions. The state of the art limits air preheater metal temperatures to about 750°F, giving an efficiency of only 64 percent. One option not previously discussed would use cogeneration to recover the

waste heat between 350 and 1000°F, which in reality is a modified version of the first cogeneration process. The fuel-to-electrical output would remain high.

In all the previous discussions, the combustor was assumed to be a fluidized bed using a helium loop or sodium heat pipe. If pulverized-coal or cyclone burner applications are considered, the combustor temperature may have to drop slightly to about 1400°F, providing a slight improvement in cycle efficiency.

INFLUENCE OF SIZE ON FURNACE DESIGN AND MODULAR CONCEPT

In the 500 to 3000 hp range, the furnace capacity will have a strong influence on the furnace configuration and whether or not the furnace walls will be cooled. The surface-to-volume ratio changes drastically in the lower size range. Therefore, the percentage of heat absorbed by the wall increases rapidly as the size is reduced and the heat-transfer coefficient is enhanced. This is illustrated in Table 1, where it is shown that, in the case of a fluidized bed 4 ft deep operating at 3 ft/s with a heat-transfer coefficient of 100 Btu/h·ft²·°F, 15 percent of the heat released is absorbed by the wall at 3000 hp and 30 percent at 500 hp. The point at which the walls must be lined appears to be about 2000 hp.

The study was also made for a suspension-fired combustor by simply assuming a heat-transfer coefficient (including convection and radiant heat-transfer coefficients) of 10 and a mean wall temperature of 270°F. We learned that although the maximum furnace velocity would be in the vicinity of 50 ft/s, a velocity of

Table 1 Furnace Capacity and Its Influence on Furnace Configuration

<u>Assumption</u>	
Coal heating value	13,000 Btu/lb
Flue gas/fuel ratio	≈12:1
Flue gas temperature	1750°F
Specific volume	55 ft ³ /lb
Stirling Engine efficiency	40 percent
Combustion efficiency (Dry gas loss, carbon loss, moisture loss, etc.)	85 percent
No cogeneration	
Bed depth	4 ft
Residence time (Case 1)	0.25 seconds
<u>HP</u>	
	<u>500</u> <u>1000</u> <u>2000</u> <u>3000</u>
Fuel flow (lb/h)	288 576 1,152 1,728
Flue gas flow (lb/h)	3,476 6,912 13,824 10,368
Q required (Heat Release x 10 ⁶ Btu/h)	3.744 7.48 14.9 19.09
A (10ft/s) Case 1*†	2.64 5.28 10.56 15.84
A (3 ft/s) Case 2 ^s	1.62 x 1.62 2.29 x 2.29 3.2 x 3.2 3.9 x 3.9
	8.8 17.6 35.2 52.8
	2.9 x 2.9 4.19 x 4.19 5.9 x 5.9 (7.2 x 7.2)
Heat absorbed through wall (Case 1) x 10 ⁶ Btu/h	0.324 0.458 0.640 0.780
Heat absorbed through wall (Case 2) x 10 ⁶ Btu/h	1.160 1.676 2.36 2.88
% of heat released	8.6 6.12 4.2 4.0
% of heat released	30.9 22.4 15.8 15.0

*Conventional combustion (wall coefficient about 10 Btu/h·ft²·°F).

†Although maximum gas velocity is usually 50 ft/s, the velocity had to be reduced in Case 1 to 10 ft/s or less to maintain a reasonable length-to-diameter ratio.

Fluid bed combustion (wall coefficient about 100 Btu/h·ft²·°F).

10 ft/s was selected to maintain a reasonable length-to-diameter ratio over the entire size range.

A more thorough study should be made using more accurate values. What has been learned is that the 3000 hp unit is not simply a modular multiple of 500. It may be constructed quite differently.

Section 3

HEAT-TRANSFER COEFFICIENTS

For conventional bare-tube convective heat-transfer, neglecting nonluminous radiation that may occur around 2000 to 2300°F, the gas-side heat-transfer coefficient in cross flow at 50 ft/s may be calculated using the following correlation:

$$\frac{hD_o}{k} = 0.0287 \left(\frac{GD_o}{\mu} \right)^{0.61} \left(\frac{C_p \mu}{k} \right)^{0.33}$$

At 50 ft/s, the flue gas duct cross section may be quite small, making it almost essential to insert a tube bundle parallel to the direction of flow. The heat-transfer correlation that can be used in this case is:

$$\frac{hD_o}{k} = 0.023 \left(\frac{GD_e}{\mu} \right)^{0.8} \left(\frac{C_p \mu}{k} \right)^{0.4} \left(\frac{T_b}{T_f} \right)^{0.8}$$

$$De = 4 \left[\frac{(L_1 L_2 - 0.785 D_o^2)}{D_o} \right]$$

Experience with finned tubes in fluidized bed combustors is limited to the following investigations:

- Dr. J. Chen - Lehigh University
- Dr. W. E. Genetti - Montana State University

- Dr. Golan - Exxon
- Dr. Moskowitz - Curtiss Wright
- Dr. V. Zakkay - New York University
- J. C. Petrie, W. A. Freeby, J. A. Buckham - Idaho Nuclear Corporation
- L. R Glicksman - Massachusetts Institute of Technology.

Papers describing their work were submitted earlier and are identified in References 14 through 21.

The desired duty per tube is 850,000 Btu/h using 5-in. O.D. tubes that are 10 ft long. The mean temperature difference for fluidized bed combustors is about 250°F maximum. The mean temperature difference may increase to 364°F in pulverized-coal-fired combustors or stokers when eastern coals of good quality are fired. Slightly higher values may be achieved with a cyclone. To achieve the desired duty at the mean temperature differences outlined, heat-transfer coefficients of 260 Btu/h·ft²·°F must be realized, and this can be done only in a fluidized bed. To make use of the other modes of operation, a conventional heat exchanger must be employed. Thus a pressurized helium loop must be used.

The literature on finned tubes is sparse. With one exception the tests have been performed on cold models. The fins used are varied, making it difficult to draw specific conclusions regarding configuration, fin and unit length, fin height, and local operating conditions. Improvements in heat-transfer rate/length of tube over bare-tube operation have varied from 160 to 500 percent. The data appear to be most consistent at 200 to 300 percent, which should provide the 260 Btu/h·ft²·°F (bare-tube equivalent) desired. Chen indicates a

reliability between 50 and 100 percent of indicated value. Some experimental work appears necessary before going to detailed design. Little service life data are available. Optimum operating conditions appear to be at 3 ft/s and small particle size. A parametric analysis should be made of the finned tube design to determine the configuration most desirable for experimental examination. Since the bed gas velocity is about 3 ft/s, the mean particle size should be about 300 μm with a top size of about 1000 μm . Therefore tube fins should be several times the top size, or no less than 1/4 in.

The outer fin diameter should exceed the inner diameter by at least the distance between fins to provide the additional effective surface required, as it has been suggested that the finned gas-side heat-transfer coefficient is close to that of the bare tube gas-side heat-transfer coefficient.

In the section on corrosion, we indicated that stagnant areas of the bed provide potentially troublesome corrosion problems. This problem may exist at the root of the fins; therefore, the problem of corrosion should be examined experimentally.

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Section 4

COST

Evaluation of the cost of the heat source components is difficult without going through a detailed conceptual design for a specific application. There are several factors making it nearly impossible to predict cost without further development of the concept. These include:

- Size - Several components in the heat-source system, including the mill for coal pulverizing, are beyond the size range normally considered commercially "off-the-shelf." The smallest commercially available pulverizer handles a minimum of 1200 lb/h. Other pulverizers are available that are used in other applications. Their suitability to this application must be evaluated.
- Combustor Cost Estimates - The combustor is an advanced state-of-the-art configuration using advanced state-of-the-art materials of construction. A unit would have to be custom fabricated. With no precedent, cost estimates could be considerably inaccurate. Cost projections on a production scale would be even less accurate.
- Coal Handling Plant - The coal preparation plant (which probably accounts for the greater portion of the cost) must be custom designed to the coal contract of the client. Coal storage may be the largest single cost component. Individual components may have to be oversized to meet the needs of the smaller unit. These components may not be commercially available in smaller sizes. Consequently, the process must absorb the additional capitalization.
- Electrical Demand - The electrical demand/process heat requirements will influence the cycle selected.
- Load Factor - The plant load factor will determine the reliability and duplicity that must be built into the coal preparation plant.
- Emissions - Emission requirements will determine which system will be the best for sulfur and particulate retrieval. There are presently little data with which to compare wet scrubbers, dry scrubbers, or lime-fed fluidized bed combustors.

To find some orders of magnitude for the cost of the components comprising the system, a list describing the cost breakdown for several much larger systems has been included.

Cost data have been tabulated in Table 2 for various combustor systems between 125 and 50,000 lb/h of coal. The data have been included to illustrate the contribution of the various components to the total cost, influence of capacity on cost, availability of cost in the range of operation of the Stirling Engine, and total cost of a combustion system. The table indicates data available is sparse in the range of the Stirling Engine. Furthermore, the variation in scope of the reported cost makes interpretation very difficult. Since the coal-handling, -receiving, and -preparation can represent about 40 percent of the cost, the fuel source and type of fuel contract will contribute significantly to the final total cost.

Tables 3 and 4 have also been included to indicate orders of magnitude of the influence material selection and heat-transfer surface have on the total cost.

The total cost of the heat source and the distribution of cost between major components are illustrated in Figures 25 and 26. These costs have been based on budget quotations, with the most reliable data being those for the combustors and fans. The scope of the remaining components is mixed and may not be complete. The accuracy is questionable.

Table 2 Cost Data for Combustor Systems (\$000)

Description	Type and Capacity (lb/h)								Coal-Fired Gas Turbine Air Heater ^a	Stirling Range [†]	Pulv. Coal Pilot Plant (1,260)
	Pulv. Coal ¹ (400,000)	Stoker ¹ (400,000)	FBC ¹ (400,000)	FBC (Conv.) ¹ (150,000)	FBC (Adv.) ¹ (150,000)	Stoker ¹ (150,000)	FBC ¹ (100,000)	FBC (40,000)			
kW Electrical	39.8	39.8	39.8	14.8	14.8	14.8	9.9	3.9	0.200		0.124
hp ¹ (x 10 ³)	53.3	53.3	53.3	19.8	19.8	19.8	13.2	5.2	0.266		0.167
Coal Flow (lb/h)	48,348	48,348	48,348	17,160	17,160	17,160	12,087	4,334	400		125
Primary Heat Source-- Combustor (\$ x 10 ³)	4620 ^{a,1}	4620 ^{a,1}	3626 ^{a,1}	4100 ^{a,1}	4080 ^{a,1}	3600 ^{a,1}	3321 ^a	2426 ^{a,1}	112.4 ^{a,1}		55.0 ^{a,1}
Auxiliaries											
FD Fans	41.25	41.25	40.4	11	11	11	50.0	35	9.5		2.6
ID Fans	63.17	63.17	52.3	11	11	11	42.0	30	5.0		2.6
Air Preheater	333.6	333.6	231.5	11	11	11	N/A ¹	N/A	24.7		---
Controls	120.0	120.0	120.0	11	11	11	35.0	20			---
Soot Blowers	102.2	102.2	---	11	11	11	---	---			---
Stoker/Distributor	---	581.5	134.0	11	11	11	---	---			---
Emission Control											
Cyclone	269.4	269.4	269.4	500 ¹¹	500	1560	20.0	150	12.00		1.29
Baghouse	---	---	---	---	---	---	138.0	---	800		4.12
Precipitator	666.0	666.0	855.0	---	---	---	---	---	---		---
Scrubber	7520	7520	---	---	---	---	---	---	---		---
Ash Handling											
Ash Handling	150 ¹¹	150 ¹¹	600	230	230	190	---	N/A	---		---
Solids Cooler	---	---	200.0	N/A	N/A	N/A	---	40.0	32.5		5.0
Limestone Feed	---	---	115.0	490 ¹¹	550 ¹¹	---	---	10	19.35		---
Coal Handling ¹¹	696.5	---	140.0	400 ¹¹	350 ¹¹	400 ¹¹	---	37.0	51.7		30.7
Coal Preparation and Receiving											
Coal Preparation	1311.0	1311.0	1311.0	630 ¹¹	630 ¹¹	630 ¹¹	524	109 ¹¹	N/A		4.16 ¹¹
Raw Coal Storage	---	---	---	1170 ¹¹	1170 ¹¹	1170 ¹¹	N/A	N/A	N/A		N/A
Coal Receiving	---	---	---	1060 ¹¹	1060 ¹¹	1060 ¹¹	N/A	N/A	N/A		N/A
TOTALS - \$/(\$/kW)	15791.1/ (396)	15777.6/ (396)	7694.2/ (193)	8580/ (579)	8570/ (579)	8610/ (581)	4130/ (417)	2847/ (730)	275/ (1375)		105.4/ (850)

^aSee legend on following page for explanation of superscript numbers.

[†]Combustor requires large high-temperature air preheater using superalloy steels to maintain high efficiency. All but the coal-fired heat exchanger for the gas turbine require carbon steel or T-22 at most--a 2-1/4Cr-1%Mo steel. Air preheater duty on the Stirling Engine is several times that of other systems. The maximum temperature level is about 1500°F requiring superalloys. The cost of the air preheater, therefore, should be several orders of magnitude greater than other systems.

¹N/A = Not available.

LEGEND

1. Study by Foster Wheeler does not include site development.
2. Study by McKee includes site development.
3. Combustor bids.
4. Study of a coal-fired gas turbine air heater based on bids to budget level.
5. Electrical equivalent, assuming engine efficiency of 40 percent and combustion efficiency of 85 percent.
6. Fluidized bed steam generator with water-cooled enclosure.
7. Budget estimate made on conceptual design detailed to proposal level.
8. Air-cooled fluidized bed with refractory-lined walls.
9. Cost of materials only for refractory-lined combustor.
10. Included in combustor scope--not broken out for detailed examination.
11. Includes site development.
12. Scrubber very seldom used as it has not been required until recently.
13. Excluding settling pond development.
14. All coal handling, transporting, and preparation from the day bin to the furnace.
15. Projected cost.

Table 3 Relative Cost of Steam Generator Components
(150,000 lb/h Industrial Package Steam Generator)

	<u>Percentage</u>	
Lower Drum	6.1	
Upper Drum	9.9	
Internals	1.2	
Bare Tubes	5.9	
MONO-WALL® Panels	9.2	Heat exchange surface--15.1%
Casing and Lagging	3.5	
Refractory and Insulation	4.8	
Shop Assembly	8.8	
Painting, Loading, and Shipping	1.4	
Miscellaneous Doors, Piping, etc.	3.24	
Engineering	5.0	
Valves	5.0	
Soot Blowers	1.0	
Burners and Safeguards	10.0	
Controls	20.0	
Fans	5.0	

Table 4 Cost Comparison of Tube Materials

<u>Material</u>	<u>Cost</u>
A-210	1.00
T-1	1.43
T-11	1.86
T-12	1.77
T-22	2.88
304	9.52
310	15.89
316	12.92
321	9.26
347	10.42

Figure 29 Approximate Cost of Combustion/kW (Based on studies of other systems)

Figure 30 Approximate Distribution of Cost of Combustion System/kW as a Function of Capacity
(Based on studies of similar systems)

Section 5

SUMMARY

It is imperative that a conceptual design study be executed for the Stirling Engine to select a system, optimize the alternatives, and arrive at a first cut in cost. Our previous discussion is summarized in Table 5. Based on this discussion, and without the benefit of further developmental work at a conceptual level, the fluidized bed appears to offer the best potential as a heat source for the Stirling Engine. The merits of using finned tube over bare tube construction are not clear. The merits of a helium cycle vs. a sodium loop appear to be more dependent upon application to the Stirling Engine than upon the heat source.

Table 5 Combustion Systems Summary Chart

COAL	System	Advantages	Disadvantages	Comments
	Stoker	<ul style="list-style-type: none"> No preparation of coal (such as pulverizing or drying) required. Particle size 1-1/4" x 0--no more than 40% <1/4". Simple design. 1.4 x 10⁴ Btu/h--500 x 10⁴ Btu/h size range. Furnace exit about 2300°F Eastern coals; 1950°F Western coals. 	<ul style="list-style-type: none"> Low heat-transfer rate.* Max. air preheat temperature 200 to 350°F. Process efficiency = 43%.† Air preheat 200°max. Excess air =250%. Low to high carbon loss. High excess air. Potential corrosion of air preheater between 1000 and 1500°F. 	<ul style="list-style-type: none"> Suitable for a pressurized He heat-transfer loop. Development required for feed systems of the small sizes needed. Sulfur recovery back end required. Impractical for rapid load swings. $\Delta T_m = 336 \text{ to } 232^\circ\text{F}.$‡

* $h_g = 5 \text{ to } 12 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$ (Lower value parallel flow); at $V_g = 100 \text{ ft/s}$ (Upper value cross flow);

Parallel flow may have to be used at the lower capacities to accommodate the bundle.

†All processes limited to 350°F flue gas exit temperature. Calculated efficiencies are only approximate.

Process efficiency may be further restricted by upper limit of minimum air preheat and the use of excess air to cool flue gases to temperatures less than 2300°F for ash deposit control.

Process efficiency calculated at 1300°F air preheat; $\eta = 74\%$; if 1000°F limit imposed, $\eta = 64\%$; if 750°F limit imposed, $\eta = 62\%$.

Typical Losses at 350°F Exit Temperature:

	Stoker, P.C., Cyclone	Fluidized Bed
Dry Gas	5.49	5.49
Fuel H_2	5.85	5.85
Air H_2	0.15	0.15
Combustion Loss	0.30	0.30
Dust	0.00	0.33
Radiation	0.28	0.28
Manufacturer's Margin	1.5	1.5
TOTAL	13.57	13.9
$\eta =$	86.42	86.1

‡ $\Delta T_m = 336^\circ\text{F}$ (Furnace exit = 2300°F, based on 100°F approach temperature).

$\Delta T_m = 232^\circ\text{F}$ (Furnace exit = 1950°F, based on 100°F approach temperature).

Table 5 Combustion Systems Summary Chart (Cont)

COAL

<u>System</u>	<u>Advantages</u>	<u>Disadvantages</u>	<u>Comments</u>
Pulverized Coal	<ul style="list-style-type: none"> • Fast transient. • Good turndown. • Can use 1300°F air preheat with special burner design.[†] • Low carbon loss 0.2 to 2%. • Furnace exits about 2300°F East-ern coals; 1950°F Western coals. 	<ul style="list-style-type: none"> • Low heat-transfer rate.[‡] • High cost of pulverizer system. • High power require-ments. • Process efficiency = 74%.[§] • Air preheat 1300° max. Excess air =250%. • Potential corrosion of air preheater at 1000 to 1500°F. 	<ul style="list-style-type: none"> • Sulfur removal by wet scrubber. • Present capacity range >100 x 10⁶ Btu/h. • Lower ranges require specially engineered mills. • Small sizes--turndown limited to capability of single burner. • 50 to 80% of all ash passes through boiler. • Coals limited to <0.4% Na₂O + K₂O dry coal basis and ash softening tempera-ture >2200°F. • Best suited for He loop; sulfur recovery back end. • ΔT_m = 336 to 232°F.[¶]

See p. 1 ().

†Air preheaters are normally designed to T_{meta} max 750°F using carbon steel. To 1000°F special materials must be selected with an incremental jump in cost. Above 1000°F the technology is not yet developed for proper material selection. The fluidized bed system has the best chance of capturing undesirable impurities that would subject the air preheater to corrosive abuse.

‡See p. 1 (†).

§See p. 1 (§).

Table 5 Combustion Systems Summary Chart (Cont)

COAL

<u>System</u>	<u>Advantages</u>	<u>Disadvantages</u>	<u>Comments</u>
Fluidized Bed	<ul style="list-style-type: none"> Fuel flexibility <ul style="list-style-type: none"> - Coal - Wood - Oil - Gas - Combustibles $H > 3000 \text{ Btu/lb.}$ High heat-transfer rate $h_g = 100 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F.}$ Dry ash handling system. No corrosion of heat-transfer surface in flue gas in the heat-recovery zone above the bed. Excess air - 10 to 20%. Sulfur recovered during combustion, end product--dry ash. 	<ul style="list-style-type: none"> Low temperature $< 1750^\circ\text{F.}$ Slow transient. Air preheat temperature over 1000°F difficult. Disposal of large quantities of fly ash. Ca/S ratio about 6:1 at low gas velocities to maintain high heat-transfer rates. Process efficiency = 74%.* Air preheat 1300°F max.. 	<ul style="list-style-type: none"> Good turndown ratio requires specific attention. Ideally suited for use with heat pipe transport loop. Can also be used with He loop. $\Delta T_m = 250^\circ\text{F.}$

*See p. 1 (+).

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Table 5 Combustion Systems Summary Chart (Cont)

COAL

<u>System</u>	<u>Advantages</u>	<u>Disadvantages</u>	<u>Comments</u>
Cyclone	<ul style="list-style-type: none"> • No preparation of coal (such as pulverizing or drying) required. • High flexibility for low-grade coal. • Low carbon loss--<0.1%. 	<ul style="list-style-type: none"> • High maintenance cost. • Low heat-transfer rate.* • Process efficiency = 74%.† • Air preheat 1300°F Excess air = 250%.‡ • Potential corrosion of air preheater surface 1000 to 1500°F. 	<ul style="list-style-type: none"> • Liquid slag discharges; requires ash fusion temperature <2400°F. • Present capacity range >100 x 10⁶ Btu/h. • Best suited for He loop. • Sulfur recovery back end required. • ΔT_m = 336 to 232°F.¶

See p. 1 ().

†See p. 1 (†).

‡Only 10 to 15% excess air is required for combustion. Approximately 250% is added downstream of combustor and upstream of heat-transfer surface for flue gas temperature control to avoid deposit formation.

¶See p. 1 (¶).

Table 5 Combustion Systems Summary Chart (Cont)

OIL

<u>System</u>	<u>Advantages</u>	<u>Disadvantages</u>	<u>Comments</u>
Residual Oil Burner	<ul style="list-style-type: none"> • Low cost. • Simple control and fuel handling. • Can use 1300°F air with special burner. 	<ul style="list-style-type: none"> • Low heat-transfer rate. • Vanadium treatment required. Must use Mg additive in a ratio of 3 Mg/1V for V > 0.5 ppm. Na must be washed to levels not exceeding 1 ppm. • Air preheater subjected to same corrosive abuse as turbine blading in the temperature range of 1050 to 1500°F. • Process efficiency = 74%. • Air preheat 1300°F Excess air 250%. 	<ul style="list-style-type: none"> • Gas turbine type of combustion system. • Sulfur may be controlled with an additive plus back-end cleanup. • $\Delta T_m = 336^\circ\text{F}$.

Section 6

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TASK 1A.3 REPORT

STIRLING ENGINE HEAT SOURCE

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TABLE OF CONTENTS

	<u>Page</u>
INTRODUCTION	1
DESIGN CONDITIONS	1
PARAMETRIC ANALYSIS AND SIZING OF THE FLUIDIZED BED COMBUSTOR TUBE BUNDLE	6
TOTAL WEIGHT AND TOTAL SURFACE	7
NUMBER OF TUBES	10
TUBE LENGTH	12
TUBE PITCH	12
BED DEPTH	14
HORSEPOWER REQUIREMENTS	14
CONCEPTUAL DESIGN	17
PULVERIZED-COAL FIRING	28
HEAT PIPE CONCEPT	30
CONCLUSION	30

LIST OF ILLUSTRATIONS

<u>Figure</u>		<u>Page</u>
1	Surface Requirements and Helium Flow Rates as a Function of Heat Exchanger Effectiveness	3
2	Influence of Tube Size and Helium Velocity on Bundle Weight	8
3	Influence of Tube Size and Helium Velocity on Surface Requirements and Overall Heat-Transfer Coefficients	9
4	Relationship of Number of Tubes to Tube Cross-Sectional Area for Various Size Tubes	11
5	Influence of Tube Diameter and Helium Velocity on Tube Length	13
6	Examination of Tube Size and Tube Pitch and Volumetric Heat Absorption	15
7	Influence of Tube Size on Bed Depth for Constant Volumetric Heat Absorption and Minimum Tube Bending Radius of One Diameter	16
8	Influence of Tube Size and Helium Gas Velocity on Horsepower Requirements as a Percentage of Horsepower Output	18
9	Influence of Tube Size on Fanning Pressure-Drop Coefficient	19
10	Conceptual Design of a Fluidized Bed Helium Loop Heat Exchanger for a Stirling Engine (Enclosed headers)	21
11	Conceptual Design of a Fluidized Bed Helium Loop Heat Exchanger for a Stirling Engine (Suspended headers)	22
12	Conceptual Design of a Fluidized Bed Helium Loop Heat Exchanger for a Stirling Engine (Headers relocated close together)	23
13	Vertical Tube-Bundle Arrangement	26
14	Alternative Concept for Firing Pulverized Coal	29

LIST OF TABLES

TablePage

1

Distribution of Surface Requirements and Cost--Fluidized
Bed Combustor and Pulverized-Coal-Fired Combustor

5

INTRODUCTION

During Task 1A.1, a superficial evaluation was made of several combustion systems, one of which would be used as the heat source for the Stirling Engine. The fluidized bed combustor, using a helium-cooled finned-tube heat exchanger or a sodium-filled finned-tube heat pipe, appeared the most promising system. The pulverized-coal-fired helium-cooled heat exchanger was somewhat less promising. The stoker-fired helium-loop heat exchanger and the cyclone-fired helium-loop heat exchanger appeared the least promising. The sodium-filled finned-tube heat pipe could not be practically applied to the last three systems. Further evaluation of the most promising combustion systems was pursued at the conceptual level by examining tube size, surface requirements, tube-bundle arrangement, and combustor design. The greatest portion of the effort in this task was expended on the bundle and combustor design of the fluidized bed combustor employing a helium-loop heat exchanger.

DESIGN CONDITIONS

The conceptual design was developed for a 1000-hp Stirling Engine using Illinois No. 6 coal. The engine efficiency was assumed to be 40 percent, and the heat source efficiency was assumed to be approximately 85 percent. To achieve these efficiencies, an air preheater was used with a flue gas inlet temperature of 1700°F and an outlet temperature of 470°F. At 70°F air inlet temperature, the air is preheated to 1300°F. The heat exchanger bundle has a duty of 6366×10^3 Btu/h. The coal flow rate is 657 lb/h. The flue gas flow rate is 7560 lb/h and the air flow rate is 6403 lb/h at 15 percent excess air.

In the pulverized-coal-fired combustor, the excess air must be increased to 330 percent because there is no furnace cooling, and the furnace exit temperature must be restricted to 2300°F to avoid slagging or fouling.

The fluidized bed temperature was assumed to be 1700°F, which is the upper limit for bed agglomeration and sulfur capture. The helium pressure was assumed to be 514 lb/in²a, with an inlet temperature of 1437°F and an outlet temperature of 1592°F. The helium flow rate was assumed to be 33×10^3 lb/h. The mean temperature differential for the bundle was 174°F. The bundle effectiveness, (i.e., the temperature rise of the helium divided by the difference in temperature between the bed and the helium at the inlet to the bundle) was 0.66. A further improvement in bundle effectiveness could be achieved by increasing the surface requirements as shown in Figure 1. A further increment in the heat-transfer surface would result in an increase in the number of tubes or an increase in the helium flow rate. In either case the fluidized bed would have to be bigger. This increase could be achieved by making the bed deeper, resulting in a greater flue gas pressure drop or a greater flue gas flow rate at a constant velocity.

A reduction in surface would reverse the trend. Incremental changes in surface requirements in either direction result in an incremental change in the pump horsepower required in the opposite direction. Selection of the most economical operating conditions and surface requirements necessitates optimization of the costs of pumping helium and flue gas and of heat exchanger surface. Development of this cost data requires detailed design, involving some iteration in the development of the details, which is beyond the scope of this task.

Figure 1 Surface Requirements and Helium Flow Rates as a Function of Heat Exchanger Effectiveness

Cycle analysis performed in Task 1A.1 indicated that a maximum efficiency of 85 percent could be achieved only by extending the state of the art of air preheater design and fluidized bed distributor design beyond its present level of development. The conceptual design was developed on the assumption that conservative air preheater tubewall surface temperatures of 750 to 950°F could be extended to 1500°F with proper alloy selection at the higher temperatures and that the carbon steel or stainless steel distributor plate could be replaced by a distributor plate of all-refractory design similar to that used in catalytic crackers. Restriction of air heat temperatures to 600°F would result in a substantial reduction in efficiency, a slight increase in combustor cost, and a substantial saving in the cost of the air preheater. Table 1 illustrates the distribution of surface and the approximate cost difference between the fluidized bed combustor bundle and the air preheater bundle at various temperature levels. Optimization of energy and surface costs should be performed to obtain the most economical and reliable heat source for the Stirling Engine.

The heat-source tube bundle immersed in the fluidized bed was sized assuming a maximum helium velocity of 150 ft/s, a fluidized bed combustor gas-side velocity of 3 ft/s and a fluidized bed combustor gas-side heat-transfer coefficient of 250 Btu/h·ft²·°F using finned tubes. The coefficient is based on bare tube surface at the inside diameter. The bare tube heat-transfer coefficient at 3 ft/s superficial gas velocity is estimated at 100 Btu/h·ft²·°F. In the event finned tubing cannot be used, 2.5 times as much surface must be installed in the fluidized bed, requiring an even further reduction in gas velocity, an

Table 1 Distribution of Surface Requirements and Cost--Fluidized Bed Combustor and Pulverized-Coal-Fired Combustor

<u>Surface Distribution (%)</u>			
<u>Air Preheater</u>	<u>Material</u>	<u>Fluidized Bed Combustor (FBC)</u>	<u>Pulverized-Coal-Fired Combustor (PC)</u>
Section 1	310	26.6	30.1
Section 2	T-22	26.6	30.1
Section 3	A-210	26.6	30.1
Combustor	800H	19.9	9.7
Ratio--PC/FBC			4.09:1

<u>Cost (\$)</u>			
Section 1	310	5.9	69.5
Section 2	T-22	10.7	12.6
Section 3	A-210	3.7	4.3
Combustor	800H	26.0	13.4
Ratio--PC/FBC			3.9:1

increase in bed depth, or both. Consequently, the bed size would increase and the power required to pump combustion air would increase. In the event a smaller tube size were selected to increase the packing of the surface in the bed, the impact of an increase in volumetric heat absorption on bed cooling and coal distribution would have to be evaluated. A maximum volumetric bed absorption of $70 \times 10^3 \text{ Btu/h}\cdot\text{ft}^3$ was assumed based on the present state of the art of fluidized bed combustors operating at 10 ft/s superficial gas velocity. Some experimental work is definitely required regarding heat transfer and coal distribution at the lower gas velocities.

Horsepower requirements for pumping helium through the fluidized bed heat exchanger were restricted to 2 to 3 percent of the horsepower output.

PARAMETRIC ANALYSIS AND SIZING OF THE FLUIDIZED BED COMBUSTOR TUBE BUNDLE

The fluidized bed envelope into which the heat exchanger tube bundle must be placed is 9 ft x 4-1/2 ft in plan and approximately 4 ft deep. The plan area is established by the gas velocity and particle sizing and their impact on heat-transfer coefficients. A 4-ft depth is used as a guideline. A further increase in expanded bed depth would result in a drop in pressure to a level exceeding that normally considered the upper tolerable level. Additional bed height (18 in.) is required between the bundle and distributor plate to permit mixing, lateral circulation, and dissipation of distributor plate jets. The length-to-width ratio of 2:1 was selected to minimize wall surface, accommodate over-bed feeders, and provide a reasonable compromise between the number

of tubes, number of rows, and tube length required. The bundle size and arrangement must be parametrically examined in terms of tube size and helium velocity. Such an examination will result in the least expensive bundle for the envelope. Ultimate selection of the bundle arrangement depends upon the surface cost, helium and flue gas pumping costs, and technical constraints such as maximum volumetric heat absorption.

TOTAL WEIGHT AND TOTAL SURFACE

The total weight and relative surface requirements were calculated as a function of tube size and helium velocity at a helium pressure of 514 lb/in²a (Figures 2 and 3). The tube size is reported in terms of O.D. and I.D. The difference reflects the tube thickness required for Incoloy 800H tubing at 1500°F and 514 lb/in²a. The thickness does not allow for corrosion. The heat-transfer coefficient is expressed as a fraction of that required by an 0.50-I.D. tube passing helium at 150 ft/s. The coefficient is based on the I.D. and assumes a value of 250 Btu/h·ft²·°F on the fluidized bed side, achieved as a result of including a finned outer surface. Since the mean temperature differential and the duty are constant, the ratio of the overall coefficients is also a reflection of the surface required. The reduction in overall heat-transfer coefficients with increased tube size and helium velocity is the result of an increase in inside diameter film coefficient. The lower heat-transfer coefficient can be overcome by increased helium velocity or the insertion of a swirling device to permit turbulence. These options can be utilized only at the expense of additional power consumption or fabrication cost. A more detailed study of the economics is required before considering either.

Figure 2 Influence of Tube Size and Helium Velocity on Bundle Weight

Figure 3 Influence of Tube Size and Helium Velocity on Surface Requirements and Overall Heat-Transfer Coefficients

The curves for tube weight vs. tube size show a substantial increase in weight with tube size. The curves reflect an increase in tube thickness as well as surface requirements. On the basis of cost/unit weight, the curves qualitatively illustrate the expected increase in the cost of a tube bundle with a corresponding increase in tube size and helium velocity. The curves do not, however, show the fabrication or header costs. The latter may increase substantially when the number of tubes is increased as a result of an increase in size and an increase in fabrication cost.

NUMBER OF TUBES

The number of tubes selected depends on the tube size and the velocity desired. As the velocity increases, the number of tubes required decreases and the required tube length increases for a given mass flow and exchanger duty. At a constant velocity, the number of tubes required increases as the diameter of the tube increases. The relationship of number of tubes and tube cross-sectional area for various sized tubes is shown in Figure 4. Many tubes are required if their diameter is small. Therefore, the smaller tubes require larger headers to ensure sufficient strength while accommodating many penetrations. A greater number of tubes also requires an increase in field operations during assembly of the bundle to the header. The combined cost of larger headers and increased fabrication operations should increase the cost of the bundle as the tube size decreases. A quantitative evaluation can be made only during a detailed study.

Figure 4 Relationship of Number of Tubes to Tube Cross-Sectional Area for Various Size Tubes

TUBE LENGTH

Figure 5 illustrates the tube length required for a particular fluidized bed envelope as a function of tube diameter and helium velocity. Matching of the tube length to the fluidized bed dimensions is illustrated by the number of loops required for a given header size. The combination of the number of tubes/loop and the number of loops required will affect the bed depth and thus the horsepower requirements for the combustion air fan.

An increase in the number of tubes results in a reduction in the length required for a given duty and mass flow. Large tubes longer than the constraining limits of the fluidized bed envelope must be bent in a serpentine fashion, forming loops between headers. The number of loops depends upon the ratio of tube length to bed width or length. For mismatched lengths, a slight adjustment may be made in the fluidized bed dimensions, tube pitch, helium velocity, surface requirements, or fluidized bed velocities to force a match.

TUBE PITCH

In the foregoing analysis, the tube pitch was assumed constant. As the tube size becomes smaller, the spacing between tubes at a constant pitch is substantially reduced, and mobility of particulates through the bed is impeded. Immobility of the coal could cause hot spots and zones of low heat release, reducing the effectiveness of the heat exchange and possibly causing local clinkers to form. A reduction in space between tubes also results in an increase in volumetric heat absorption in the bed. Stated in another way, too much surface packed into the bed results in overcooling, which can cause increased carbon carryover or instability in the fluidized bed. Fluidized beds

Figure 5 Influence of Tube Diameter and Helium Velocity on Tube Length

are presently being designed for 110×10^3 Btu/h·ft². Assuming a conservative value of 70×10^3 Btu/h·ft², the pitch for tubes below 2.25-in. O.D. must be increased with a decrease in tube size. The influence of tube size on tube pitch and volumetric heat absorption is illustrated in Figure 6.

As the tube size increases from 2.25-in. O.D., the volumetric heat absorption will decrease. The tube pitch cannot be reduced below two diameter, because the minimum bending radius dictated by hot bending without causing excessive thinning is one diameter for tubes between 1/2- and 3-in. O.D.

BED DEPTH

The bed depth illustrated in Figure 7 as a function of tube size and helium velocity is dependent upon the number of vertical rows and loops required. A variable tube pitch must also be taken into consideration to account for the constant volumetric heat release required. The optimum bed depth for the bundle duty and helium mass flow selected appears to be achieved with tubes 1.5- to 2.25-in. O.D.

HORSEPOWER REQUIREMENTS

The bed depth for a given bundle duty and thus a given combustion air flow affects the horsepower required for the combustion air fan. The optimum horsepower required for this fan is achieved when using 1.5- to 2.25-in. O.D. tubes. Large tubes with a minimum tube pitch of twice the diameter require deeper beds. Smaller tubes with a constant volumetric heat absorption, increased number of tubes, and vertical rows also require a deeper bed and, consequently, a combustion air fan with more horsepower.

Figure 6 Examination of Tube Size and Tube Pitch and Volumetric Heat Absorption

Figure 7 Influence of Tube Size on Bed Depth for Constant Volumetric Heat Absorption and Minimum Tube Bending Radius of One Diameter

The total horsepower must include the combustion air fan and the horsepower to pump the helium through the heat exchange bundle. The latter depends upon the helium velocity, as illustrated in Figure 8, and the tube length, as illustrated by the Fanning coefficient in Figure 9. Figure 8 indicates that the horsepower required for helium is small when compared with that for the combustion air. The optimum requirements occur with the use of 1.5- to 2.5-in. O.D. tubes with a helium velocity of 100 ft/s. At 150 ft/s, the total horsepower required is slightly higher for the optimum tube selection. The cost of surface, however, is probably somewhat less--attributable to a lower weight and fewer tubes. A truly optimized bundle configuration cannot be achieved until the costs of materials and energy are factored into the selection.

CONCEPTUAL DESIGN

The bundle configuration having been determined, the containing vessel may be developed. The optimum bundle contains 1-1/2 in. I.D. tubes with 2.25-in. O.D. on 4-1/2 in. x 4-1/2 in. centerlines. The bundle consists of 1-1/2 loops, each containing five rows of tubes. The bundle fits into a bed volume 4-1/2 ft x 9 ft x 4 ft deep. An additional 18 ft is provided between the distributor plate for jet dissipation.

Since cogeneration is not being considered as an application, a refractory wall enclosure is used. The wall consists of 2 in. of hard-face castable over 7 in. of castable attached to 1/4-in. reinforced carbon steel plate. The bed dimensions are 9 ft x 4-1/2 ft.

Figure 8 Influence of Tube Size and Helium Gas Velocity on Horsepower Requirements as a Percentage of Horsepower Output

Figure 9 Influence of Tube Size on Fanning Pressure-Drop Coefficient

Three arrangements have been considered. In the first case, illustrated in Figure 10, the headers are enclosed in an insulated vestibule outside the bed, providing protection of the headers from erosion by bed material. The headers leave the enclosure at one end. This construction suffers because of the external expansion joint required to absorb lateral, longitudinal, and vertical expansion differentials between the Stirling Engine and the fluidized bed. However, this concept is attractive because the headers and bundle supports are located in a cool zone.

The second arrangement, illustrated in Figure 11, suspends the headers from constant-load springs in the freeboard zone above the bed. The concept utilizes a loop-and-a-half arrangement (compared with the single loop shown in Figure 11 to illustrate the effect of bundle arrangement on overall design). Conversely, the drawing illustrates the constraints of the Stirling Engine construction on bundle design. A third arrangement, shown in Figure 12, illustrates what must be done to relocate the headers close together to provide a straight line of piping to the Stirling Engine.

The supports for the headers, which are in the freeboard area above the fluidized bed, are located in vestibules open to the flue gas pass but in stagnant areas not subjected to particulate loading or high heat-transfer rates. For further protection against the coal fired from above the bed, Incoloy 800H shields must be installed over the headers immediately above the bed.

Figure 10 Conceptual Design of a Fluidized Bed Helium Loop Heat Exchanger for a Stirling Engine (Enclosed headers)

Figure 11 Conceptual Design of a Fluidized Bed Helium Loop Heat Exchanger for a Stirling Engine (Suspended headers)

Figure 12 Conceptual Design of a Fluidized Bed Helium Loop Heat Exchanger for a Stirling Engine (Headers relocated close together)

The helium loop and heat source are major problems that must be dealt with during a detailed design development. The headers must be protected from excessive heat transfer (resulting in high metal temperatures and very low allowable stresses), and the bundle must be supported with cool supports that will accommodate differentials in load during expansion and permit unimpeded expansion differentials between the Stirling Engine.

Apparently, 8- to 10-in. headers are the smallest that will support a tube bundle of 1-1/4 in. x 1-1/2 in. diameter tubes or smaller over a span of 4-1/2 ft. These headers are inadequate for the larger 9-ft spans, where fewer tubes with larger diameters may be employed. Larger diameter headers may be required, and intermediate supports to reduce the span may also have to be considered for the larger tubes.

The headers should be located in cool gas zones to restrict metal temperatures to the same level as the discharging helium--about 1550°F. This can be achieved by locating the header outside the bed or discharging flue gas cavity, as shown in Figure 11 or 12. If the header must be located in the hot flue gases to preclude expansion problems, a stagnant gas zone should be created to avoid excessive heat transfer.

The header supports should be protected from excessive temperatures and particulate impingement. This protection can be achieved by creating vestibules stuffed with packing using cool combustion air, 1300 bleeds feeding the vestibule, or using water-cooled hangers.

Location of the bundle in the flue gas stream and support of the entire bundle on springs reduce the wall penetrations to one and thereby minimize the problems associated with expansion differentials between the helium loop and the combustor. The problems are further simplified by the use of constant-load springs. This arrangement is still not free from loads introduced by the sealing gland or friction forces in the constant-load springs. Therefore, a hydraulically supported combustor casing may have to be developed. The hydraulic system is responsive to any expansion differential between the turbine and the combustor that is registered by a strain gage on the interconnected piping. The strain gage provides a signal to actuate the hydraulic system which, in turn, makes the necessary compensation to relieve the stress.

Hangers are illustrated in all the arrangements to tie together the individual loops and to complete the necessary in-bed support. These ladder-type hangers will be at bed temperature (1700°F). Materials of suitable allowable stresses have not been selected. Water-cooled supports may have to be developed as an alternative. Development of the hanger design is beyond the scope of this program.

In the horizontal tube-bundle arrangement, the tubes must span 4-1/2 to 9 ft unsupported. At the required metal temperatures and pressures, the section modules of the smaller tubes may be inadequate for self-support. A vertical arrangement (Figure 13) may be needed. In this arrangement, which may ultimately be preferred, the headers are located in the plenum chamber, where the air temperatures are approximately 1300°F. This arrangement introduces

Figure 13 Vertical Tube-Bundle Arrangement

some expansion problems where the tubes penetrate the distributor sheet as well as where the tubesheet comes in contact with the enclosure wall. Erosion of the tubes as they exit from the distributor plate on the fluidized bed side would have to be examined closely. The tube-bundle supports would be reduced considerably; however, aligning brackets would have to be installed.

The final tube-bundle arrangement can be selected only after a more detailed layout and analysis are made of the individual problems already identified.

Both under-bed and over-bed feed systems are shown because of the uncertainty about the method to be used. If the over-bed system were selected at this point, a special feeder would have to be developed. Over-bed feeding does not require fine crushing of coal. With under-bed feeding, the coal must be crushed to 1/4 in. x 0. For the Stirling Engine heat source, there are definite advantages in developing a new feeder.

The cycle studies in Task 1A.1 indicated the need for a high-temperature air preheater to maintain high efficiency--even though the design of such a heat exchanger would mean advancing the state of the art. For all three arrangements, we propose that the hottest stage of air preheating be accomplished in the flue gas stream immediately above the fluidized bed, thereby taking the utmost advantage of the refractory enclosure. In this way the initial stage of air preheat can proceed in the more traditional fashion, outside to the fluidized bed in a tubular air preheater.

PULVERIZED-COAL FIRING

An alternative concept for firing pulverized coal (Figure 14) was developed to provide a contrast between the fluidized bed and other energy sources. A minimum of three vessels is required: a combustor, a heat-exchange vessel, and at least one air preheater. In detailed development of the air preheater, it might be best--for expansion purposes--to separate the last stage of air preheat from the first two.

The combustor is a vertically down-fired unit 12 ft long and 6 ft in diameter. The wall construction consists of 2 in. of refractory and 6 in. of hard-castable on 6 in. of castable backed with 1/4 in. reinforced carbon steel wall. The burner design fits into a gas-turbine-like combustor. Excess air (330 percent) is introduced into a windbox surrounding the combustor to prevent quenching of the flame. The two gas streams merge in an open chamber 4 ft downstream of the burner. Down-firing was selected to provide easy retrieval of hot ash dropping out of the flue gas stream. (The flue gas leaves the combustor through the side immediately above the ash hopper.)

The Stirling Engine heat exchanger is a vertically hung, double-loop unit consisting of four rows of 2-1/4 in. O.D. tubes on 4-1/2 in. x 4-1/2 in. centerlines. The headers are hung from constant-load springs, which provide absorption of expansion differentials between the heat source and the Stirling Engine. The flue gases discharge from the upper portion of the bundle into an air preheater. The heat exchanger enclosure consists of refractory wall construction approximately 9 in. thick.

Fig. 12

Figure 14 Alternative Concept for Firing Pulverized Coal

The arrangement proposed requires elevation of the Stirling Engine seven feet above grade to accommodate a straight run of piping for helium. In the final design, the elevation may have to be increased substantially as a means of reducing the width to achieve an improved flow pattern through the bundle.

HEAT PIPE CONCEPT

The heat pipe fluidized bed concept would be identical to that of the helium-loop fluidized bed concept except that the bundle would be replaced with eight rows of 5-in. O.D. finned heat pipe, four rows high and 9 ft long. Two heat pipes from each of the eight Stirling Engine cylinders would be subdivided into two rows within the fluidized bed, forming four rows of tubes in a vertical plan.

The major problems with the heat-pipe design would be sealing of the pipes at the bed wall and accommodation for expansion differentials between the heat source and Stirling Engine within the seal. Another problem involves the removal of tubes intact by disengagement of the seal and withdrawal the individual pipes. Further evaluation of this concept requires a detailed layout which is beyond the scope of the task.

CONCLUSION

Critical areas requiring further detailed study include:

- Optimization of air preheater surface
- Detailed design and cost estimation of the air preheater
- Optimization of the immersed heat-exchanger bundle surface and horsepower requirements

- Detailed development of the header-to-casing seal
- Detailed design of the bundle support system
- Detailed layout and sizing of the bundle and headers and examination of the impact of tube size, tube thickness, and tube support requirements on tube arrangement
- Development of a suitable grid plate design
- Development of an over-bed feeder
- Evaluation of the economic merits of installing turbulators within the tubes in the immersed tube bundle
- Detailed design of the tube fins to finalize the fluidized bed gas-side heat-transfer coefficients.