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MASTER

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

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Advanced Mechanical Technology, Incorporated
Newton, Massachusetts

U. S. DEPARTMENT OF ENERGY

CONVERSION

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

VOLUME 1: TECHNICAL REPORT

Prepared for:

Argonne National Laboratory
Components Technology Division

Prepared by:

ADVANCED MECHANICAL TECHNOLOGY, INC.

141 California Street

Newton, Massachusetts 02158

September 15, 1980

Sponsored by:

Department of Energy
Assistant Secretary for Fossil Energy
Office of Coal Utilization

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The prime contractor was Advanced Mechanical Technology, Inc. of Newton, Massachusetts. The principal authors from AMTI were: Mark Schuetz, Project Engineer; Dr. Joseph Gerstmann, Vice President and Program Manager; and Dr. Lawrence Hoagland, Director of Research; with assistance from Dr. Walter Syniuta, President; Rolf Krisher, Design Engineer; and Cyril Randall, Chief Design Engineer.

The principal subcontractor in the area of Stirling engine design was K.B. United Stirling (Sweden) AB & Co. Christer Bratt and John Berntell were the principal contributors to the project. Bengt-Ove Moodysson, Herbert Nilsson, Stig Gummesson and Lars Sjoestedt also contributed. Ricardo Consulting Engineers, Ltd. of England were engaged as consultants to United Stirling in the area of engine design. The principal contributor from Ricardo was V.C.H. Pope, with additional contributions from I.D. Johnstone, C.M. Andrews, and A.R. Crouch.

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1. INTRODUCTION AND SUMMARY

1.1 PROJECT BACKGROUND AND OBJECTIVES

This project was Phase I of a multi-phased program for the design and development of Stirling engines for stationary power generation applications in the 500-3000 horsepower range. Phase I comprised the conceptual design and associated cost estimates of a stationary Stirling engine capable of being fueled by a variety of heat sources, with emphasis on coal firing, followed by the preparation of a plan for implementing the design, fabrication and testing of a demonstration engine by 1985.

The main effort in Phase I was the generation of state-of-the-art conceptual designs having greatest potential for prototype testing in 1985, along with advanced conceptual designs based on new technologies which may require significant R&D before commercialization. The conceptual designs include not only the engine and combustion system but also a heat transport system for integrating the engine heater head with such energy sources as conventional oil/gas combustors, coal combustors, combustors using coal-derived liquid fuels, and low/medium Btu gas combustors.

Advanced Mechanical Technology, Inc. (AMTI) was the prime contractor for Phase I, with direct responsibility for system conceptual design and for conceptual design of the heat source and heat transport sub-systems. KB United Stirling (Sweden) AB & Company (United Stirling) was a subcontractor responsible for engine conceptual design. They were assisted in this effort by Ricardo Consulting Engineers, Ltd., of England. Thermacore, Inc., Leola, Pennsylvania served as consultants to the project in the area of heat pipe conceptual design.

The development and evaluation of conceptual designs has been separated into two broad categories: the "A" designs which represent the present state-of-the-art and which are demonstrable by 1985 with minimum technical risk; and the "B" designs which involve advanced technology and therefore would require significant research and development prior to demonstration and commercialization, but which may ultimately offer advantages in terms of lower cost, better performance, or higher reliability. The majority of the effort in Phase I was devoted to the "A" designs.

"A" Designs

The emphasis in the "A" designs was on proven concepts which are demonstrable by 1985 with minimum technical risk. The large engines are to be capable of being adapted to firing with coal as well as oil, gas, coal derived liquids or low/medium Btu gas combustors.

When "dirty" fuels such as coal are employed, it is not desirable to transfer heat directly from the combustion gases to the engine heater tubes because of the low heat fluxes allowed by tube fouling. Low heat flux leads to excessive internal void volume in the heater head. This is detrimental to engine performance. Therefore, the approach adopted in this study was to employ an intermediate heat transport system for transferring heat from the combustion gases to the heater head. With this design approach the system is capable of burning a wide variety of fuels ranging from the dirty fuels such as coal, residual oils, refuse, and biomass to the cleaner fuels such as gas and oil. Design efforts were thus concentrated on coal firing for the demonstration engine system with the understanding that this heat transport system concept allows sufficient flexibility for adaptation to use other fuels as

well.

The overall power system conceptual design work was divided into three principal subsystems as follows: heat source, heat transport system, and Stirling engine. Conceptual design work on these subsystems is described under those headings in major sections of this report.

Heat Source

The primary effort in this segment of the program was directed toward evaluation of current technology in coal combustion equipment. The state-of-the-art systems evaluated were stoker, pulverized, cyclone, fluidized bed, and low Btu gas. Selection criteria employed in this evaluation consisted of exhaust emissions, efficiency, fuels flexibility, size flexibility and cost. The fluidized bed was selected as the first choice for the combustion system due mainly to its superior emissions characteristics, its potential for lower cost, and its ability to transfer energy to the heat exchanger surface without severe tube fouling and slagging problems. Although fluidized bed combustion is a relatively new technology, there are currently several installations in operation and the equipment is offered commercially so it may be considered state-of-the-art technology.

Heat Transport System

The heat transport system design studies considered three different methods for indirect heat transport: forced convection liquid loop, forced convection gas loop, and heat pipe. In addition, the possibilities for direct firing with coal combustion products and low Btu gas were also examined. A two-stage sodium heat pipe was selected as the best heat transport system due to its superior characteristics.

with respect to initial cost, performance and the ability of the chosen design concept to provide acceptable life, reliability and field serviceability characteristics. Although there appears to be some possibility for direct firing with a fluidized bed coal combustor, this was not considered to be state-of-the-art technology and was classified as an advanced technology "B" system.

Stirling Engine

The technical approach followed on the large engine design was to retain the basic double-acting, 4-cylinder, twin-crank, U-form engine module developed by United Stirling for their latest small power range engines, (P-40 and P-75 - Figure 1-1) and to scale up the design to a 500 hp module which would serve as a building block for a line of large engines covering the power range from 500-3000 hp. Both direct fired (clean liquid or gaseous fuels only) and indirect fired (utilizing a heat transport system) engine designs were considered. In the process of scaling up the United Stirling P-75 engine to a large direct-fired unit it became clear that certain changes to the design were desirable, e.g., 4 regenerators per engine cylinder instead of 2, separate cylinder block quadrants for each individual cylinder (to be bolted onto a common crankcase) and possibly a different heater tube cage arrangement to facilitate making repairs on one cylinder with minimum disruption of adjacent cylinders. The heater temperature, 720°C, was elected in order to keep down the cost of hot parts and provide long life as would be required for a large stationary engine. Details are presented in Section 2.3.1.

Similar features were found desirable for an indirect fired engine designed to interface with a sodium heat pipe. For this case, however, the preferred design employs 12 regenerators per engine cylinder arranged symmetrically around each cylinder. Several alternative schemes were considered for handling the interface between the engine heater head and the secondary heat pipe, but the preferred design has the engine placed upside-down over the fluidized bed combustor and

FIGURE 1-1: P-75 CROSS SECTION

incorporates one secondary heat pipe per engine cylinder. Further details are provided in Section 2.3.2.

Complete Coal-Fired Stirling System

The conceptual design for the state-of-the-art coal-fired Stirling plant is illustrated in Figures 1-2 and 1-3. The system consists of a fluidized bed combustor (FBC) with the Stirling engine mounted upside-down above the FBC. A two-stage gravity assisted sodium heat pipe is used to extract heat from the FBC and transport it to the Stirling engine heater tubes.

The two-stage heat pipe system utilizes a vertical array of primary heat pipes with their evaporator sections extending down into the bed. Nominally adiabatic portions pass through the FBC freeboard area, and the condenser sections interface with the secondary heat pipes which surround the engine heater tubes. Several (e.g. 64) primary heat pipes supply each secondary heat pipe. One secondary heat pipe then serves each engine cylinder, providing heat at a uniform temperature to all of the cylinder's heater tubes.

This two-stage heat pipe provides high reliability and simplifies the engine/heat pipe interface because:

1. Failure of an individual primary heat pipe will not require system shutdown. In fact, 15 or 20% of the primary pipes can fail before the performance loss will mandate repairs.
2. The secondary heat pipe is isolated from the harsh conditions of the FBC and hence its reliability is considerably greater than that of a primary heat pipe, which in turn can be easily replaced.

LARGE ENGINE - 500 HP MODULE

FIGURE 1-2

SINGLE MODULE FLUIDISED BED

1. Coal Storage	8. Forced Draft Fan
2. Sorbant Storage	9. Baghouse
3. Feed Hoppers	10. Ash Storage
4. Stirling Engine	11. Stack
5. FBC and Heat Transport	12. Induced Draft Fan
6. Reinjection Cyclone	13. Generator
7. Preheater	14. Office/Control Room

FIGURE 1-3: OVERALL PLANT SKETCH
1500 hp COAL FIRED
STIRLING ENGINE

3. One secondary heat pipe per cylinder simplifies the engine-heat pipe interface design by minimizing stresses which arise due to manufacturing tolerances and thermal gradients.

For system power levels greater than 500 hp, a multi-module engine/FBC assembly can be arranged in a variety of ways. Two of the more likely multiple arrangements are in-line (Figure 1-4) and opposed (Figure 1-5). In the in-line arrangement from two to six 4-cylinder engine modules are placed end-to-end and connected to a single large external drive shaft placed alongside the engine modules. A single generator is driven from this shaft. Each engine module is connected to the drive shaft through a coupling arrangement provided to facilitate power modulation and allow maintenance of individual modules.

In the opposed module arrangement (Figure 1-5) the engine modules are placed on each side of the main drives so they appear in pairs of two opposed modules placed end-to-end along the external shaft as illustrated. This scheme has the possible advantage of a more compact (shorter) arrangement and it may ease the balancing requirements for individual engine modules (see Section 2.3.3).

A complete coal-fired system incorporating 3 engine modules in an in-line arrangement is illustrated in Figure 1-3. In addition to the fluidized bed and engine/generator assembly, this drawing shows the other major components required, such as coal and limestone silos, conveyors, feed hoppers, combustion air forced-draft and induced-draft fans, cyclone separator, air preheater, baghouse and ash removal equipment. It is readily apparent that the total power generation plant requires considerably more physical space than that occupied by the fluidized bed combustor and engine generator equipment alone.

LARGE ENGINE - 3000 HP ASSEMBLY

FIGURE 1-4: PLAN VIEW

6 MODULES - IN-LINE - FLUIDISED BED

LARGE ENGINE - 3000 HP ASSEMBLY

FIGURE 1-5: PLAN VIEW

SIX MODULES - OPPOSED

Typical system performance is illustrated by the block flow diagram of Figure 1-6 which depicts a heat and material balance for a single engine module operating at rated load conditions (Heater-720°C; Cooler Exit-50°C, 900 RPM, 15 MPa mean pressure). In Figure 1-6 the boiler efficiency is assumed to be 83% based on calculations for a reasonable preheater effectiveness. Further detailed analysis may prove that a more effective preheater would be more economical. The projected usable electric and thermal outputs of the Stirling system are shown in Figure 1-7 as a function of engine exit cooling water temperature. Not shown in Figure 1-7 is the effect of cooling water temperature on engine power, and efficiency. See Section 2.3.1 for engine performance curves as a function of temperature. An alternate representation of the energy flows which identifies the component efficiencies is shown in Figures 2-70, 2-71 and 2-72 for various applications (See Section 2.4.5).

Equipment Costs

The estimated equipment costs for a coal-fired Stirling engine generator set are summarized in Figure 1-8 on the basis of 1980 dollars per kW installed net electric capacity for systems in the nominal range of 500-3000 hp (375-2200 kW). See Figure 1-6 for net capacity used as the basis for cost estimating (392 kW(e)). Costs are subdivided into various components or sub-systems such as engine, generator/controls, heat source (FBC), heat transport, installation, and heat source auxiliaries (coal and ash handling equipment). Two different cost estimates are shown; one based on a low production volume for the heat source and auxiliaries and the other based on a high production volume "packaged system" for the heat source and auxiliaries. The estimated total system installed costs for these two cases are \$1483/kW(e) for the low production volume heat source case and \$950/kW(e) for the high production volume heat source case. An engine production volume of 1000 units per year was assumed for both cases.

Heat, Mass and Work Flows

————— solids flow
 - - - - - gas flow

All quantities nominal (typical seasonal average)

per 500 Hp module

* Brake Shaft Power is the shaft power from the engine to the generator. Engine mechanical losses are included but auxiliaries are not included as they would most likely be electrically driven.

FIGURE 1-6: COAL FIRED STIRLING
FLOW DIAGRAM

FIGURE 1-7: COAL PILE TO BUSBAR
EFFICIENCY VERSUS COOLING
WATER ENGINE OUTLET TEMPERATURE

FIGURE 1-8: ESTIMATED COST OF A COAL-FIRED STIRLING ENGINE

	Low-Volume Production Heat Source \$/kW	High-Volume Production Heat Source \$/kW
Engine	270	270
Generator, Controls, Heat- Recovery, Installation	<u>195</u> 465	<u>195</u> 465
Heat Source (AFBC)	221	129
Heat Transport (2-Stage Heat Pipe)	64	49
Installation	<u>221</u> 506	<u>65</u> 243
SUBTOTAL	971	708
Heat Source Auxiliaries	256	161
Installation	<u>256</u> 512	<u>81</u> 242
TOTAL	1483	950

Power Production Costs

A. Electricity Generation Only: In order to determine the cost of electric power produced by the coal-fired Stirling engine, the equipment costs summarized above must be combined with estimates of the operating and maintenance costs including fuel and sorbent charges, equipment repair, waste disposal services, and capital charges. Figure 1-9 summarizes all of these costs for a case corresponding to the high production volume combustion system mentioned above and based on the following additional assumptions:

1. Engine cooling water exit temperature of 50°C, and nominal system performance as shown in Figure 1-6.
2. 8000 operating hours per year at an average 70% load factor.
3. 25% annual charges for capital invested including taxes and insurance.
4. Service and maintenance intervals for various components based on their current projected lifetimes as defined in Sections 2.2.6 (Heat Pipe) and 2.3.4.2 (Engine) of this report. Coal combustion equipment O&M costs were estimated at 10% of equipment costs per year.
5. Coal cost @ \$1.65/10⁶ Btu and limestone @ \$6.50/ton and a 2:1 ratio of coal to limestone.
6. Ash removal cost at \$7.50/ton and ash content of coal @ 10%.
7. No credit for waste heat utilization

Figure 1-9 shows a power production cost of 9.3 cents/KWH based on these assumptions. Approximately half of this amount, \$.043/KWH represents capital charges while about 1/4 represents fuel and sorbent

FIGURE 1-9: ESTIMATED ELECTRIC POWER COSTS OF A
COAL-FIRED STIRLING ENGINE

(\$/KWH)

	<u>CAPITAL</u>	<u>O&M</u>	<u>TOTAL</u>
Engine, Generator, Controls, Heat Recovery	.021	.017	.038
Heat Source	.009	.002	.011
Heat Transport	.002	.004	.006
Heat Source Auxiliaries	<u>.011</u>	<u>.003</u>	<u>.014</u>
Sub-total, Non-fuel Costs	.043	.026	.069
Coal	-	.020	.020
Sorbent	-	.002	.002
Ash Disposal	-	<u>.002</u>	<u>.002</u>
Sub-total, Fuel Costs		.024	.024
TOTAL COST	.043	.050	.093

BASIS: 8000 hrs/yr, 70% load factor, 25% capital charge rate

and ash disposal costs and $1/4$ represents O&M costs. This compares with an estimated \$.106/KWH for a Diesel engine powerplant based on current installed costs for Diesel installations of comparable size (\$550/kw) and a diesel fuel cost of \$0.90/gallon, along with O&M costs of 1.2 cents/KWH (25). Thus, the coal-fired Stirling offers a 14% savings in power cost over the Diesel. As fuel prices escalate the advantage of the coal-fired Stirling over the Diesel will become even greater.

The example above was based on a 70% load factor for the Stirling system. Due to the relatively high capital cost and relatively low fuel cost this equipment must be operated at a relatively high load factor in order to show an economic benefit over the Diesel. Thus, high load factor applications should be selected for this equipment.

B. Waste Heat Utilization: Although the above cost comparison of a coal-fired Stirling power system versus a current Diesel installation shows a favorable advantage for the Stirling, even greater benefits accrue in applications where waste utilization provides a further economic advantage. In most cases where waste heat from the Stirling engine cooling water is used for space heating, process heating, etc. it becomes necessary to raise the cooling water temperature above the value that otherwise might be used and thereby make a small sacrifice in engine power output and efficiency. However, there is usually a net gain in overall economic benefit providing the economic value of the waste heat is substantial in comparison with the electric power costs. Figure 1-10 provides a useful summary of this relationship by showing the net cost of electric power generated from a coal-fired Stirling as a function of cooling water temperature and the parameter UxV , which is the product of the waste heat utilization factor U (fraction of available waste heat that is used) and the economic value of the waste heat V (\$/10⁶ Btu). For example, assuming that 50% of the engine reject heat is used at 80°C and the value of that heat is that of oil (\$8/10⁶ Btu), then the resulting cost of electricity would be approximately \$.079/KWH. These relationships are discussed in greater detail in Section 4.

FIGURE 1-10: ESTIMATED COST OF ELECTRICITY VERSUS COOLING WATER ENGINE OUTLET TEMPERATURE AS A FUNCTION OF WASTE HEAT VALUE AND UTILIZATION FACTOR

Advanced Technology Concepts ("B" Designs)

The emphasis in the "B" designs has been on concept generation and evaluation. The concepts reported as "B" designs represent unproven ideas which could improve the cost, performance or life of the system. Several advanced concepts are discussed in Section 3.0. The ones capable of providing the greatest impact on system capital cost are:

1. Packaged FBC and coal handling equipment
2. Engine direct-fired with FBC (no heat pipes)

Coal combustion and coal handling equipment is typically manufactured in low production quantities and field erection costs often equal or exceed factory fabrication costs. Small units produced in large quantities and designed as a "packaged system" with a minimum of required field erection work offer significant potential for cost savings. This is considered an advanced concept since such designs would not be used for a 1985 demonstration system. See Section 3.1.1.

It appears technically possible to develop a Stirling engine which is direct-fired by a fluidized bed combustion system. This possibility offers the opportunity to eliminate the sodium heat pipe heat transport system, thereby eliminating the capital and maintenance costs associated with the heat pipes. However, certain design constraints on arrangement of heater head tubes in the fluidized bed and integration with the engine require much greater study than was possible during the course of this program, particularly since major changes would be required in the basic engine module. Since this type of system has not yet been built and operated it must be considered an

advanced concept subject to possible future R&D efforts. The potential direct cost savings may be assessed by eliminating heat transport costs in Figures 1-8 and 1-9. Other cost savings in installation cost, structural requirements, better packaging and even O&M costs could result from improved system performance and greater system simplicity. See Sections 3.2.2. and 3.3.2 for more detail on the direct-fired concept.

In addition to these two concepts offering cost savings potential, there are many special application situations where the Stirling's benefits are enhanced. Some of these situations are:

1. Alternate Fuels - Cases where a low cost or "tree" by-product fuel is available (e.g. sawmill waste) in a high load factor situation.
2. Relaxed Emissions Regulations - In remote sites where NO_x and SO_x emissions are not regulated, additional benefits could accrue by reducing or eliminating consumption of bed sorbent material and increasing bed temperatures. This situation can have a favorable impact on both capital cost and operating cost.
3. Multi-Fuel Capability - The FBC system is potentially capable of burning a wide range of fuels such as coal, wood chips, municipal waste, agricultural wastes, heavy oils, tar sands, etc. in the same combustion equipment. Although the FBC system would be developed initially for coal, with further R&D it should be adaptable to many other fuels as well. This characteristic will be very attractive in the years ahead for any fuel burning equipment.

Finally, under the category of advanced concepts, several specific research topics are suggested relating to the detail design of the major components discussed above. These represent areas where further advances in state-of-the-art technology would benefit the future economic viability of a coal-fired Stirling engine. These topics are discussed throughout Section 3.

1.3 CONCLUSIONS

1. Based upon the conceptual design study and preliminary analysis, a solid-fueled, stationary Stirling engine using current state-of-the-art technology can be demonstrated in the 1985 time-frame.
2. The recommended concept is based upon fluidized bed combustion, two-stage sodium heat pipes, and a dry-sump, four-cylinder, double-acting engine module.
3. At current fuel prices, the power costs of a coal-fired Stirling engine have a slight advantage versus a comparable Diesel powerplant.
4. The Stirling powerplant economics improve with waste heat utilization and as fuel prices escalate when compared to conventional powerplants.
5. Coal combustion costs dominate the Stirling powerplant economics.
6. Further R&D efforts have the potential for significantly reducing capital and operating costs, resulting in an even more favorable economic outlook.

2. STATE-OF-THE-ART CONCEPTUAL DESIGNS

This section examines various engine, heat source, and heat transport design concepts that are capable of being built and demonstrated successfully by 1985. Considering this short time frame, it is believed that only those design concepts that are state-of-the-art (i.e., have already been demonstrated) can be considered under this category. Although the entire package under consideration has not been built and demonstrated as a system, each individual component must have been demonstrated as a separate component in order to be considered for state-of-the-art status.

The principal focus of this study was on methods for utilizing coal and other non-scarce fuels in large Stirling engines. Since the combustion systems for these fuels of interest tend to be large and not readily integrated into a compact engine package, it seemed most appropriate to study the combustion system alternatives and basic engine design alternatives separately. Then the chosen component concepts were integrated into a system via suitable intermediate heat transport means. This scheme is referred to as an "indirect fired" engine in contrast with the more conventional "direct fired" engine operating on refined petroleum fuels.

Accordingly, this section on state-of-the-art designs is divided into four major sub-sections which describe the conceptual design studies on 1) the heat source system, 2) the indirect heat transport system, 3) the basic engine system and 4) the complete system integration and performance assessment. The case of a direct-fired engine utilizing clean liquid or gaseous fuels is included in section 2.3, Engine Conceptual Design.

2.1 HEAT SOURCE

Although the program objectives stated that this conceptual design

study should include consideration of a wide range of fuel types, the primary focus of attention was directed toward coal and coal derived fuels. The direct combustion of raw (unrefined) coal within the applicable emissions constraints was considered to represent the most difficult combustion system design task, and it was believed that some of the methods available for burning coal might also be suitable for other fuels as well. Some obvious examples are various types of biomass, municipal refuse, agricultural and forest wastes, etc. For these reasons the discussion below focuses on alternate methods for coal combustion and selects the fluidized bed combustor as the one best suited for this application. Following the selection, a conceptual design for an FBC system is presented and an estimate of the cost for this combustion system is developed.

2.1.1 Initial Choices

Examining the state-of-the-art in coal combustion technology, five choices for the heat source were selected for evaluation. All five systems are commercially available, but not necessarily in the sizes required for the Stirling engine. However, all five pass the test of having potential for demonstration in 1985.

The five systems are the following:

- a) Stoker - Characterized by traveling grate coal feed, this approach is the dominant method used in industry today for the smaller installations.
- b) Pulverized - Firing small particles of coal in a spray similar to oil sprays, utilities have preferred this approach, especially for large systems.

- c) Cyclone - Utilizing a very intense vortex chamber, cyclone combustors achieved a small share of the utility applications.
- d) Low Btu Gas - Proven coal conversion methods presently available are included in this category.
- e) Fluidized Bed - Just beginning to make an appearance in the commercial market, FBC holds promise over a wide size range.

Other coal combustion technologies presently in the laboratory or pilot stage were not considered suitable for state-of-the-art conceptual design evaluation.

2.1.2 Selection Criteria

Five criteria were applied in evaluating the coal combustion options: emissions, fuels flexibility, size flexibility, efficiency, and cost. Discussion of each criterion follows.

Emissions

Emissions requirements are not well defined in statutory terms. The national standards for stationary sources apply only to installations over 250 million Btu/hr (1). The proposed Stirling application, even in multi-engine installations, will be well below 250 million Btu/hr. However, local standards exist which must be considered for local power generation even in small installations.

Each local air quality district, under the direction and enforcement of the EPA, must develop a plan to meet certain air quality standards. Consequently, the specific allowable pollutant levels from a given installation may vary considerably from one local environment to another. For example, Boston air quality regulators enforce an SO₂

standard which is less than one fourth of the national standard. Oxides of nitrogen, however, are uncontrolled. The local standards may or may not apply to specific size sources. In some cases emissions levels are assigned on a unit-by-unit basis (2).

Lacking an objective set of emission targets, we must invoke the principle of "best available technology" for the purpose of this study. Each combustion system is evaluated here on the basis of controlling emissions to the level of the state-of-the-art. This criterion implies that no credit may be taken for possible use of low sulfur coal, and that in the cost evaluation, stack clean-up must be included.

Fuels Flexibility

The fuels flexibility criterion is primarily intended toward the use of a wide range of coals. A secondary consideration is the capability of burning biomass or other alternative fuels. All candidate systems are capable of using oil or gas back up at reasonable additional cost. Since coal properties vary widely, one of the first considerations that a designer of a coal-fired system must employ is the coal type. The coal properties are critical in the design of material handling, combustion, heat transfer, and stack clean-up equipment. Thus, typically an installation is tailored to burn a specific coal type. This leads to high cost "engineered systems". Combustion systems which are relatively insensitive to coal properties are therefore desirable in the design of the Stirling system, particularly since the Stirling system is intended for eventual mass production of a single design.

Size Flexibility

Size flexibility is a requirement for the Stirling engine, since the program is directed to engines in the 500 to 3000 hp range. Also, multi-engine installations must be considered, along with the effect on

the combustion system design. Thus, combustion systems which are technically and economically feasible in the smaller sizes, and are capable of accomodating multi-engine installations are most desirable.

Efficiency

Efficiency of the combustion system is of prime importance as a screening criterion rather than a ranking criterion. From an overall system point of view, in order for the system to be economically viable, it must be possible to achieve an overall (heat-to-engine/coal-heat) efficiency of 80% or better. Improvements over 80% will be determined primarily by design details not included in the conceptual design (such as economic optimization of preheat temperature, etc.). Note that in order to achieve 80% efficiency, the combustion system must be capable of accomodating a high degree of air preheat. This is because the high heater head temperature of the Stirling results in an exhaust temperature from the heater head of over 1500°F. Thus, the remaining heat energy in the combustion gases must be recovered by a regenerative air preheater.

Cost

Cost is ultimately the most important criterion, and the most difficult to evaluate. The combustion system which is potentially the lowest cost in production quantities, rather than as one-of-a-kind engineered systems, is the most desirable.

2.1.3 Discussion

A summary of the five candidate heat source systems and their essential characteristics according to the selection criteria is shown in Figure 2-1. Following is a brief review of each of the candidate systems.

SYSTEM	EQUIPMENT DESCRIPTION	FUELS	EMISSIONS	TURNDOWN	CONVERSION EFFICIENCY	SIZING	CAPITAL COST	COMMENTS
FBC	1) Coal & Limestone receiving & handling 2) Fluidized Bed 3) Preheater and/or Steam Generator 4) Cyclone Separator 5) Baghouse 6) Ash & Limestone Disposal	gas, oil, coal(any) wood, garbage, waste.	Meets all existing Federal Standards. Local standards might be restrictive. Spent limestone disposal might be small problem.	Reported in literature up to 3.5:1 For Stirring applications will probably be 2:1	Combustion >95% Total preheat is possible, provided high temperature distributor plate can be designed.	Beds suitable for 500-3000 hp are proven. Conducive to modular approach.	\$175 - \$350/kw total installed cost including auxiliaries	1) Auxiliaries - Approximately 5% of shaft power. 2) Possible tube corrosion problems. 3) Heat transfer not totally predictable, especially with fin tube. 4) Load control not instantaneous. 5) Direct fired might be possible as advanced concept. 6) <u>NO TUBE SLAGGING OR FOULING.</u>
LOW BTU GAS	1) Coal receiving & handling 2) Sorbent receiving & handling 3) Gasifier 4) Combustor w/ provisions for cleaning (direct fired) 5) Preheater and/or steam generator 6) Wet scrubber 7) Ash disposal 8) Acid sludge disposal	medium to low sulfur coals, possibly wood.	NO _x can be controlled. Otherwise ok with wet scrubber. Acid sludge disposal a problem.	little to none	Coal to gas <90% Combustion <90% Total preheat is possible.	<u>RILEY:</u> 45-90 x 10 ⁶ Btu/hr depending on the coal	<u>RILEY:</u> Gasifier \$950,000 Wet Scrubber \$250,000 (rough estimate from Riley)	1) Physically large. 2) Heater tube slagging & fouling could be problematic.
STOKER	Items 1, 2, 5, 6, 7, 8 above in low BTU gas. Also requires: 1) Stoker & Spreader 2) High Temp. Heat Exchanger w/provision for cleaning	coals, other solids.	NO _x problematic. Wet scrubber required. Acid sludge disposal problem.	Large Ratio	Combustion: High Little to no preheat (450°F max). Must make steam.	Stirling sizes available in boilers. However nothing available at high temperature.	Low, except wet scrubbers usually expensive.	Inability to preheat above 450°F and tube slagging rule out the stoker. NO _x is a problem and wet scrubbers are a problem.
CYCLONE COMBUSTION	Same as stoker except: 1) no stoker 2) Cyclone combustor	coals, other solids.	NO _x prohibitive. Wet scrubber required. Acid sludge disposal problem.	Not investigated	High	Large	High	Not practical in small sizes
PULVERIZED COMBUSTION	Same as cyclone except 1) no cyclone 2) Pulverized combustor 3) Coal preparation OR different coal handling with safety equipment	coals	NO _x problematic. Wet scrubber required. Acid sludge disposal problem	Not investigated	High	Large	High	Not practical in small sizes

FIGURE 2-1: COAL COMBUSTION COMPARISON

Stoker

Below the utility size level, stoker fired coal combustion is the dominant system in use. As the size goes down, stokers enjoy an even greater advantage over pulverized and cyclone systems, primarily for reasons of first cost, and despite a 4 to 7% efficiency penalty (3). Emissions performance is acceptable only if scrubbers for SO₂ and particulates removal are employed in the proposed application (no guarantee of low sulfur fuel).

Fuels flexibility with stokers is good. The stoker is generally useable with most coal types, and applicable to refuse burning under certain conditions (4). Coals need to be screened to a given size range for best operation.

Size flexibility with stokers is excellent, since they scale directly in terms of grate surface area. Travelling grates may be assembled in virtually any width and length within relatively flexible limits. However, like all coal fired systems, the coal handling, stack scrubbers, and ash disposal systems tend to be most economically applied in large systems.

Stokers are capable of acceptable combustion efficiencies and heat recovery efficiencies in the 80% range when applied to steam boilers. In the Stirling application however, approximately one-half of the combustion heat is transferred to the engine directly, while the other half must be recovered in the form of air preheat. Consequently, air preheat temperatures in the 1300°F to 1500°F range are required. Air preheaters are not recommended for stokers above 450°F due to coal caking problems (5). Therefore the heat-to-engine/coal-heat efficiency would be prohibitively low (e.g. 40-60%).

Of all the "conventional" coal combustion systems, stokers are the least expensive. The requirements for SO₂ scrubbers, however, can

nearly double the cost of a coal fired boiler (6). For example, a 10 million BTU/hr installation would require approximately \$500,000 in scrubber costs (1976) (3).

Pulverized

Emissions performance of pulverized coal combustion systems is acceptable only if SO₂ scrubbers and high efficiency cyclones and dust collectors are employed (3). The nature of the combustion process is such that very fine particulate fly ash is produced, thus requiring high dust collection efficiency. Oxides of nitrogen tend to be a problem due to the high combustion temperatures, but control methods show promise of acceptable performance.

Fuels flexibility is rather poor, since pulverizers are sensitive to coal rank. Other potential fuels such as refuse and biomass can be used only in limited mixtures (4).

Size flexibility is also rather poor since the pulverized systems are rarely used in sizes under 200 million Btu/hr. If pulverized coal is not already available at the site, small-scale pulverization may be impractical and economically unattractive. For example, the smallest unit listed by the Riley Stoker Corporation is suitable for 100 million Btu/hr (7).

Efficiency of the pulverized coal heat source is acceptable. Full air preheat may be employed in principle.

The cost of a pulverized coal system suffers from the same scrubber requirement as the stoker; plus it requires more efficient ash and dust collection and the additional costs of pulverization. It is only when the system is rated at over 200 million Btu/hr that the pulverized system becomes attractive relative to the stoker (3).

Cyclone

Cyclone coal combustors were developed primarily for their high heat release characteristics (20 to 40 times pulverized) and capability to use high ash content, low fusion temperature coals. With the advent of stricter emission standards (mainly NO_x), cyclones disappeared from the market. The last unit was sold in 1973. The two basic problems were: a) it could not burn low sulfur coal due to the high ash fusion temperature which is typical with that type of coal, and b) despite great efforts, oxides of nitrogen remain a serious problem. An excellent summary is given in reference 8.

The other criteria were not examined in any detail due to the serious emissions problem. Current laboratory research may promise a solution to the problem, but cannot be classified as state-of-the-art.

Low Btu Gas - Conventional

Proven gasifiers such as Wellman-Galusha and Riley-Morgan are attractive candidates for the Stirling application since they would allow for direct firing of the heater head. With the high air preheat temperatures available, the gas can be readily burned. In effect, the indirect heat transport system is the combustible gas itself, transporting chemical energy from the coal in the gasifier to the engine.

Emissions are acceptable with suitable stack clean-up, which tends to be less costly than that for direct coal fired systems because of reduced particulate loading. Sulfur, present in the low Btu gas as H_2S , appears as SO_2 in the stack and requires scrubbers. High capital cost and lowered efficiency (since the gas must be cooled) rule out the possibility of removing the sulfur before combustion of the gas (9). Oxides of nitrogen are not expected to be troublesome, since the flame temperature can be held quite low. Condensible tars in the gas stream can be burned directly along with the gas. It appears that the Stirling

should be able to burn the low Btu gas directly as it is generated by the gasifier, assuming that tube fouling is not an insurmountable problem. Methods of scrubbing particulates from the hot (1200°F) gas stream that need to be developed for gas turbine applications could possibly solve the tube fouling problem (35). It is important to note that the Stirling heater head is likely to be less sensitive to trace amounts of ash in the gas stream than are the gas turbines.

Fuels flexibility is limited. The gasifier is usually set up for specific coals, although it is modifiable for other coals. However, different gasifiers are available which gasify everything from wood to anthracite culm.

Size flexibility is also limited - primarily by availability. Available units would be suitable for a 15 MW(e) multi-engine installation at the low end (9). It is possible to design smaller units; however to date little or no market exists for such equipment.

Efficiency is acceptable so long as the gas can be burned hot and dirty (i.e. direct from gasifier with no sulfur clean-up which requires gas cooling before combustion). Coal to gas efficiency of 90% is possible. Full air preheat may be utilized; however, the cost of a sufficiently effective preheater may be high.

Cost-wise, the low Btu gas system is not unreasonable. The gasifier for a 15 MW(e) installation is priced on the order of \$70/kw(e) (9). Of course added to the figure must be the cost of a hot particulate separator and/or tube cleaning system, the combustor, and stack clean-up equipment (wet scrubbers most likely).

Gasification of coal in a fluidized bed shows some promise for overcoming the disadvantages of conventional gasification. However,

this technology is not considered to be state-of-the-art nor does it show much potential for cost reduction over fluidized bed combustion. For these reasons, it will be dealt with in Section 3.0, Advanced Concepts.

Fluidized Bed Combustion

Although not widely in use as yet, FBC has just become commercially available and is rapidly gaining favor as a practical coal combustion technique, particularly in the smaller sizes. For the Stirling conceptual design, the FBC (atmospheric) is considered state-of-the-art, as units are now commercially available.

Low SO_x and NO_x emissions make the FBC an attractive candidate for this application. With the appropriate sorbent material in the bed, SO_2 is controllable. The amount of sorbent required per pound of coal burned depends on many factors including the bed temperature, sorbent particle size, % sulfur in the coal and the sorbent properties. However, for reasonable particle sizes, a typical limestone is capable of providing high sulfur capture (>90%) at temperatures as high as 1650°F (18, 36, 37). With low bed temperatures (1500-1650°F) oxides of nitrogen are not a problem. A hot gas cyclone is utilized for recycle of unburnt carbon, and a baghouse is probably required for particulate control of stack gases.

Fuels flexibility is excellent compared to other coal systems. Coals of virtually any composition, biomass, and refuse may be utilized (10).

Size flexibility also appears good. FBC boilers are commercially available in the firing rates required by the 500 to 3000 hp Stirling (11) and are under serious development for utility size installations (18). Power scaling of FBC systems is typically by bed cross-sectional area, however, bed size can have a significant influence on the bed hydrodynamics. For this reason and others, fluidized bed designers have often gone to the modular approach, i.e. installing several

modules of a smaller system rather than using one large system.

Efficiency is acceptably high when particulate recycle is used. Full air preheat is possible; however, the use of full air preheat has not been proven because of potential distributor plate problems.

FBC enjoys the important cost advantage over all other coal combustion systems of requiring no stack clean-up other than for particulates. When high sulfur coal is considered, FBC has the edge on costs over conventional firing (12).

The main limitations of FBC relate to turndown, load following and possible corrosion and erosion. Turndown range is limited by elutriation of bed material at high loads, and loss of fluidization at lower loads. Turndown ranges from 2:1 to 5:1, with 3:1 commonplace. Bed temperature time constants for reaction to step changes in load range from one minute to twelve minutes depending mainly on coal feed size. Information on corrosion in FBC's at the high temperatures of interest is limited. Some investigations have reported serious corrosion problems, particularly under alternating reducing/oxidizing conditions or when using high nickel alloys. Others have reported excellent results. The opinion of most is that with proper material selection and FBC design, normal tube life should be achieved.

Tube Fouling

An important combustor characteristic not included among the five previously described selection criteria relates to the relative tendencies of the products of combustion to foul the heat exchanger surfaces with fly-ash or slag. Here we are concerned with both fouling by fly-ash, which is readily removable by soot blowing, and with fouling by solidification of molten slag, which is not readily removable. While the various conventional combustion techniques all produce different qualities and amounts of fly-ash, fly-ash fouling is easily handled by soot blowing and so was not considered as a criterion for ranking

alternative combustion techniques. However, slag deposits are not readily cleanable, and thus their prevention is critical to the heat exchanger design.

It is the practice in some conventional boilers to employ radiation heat transfer to cool the combustion products below the ash-fusion temperature before passing the hot gases over convective heat transfer surfaces. Thus, any entrained slag solidifies before reaching the heat exchanger surface. In the intended application, however, this practice may not be practical due to the high working gas temperature of the Stirling engine which is only a few hundred degrees below typical coal ash-fusion temperatures (2000°F).

For the Stirling application, a possible method of cooling the combustion gases below ash fusion temperature before passing over the heater head tubes would be gas tempering (13). This involves recirculating cooled combustion gas and thus reducing the temperature of the mixture before reaching the convective tube bank (See Figure 2-2).

For steam generating power plants this practice presents little heat absorption penalty. The gas is tapped after the economizer and before the air preheater, thus maintaining approximately equal mass flow on the two sides of the preheater.

However, for the Stirling application where the majority of the heat in the flue gas goes to preheat of combustion air, the tempering air would have to be tapped after the preheater, thus causing grossly unequal mass flow on the two sides of the air preheater. This means for a preheater of a given effectiveness, the stack gas will not be cooled as completely, resulting in increased stack heat loss and lower system efficiency. See Figure 2-2 for a flowchart of the two cases.

GAS TEMPERING IN A STEAM GENERATING APPLICATION

GAS TEMPERING IN THE STIRLING APPLICATION

FIGURE 2-2: GAS TEMPERING

This loss might be reduced somewhat by employing a two stage preheater where the tempering air is drawn off after the first preheater. Even with this design, the increased cost and complexity of a two-stage preheater would have to be traded off for decreased stack loss.

Finally, the use of gas tempering would provide at most a few hundred degrees temperature difference between combustion products initially at 1900°F-2400°F and the heat exchanger at 1300°F to 1500°F, resulting in low heat transfer rates and a large, costly heat exchanger.

FBC is an exception to this general tube slagging problem and the complex solutions which are associated with it since the entire bed operates below the ash-fusion temperature. For this reason, there is no possibility of slagging the heat exchanger surfaces. Moreover, the scrubbing action of the fluidized bed material is highly effective in keeping the heat exchanger surfaces clean, thus precluding the need for any soot-blowers. Therefore, with regard to both slagging and tube fouling, the FBC is far superior to the other combustion systems.

2.1.4 Selection

Based upon the evaluations discussed so far, fluidized bed combustion is the clear choice for a coal-fired Stirling. The main reasons behind this choice are the low emissions without the need for flue gas desulfurization, the excellent fuels flexibility, size flexibility, relatively high efficiency, and costs which promise to be competitive with the least expensive conventional coal-fired systems. Most important however, is the fact that FBC is the only method which permits direct utilization of products of combustion without fear of tube slagging or fouling.

On the other hand, FBC is not yet widely practiced, and the present cost estimates for commercial units may prove to be optimistic. In addition, it is not yet clear whether FBC may have corrosion/erosion

problems, the distributor plate design could be a problem with full air preheat, and it is possible that other long-term operating problems may not have been identified. Nonetheless, on balance, FBC is the most attractive choice.

The second choice is low Btu gasification. Technical requirements may be met at reasonable costs for multi-engine installations, and this approach makes possible a relatively simple direct-fired heater head, thus simplifying the engine design. However, with conventional gasification technology, scrubbers for SO_x capture must be used in order to meet emissions standards. This adds the cost of the scrubber and the cost of sorbent removal, which in the case of wet scrubbers involves disposal of an acidic watery sludge.

2.1.5 Conceptual Design of FBC System

While fluidized bed combustion (FBC) is considered state-of-the-art for purposes of this program, there is no existing fluidized bed which could be used as-is for the Stirling application. This is due to several unique features of this application which will make the fluidized bed different.

First, FBC combustion temperatures are higher and in-bed heat exchangers are hotter than for typical FBC boilers. This is because the Stirling engine heater head is required to operate at high temperature (about 720°C) and the heat transport system imposes an additional 30°C or more, for a resultant heat exchanger temperature of about 750°C . Even with a bed temperature of 900°C (1650°F), this leaves only about 150°C available for bed-to-tube heat transfer. Heat exchanger and FBC design analysis shows that the heat flux to the tubes is limited on the combustion side. Thus, the heat exchange surface area required is larger than usual because of the low temperature difference between the bed and the Stirling heater head tubes. First law analysis requires the tubes be closely packed and

also requires the bed heat release rate per unit cross sectional area be kept below a maximum for a given tube packing arrangement. Thus, not only is a different heat exchanger required, but bed fluid dynamics may be affected as well. Furthermore, this shows that the bed temperature should be as high as sulfur capture limitations will allow (1650°F, 900°C).

Secondly, in order to achieve high efficiency heat input to the engine, full air preheat must be utilized. This affects most significantly the design of the air distributor plate.

Finally, the economics of the system requires that installation costs be minimized. This implies a "packaged boiler" concept, mass produced, factory assembled and shipped to the site. While such a design is not necessarily desirable for an initial prototype, it is important to demonstrate a system which could be modified and eventually turned into a mass produced system.

A conceptual layout of the fluidized bed (minus heat exchanger) is shown in Figure 2-3. The layout schematically shows all of the essential components and the function of the unit as a whole.

Bed Size and Firing Rate

This particular size fluidized bed would be suitable for firing a 500 hp engine module. Multi-module engines would be fired by multi-module fluidized beds. The total full load heat input to the Stirling heater head would be 4.03×10^6 Btu/hr ($1.18 \text{ MW}_{\text{th}}$) per 500 hp module. If the overall heat input efficiency (heat to heater head $\div$ coal higher heating value) is 83%, then the total firing rate is approximately 4.9×10^6 Btu/hr ($1.42 \text{ MW}_{\text{th}}$). With a 3'-7" x 3'-7" (1.1m x 1.1m) cross section and a 1 m active bed height, it will be possible to pack the bed with sufficient heat exchanger tubes to transfer the required heat at the required temperature. This will be discussed in more detail below and in Section 2.2.5, Heat Pipe Conceptual Design. Furthermore, this size bed is conducive to

FIGURE 2-3: FLUIDIZED BED COMBUSTOR

ADVANCED MECHANICAL TECHNOLOGY, INC. <small>741 CALIFORNIA STREET • NEWTON, MASSACHUSETTS 02154</small>				☐ AT ☐ DATE		☐ REVISION ☐ ECN		☐ APPROVED ☐ REV	
UNLESS OTHERWISE SPECIFIED				DRAWN		PROJECT NO.		NO. OF SHEETS	
MATERIAL -				CHECKED		NEXT ASSY NO.		DRAWING NUMBER	
NAME				APPROVED		SHEET NO.		REV	

placement of several identical FBC's side by side as is required for the multi-module engine. See Section 2.4.1, System Layout, for an illustration of a three module (1500 hp) engine-FBC system.

Superficial Velocity and Particle Size

The bed is 3'-7" x 3'-7" (1.1m x 1.1m) in cross section, which (after allowing for heat pipe area) gives an area heat release rate of 0.44×10^6 Btu/hr-ft² (1.4 MW/m²). The superficial velocity with 20% excess air at this firing rate is approximately 6.1 ft/sec (1.9 m/sec). This is lower (more conservative) than the design superficial velocity often reported in the literature (14). This is due in large part to the reduced area firing rate which is required due to limits in tube packing. This means that the particle size will have to be rather small to ensure that the bed is operating sufficiently above minimum fluidization velocity to assure good fluidization. A mean particle size of 0.048 inches (1.2 mm) gives a minimum fluidization velocity of 1.8 ft/sec (0.55 m/sec) at 1500°F (815°C). Thus, full load superficial velocity is approximately 3.4 times minimum fluidization velocity. The full load superficial velocity also corresponds to the terminal velocity of a 0.008 inch (0.2 mm) limestone particle. It is important to keep in mind that the particle size of the bed is essentially the particle size of the sorbent because the coal is only a small percentage of the bed material. In fact with the overbed feed scheme under consideration, larger coal sizes are desirable.

The conclusion to be drawn here is that despite the reduced heat flux to the tubes, it is possible to design a fluidized bed with particle size and superficial velocity in the lower range of conventional practice. The final design values for superficial velocity and particle size will be determined in later phases of the program.

Heat Transfer Characteristics

To date, the heat transfer coefficient in fluidized beds has to a large extent escaped researchers' attempts at correlation. Hartnett and Irvine (15) give 34 correlations by different authors for Nusselt numbers under various conditions. Consequently, in previous development programs serious errors have been made in prediction of heat transfer coefficients (16). Based on trial of a few published correlations as well as conversations with experts in the field, a preliminary estimate for the convective component of the heat transfer coefficient was determined to be $40 \text{ Btu/hr-ft}^2\text{-F}$ ($0.023 \text{ W/cm}^2\text{-C}$) at the velocities of interest.

With regard to heat transfer, the only real difference between conventional steam generation in fluidized beds and the Stirling application is the significance of radiation. In conventional FBC steam generation equipment, the log-mean temperature difference between the bed and the heat exchanger tubes is much larger than for the Stirling application, due to the higher temperature Stirling heater head. Thus, in a conventional FBC system, the large temperature difference drives high convective heat flux rates, and the radiation contribution is minor. However in the Stirling application, the log mean temperature difference between the bed and the heat pipes is relatively small, thus the convective heat flux is small compared to conventional practice. Because the radiative heat transfer coefficient, h_r , is approximately proportional to the mean heat exchange temperature to the third power (i.e., $(T_{\text{bed}} + T_{\text{tube}})^3$), h_r increases as the tube and bed temperature increase (38) (Even though the actual heat flux, Q/A , decreases with increasing tube temperature.) For an assumed bed temperature of 1650°F (900°C) and a tube temperature of 1400°F (750°C), the equivalent heat transfer coefficient due to radiation, h_r , was calculated to be approximately $30 \text{ Btu/hr-Ft}^2\text{-F}$ ($0.017 \text{ W/cm}^2\text{-C}$).

Thus the total heat transfer coefficient was estimated to be 70 Btu/hr-ft²-F (0.04 W/cm²-C). Tube arrangement and surface area are discussed in Section 2.2.5, Conceptual Design of Heat Pipes.

Freeboard.

The heat pipes will extend down vertically into the fluidized bed, as will be seen in Section 2.2.5, Conceptual Design of Heat Pipes. Thus, a higher freeboard requires longer tubes. The longer the tubes, the more material is required, and the stiffer the tubes need to be. Thus cost is increased.

Furthermore, the higher the freeboard, the larger the package. This increases shipping size and weight. Because we must design for an eventually packaged boiler concept, this is undesirable.

Last but not least, because the engine and generator mount on top of the fluidized bed, increased freeboard height increases structural requirements of the mounting platform or building. This also affects cost.

With all these good reasons to reduce the freeboard height, it is important to note the consequences of decreasing freeboard height. If the freeboard is too low, excessive particulate loading in the cyclone will occur. In addition, the freeboard is important from a chemical kinetics point of view with regard to NO_x destruction and CO burnup.

The preliminary freeboard height shown in Figure 2-3 (1 meter) was selected on the basis of engineering judgement. Insufficient time was available during this study to allow the trade-offs to be analyzed in enough detail to determine the optimum. The final design height will have to be determined in later phases of the program.

Particulate Recycle

The purpose of particulate recycle is to capture unburnt carbon and sorbent which is elutriated from the bed. Several researchers have shown that this is an effective way to maintain high carbon utilization (17) and to increase sulfur capture efficiency (18).

Thus, particulate recycle is likely to be necessary to maintain high combustion efficiency. In light of the preceding section on freeboard height, it is obvious that it is better to utilize particulate recycle than to increase freeboard height sufficiently to eliminate the need for it.

Coal Feed and Ash Removal

Again, time has not allowed detailed investigation of this area. The conceptual layout (Figure 2-3) shows over-bed coal and limestone feed and underbed ash removal. At least two commercially available fluidized bed combustors use this approach. Also, a qualitative look at the possibilities indicates that this is probably the simplest, and cheapest approach. Furthermore, underbed ash removal allows some degree of bed depth control, which will possibly be one means to turndown the FBC. However, should this control requirement be eliminated (for example by requiring entire modules to be shut down for turndown) then a simple fixed weir might be better than underbed ash removal. Again, this would be reviewed in later design phases of the program.

Probably the most important thing to note here is that there is nothing particular in the Stirling application which for technical reasons requires the feeding to be any different from conventional practice.

Conclusions

Based on the conceptual work done to date on the FBC system, some tentative conclusions may be drawn. In particular, the tube packing and heat flux limitations will dictate the maximum firing rate per unit of bed area. Bed temperature will be limited by sulfur capture considerations but will likely be higher than the optimum from sorbent utilization considerations, thus requiring greater use of sorbent. Radiation heat transfer will be important. Low freeboard height is desirable. The superficial velocity will probably be less than that reported for conventional steam generating equipment and, as a result, the sorbent particle size will probably be slightly smaller than for conventional steam generating equipment. Full air preheat will dictate a new distributor plate design.

The estimates which were generated for some of the important design parameters (heat transfer coefficient, superficial velocity, bed size, freeboard height, particle size, etc.) are not intended to serve as design specifications. Rather, they are merely intended to show that it is feasible to design an FBC which interfaces with the proposed heat transport system as described in section 2.2.5.

2.1.6 Recommended Supporting Technology

Whenever an old idea is applied in a new way some problem areas will arise which require development. The FBC applied to the Stirling is no different. As a result of the work behind Sections 2.1.1 through 2.1.5 of this report, we have identified the parts of the system which we feel offer the highest technical risk. Some of these areas of higher technical risk are peculiar to the Stirling application and some are of a more general nature, applicable to most fluidized bed combustors. However, even where problems are peculiar to the Stirling application, solutions to these problems could open up new applications and markets for fluidized bed combustion in general.

Problems Peculiar to the Stirling Application

The areas identified as higher risk but peculiar to the Stirling application are tube corrosion, tube layout, radiation heat transfer and distributor plate design. These areas are of course important in conventional FBC design, but in the Stirling application the problems are slightly different than in conventional applications. The following paragraphs explain each problem.

Tube Corrosion - Tube corrosion is important in any boiler design. However, in the Stirling FBC application the tubes must not only resist exterior corrosion and erosion, but also they must be compatible with liquid sodium on the inside, and they must retard hydrogen diffusion into the heat pipes. Since an external hydrogen diffusion barrier is envisaged (see Section 2.2.5, Heat Pipe Conceptual Design), it would be endangered by the tube corrosion/erosion.

Tube Layout - Section 2.1.5, Conceptual Design, pointed out that the heat release rate per unit of bed cross sectional area is limited by the heat flux to the heat pipes. Thus, closely spaced tubes should allow higher heat release. However, if the tubes are too closely spaced, the fluid mechanical, heat transfer and mixing properties of the bed could be seriously affected. The object of tube layout design in the Stirling application is to provide the maximum heat transfer area per unit bed cross sectional area without causing gas bypassing, slugging, etc.

Radiation Heat Transfer - In conventional steam generating FBC design, radiation is not very important. As pointed out in Section 2.1.5, Conceptual Design of FBC, radiation is very important in the Stirling application. The limited work to date on radiation heat transfer in fluidized beds does not provide us with an accurate correlation for the prediction of bed emissivity.

Distributor Plate Design - The distributor plate design problem was noted as being an area which could benefit from supporting technology programs. Past FBC demonstration programs have experienced difficulties with the distributor plate. The B&W/EPRI fluidized bed (18) is one example. The authors of reference 18 felt that the solution to their problems was a water-cooled bubble cap distributor. The Stirling application cannot use a water cooled distributor however, because of the efficiency requirement of full air preheat. On the other hand, in reference 18, the author implies that the cause of the problem is thermal expansion. It might be the case that air preheat will promote uniform thermal expansion and thus would not give rise to large thermal stresses. Furthermore, the B&W facility is 6' x 6', and the Stirling FBC is considerably smaller. This, too, would tend to lessen the severity of the problem.

The experience of Philips Research Laboratories is also of interest (19). Philips built and tested a small (10 hp) FBC-heat pipe-Stirling engine. They used a sintered metal distributor plate with air preheat temperatures of 600°C. Unfortunately, they only ran the facility for 50 hours before writing the report. Furthermore, sintered metal distributor plates typically have large pressure drops across them. This increases the power used by the combustion air fans.

Generally Applicable Areas for Development

Ash removal is one area of higher technical risk in the state-of-the-art FBC design which is probably of interest to designers of all types of FBC's. A fixed wiper is a viable device to remove ash and to control bed depth. However, the depth can only be controlled to one operating level. If ash is removed from the bottom of the bed then active depth control is possible. This might be important for the Stirling application from a load-following point of view. However, ash removal from the bottom of the bed requires an ash removal valve which can provide flow control of both bed material and air. Furthermore, this valve must operate at bed temperature, and is subject to erosion

and corrosion by the coal ash and the spent sorbent.

Recommended Approach

Because the previously mentioned areas for development are of vital importance to successful demonstration of the large coal fired Stirling engine, it is recommended that the development work in these areas be under the control of the prime contractor responsible for the demonstration of the engine. Furthermore, it is recommended that a smaller, less expensive demonstration program be used as a development tool to resolve these and other problems.

2.1.7 Heat Source Cost Estimates

For the purpose of the conceptual design costing exercise, several assumptions are necessary to estimate the appropriate costs chargeable to the heat source.

a) The FBC basic design as envisaged has all the major elements similar to commercially available designs. The bed operating temperatures, fuel/sorbent feeds, ash disposal, air flows, carbon reinjection, and other design features are within the range of present design practice.

b) Cost estimates are considered "budgetary" in nature, and do not reflect potentially important site-specific or application-specific criteria. In commercial trade, these type estimates are appropriate for initial screening at the conceptual design level.

c) The heat source cost estimate is broken into three categories: the heat source base equipment, the auxiliary equipment, and installation for both of the above.

d) The heat source base equipment includes the FBC firebox,

small coal and sorbent hoppers, coal and sorbent feeding equipment (including controls), ash removal from the bed, reinjection cyclone, preheater, F.D. fan(s), I.D. fan, and stack. This is all of the heat source equipment shown inside the building in the overall system layout (See Section 2.4.1).

e) The auxiliary equipment includes coal and limestone storage and transporting equipment, baghouse, and ash storage and transporting equipment. Simply stated this is all of the equipment outside of the building in the overall system layout.

f) For purposes of this estimate, the estimates obtained for FBC steam boilers are used intact. This assumes that the steam generator cost is approximately equal to the cost of an air preheater.

Two approaches to system design are possible for installations of this type. An "engineered" system requires custom design and field assembly of plumbing, controls, etc. for each installation and may be considered "one-of-a-kind". Most large boiler systems are engineered and field erected component by component. "Packaged" systems are pre-engineered and serially produced, requiring only specification of the particular model and its field connections. Delivery to the site is usually as assembled by the factory. Small commercial/industrial boilers are usually packaged types.

Budgetary estimates were obtained from four sources in order to obtain reasonable confidence in the numbers. The four sources, called A, B, C, D are described below:

Source A: Manufacturer specializes in FBC systems suitable for hot air generation. The capacity of the unit on which the estimate is based is 15 million Btu/hr output. It can properly be described as an "engineered" system as opposed to a "packaged" system.

Source B: Manufacturer specializes in FBC steam boilers for commercial/light industrial use. The estimate is based on 20 million Btu/hr capacity, and is a true "packaged" design. For the Stirling application, the packaged concept is advanced and is discussed in Section 3.1.

Source C: Engineer/Contractor firm engaged in FBC systems -- published data (20). The estimate is based on 100 million Btu/hr capacity. The unit is an engineered system.

Source D: Manufacturer specializing in material handling systems, does not manufacture the basic FBC system, but purchases a packaged FBC from an outside source. Material handling system estimates are based on 15 million Btu/hr capacity.

To place all the estimates on a common basis, and to facilitate easy comparison to other commercially available boilers, the costs are presented as \$/kw of heat output. (For conventional boilers this is taken as steam output. 1000 lbs/hr steam = 293 kW.)

Three of the four sources recommended estimating installation costs at 100% of equipment costs. Admitting that this "rule-of-thumb" is highly approximate, they nonetheless suggested that to be safe, it is a reasonable figure to use. For the two engineered FBC units (A and C) and for all of the material storage and handling, 100% of equipment cost is probably reasonable. For "packaged" FBC units alone, however, 100% of equipment cost is probably high. Nonetheless, for Figure 2-4, 100% of equipment cost is used.

The calculation results are given in Figure 2-4. As might be expected, the systems which include the "packaged" FBC (B and D) are lower in cost than the engineered systems. For the purpose of a current technology demonstration Stirling installation, systems A and C would be the most appropriate cost estimates. For advanced systems a packaged FBC estimate might be more appropriate. Thus, the

FIGURE 2-4: HEAT SOURCE COST ESTIMATES

(all figures in \$/kW of heat output)

	<u>A*</u>	<u>B**</u>	<u>C*</u>	<u>D**</u>
Base Equipment	77	43	70	43†
Auxiliary Equipment	94	44	76	63
Installation	<u>171</u>	<u>87</u>	<u>146</u>	<u>106</u>
Totals	342	174	292	212

* Engineered FBC systems with field erection

** Packaged FBC systems

† FBC cost assumed same as B.

equipment costs of sources B and D and 50% installation cost is reasonable for mature technology projections.

2.2 HEAT TRANSPORT SYSTEM

When heat for the Stirling engine is supplied by combustion of solid fuels such as coal, refuse, etc., or by residual oils or other types of heavy fouling fuels, the allowable heat transfer coefficient between combustion gases and heater head tubes is quite low due to tube fouling characteristics (21). This condition is undesirable for direct fired engines because it adversely affects the heater head void volume. In this situation it becomes desirable to utilize an indirect heat transport system which is capable of interfacing with the combustion gases through a large surface area (low heat flux) to contend with the fouling conditions while simultaneously interfacing with the engine heater head tubes through a small surface area (high heat flux on engine side) to satisfy the need for minimum void volume in the heater head tubes. In this section several candidate heat transport schemes are considered before selecting the two-stage heat pipe as the one best able to satisfy this requirement.

2.2.1 Initial Choices

The initial approach considered the heat pipe, a liquid forced convection loop, and a gaseous forced convection loop. In addition, however, it was decided to keep an open mind to the possibility that some combustion systems might be amenable to direct firing, and thus would not require an indirect heat transport system at all. Both the FBC and low Btu gas concepts seem to have the potential for direct firing.

2.2.2 Selection Criteria

The "best" selection criteria are not immediately obvious. Each of the systems has its own set of problems, advantages and disadvan-

tages. Furthermore, this difficulty is compounded by the fact that the heat transport systems in question are not yet designed. Thus, some initial design investigations must be done on each of the candidate systems. Nonetheless, in order to make an intelligent decision one must compare all systems on an equal basis. For this purpose, the following points for comparison are used:

- 1) Safety
- 2) Parasitic power requirement
- 3) Field serviceability
- 4) Heater head void volume
- 5) Technical risk
- 6) Effect on engine configuration
- 7) Materials cost
- 8) Manufacturing cost
- 9) Repair and maintenance cost.

A valid selection procedure requires more than several points for comparison. It is necessary to rank their relative importance in order to arrive at a final ranking of the systems being compared. Each factor in the list has an effect on life-cycle cost. Furthermore, only the first point, safety problems, and the fifth point, technical risk, need be considered beyond their direct effects on life-cycle cost. For example, the parasitic power requirement can be reflected as an incremental cost of additional electrical generation capacity plus the operating cost required to generate that electricity. Field serviceability affects repair and maintenance cost. Heater head void volume can be considered in terms of cost because of its effect on engine specific power and efficiency. The effect on engine configuration is important insofar as it affects materials and manufacturing cost. However, other effects must be taken into account such as increased space requirement (cost at the site) or maybe a change in cycle volume between the cooler and bottom of the cylinder (which affects engine specific power) or even technical risk. In conclusion, the final selection criteria will reduce to life-cycle cost, safety problem potential and technical risk.

2.2.3 Discussion

Each of the four candidate heat transport systems were studied in sufficient detail to permit selection of the best system for this application. The selected concept, the heat pipe, was then studied in greater detail to provide the conceptual design described in section 2.2.5. The following paragraphs describe the main features of each candidate system.

2.2.3.1 Liquid Forced Convection

For any liquid convection loop, one would like a material which is liquid over the entire operating temperature range at or near atmospheric pressure. It must be chemically stable and inexpensive. Because of the temperatures (1300-1400°F), we are forced to consider the liquid metals. Some suitable materials might be lead, lithium, sodium, and tin. Sodium would probably be the best candidate material.

Unfortunately, the alkali liquid metals are extremely corrosive, hazardous and difficult to handle. For example, small quantities of sodium and water mixed together in air will explode spontaneously. The larger the sodium inventory, the greater the danger. While this system might have some small advantages over the sodium heat pipe, (the liquid sodium loop could possibly be field serviceable and the heat pipe is definitely not) the advantages are small compared to the principal disadvantage, which is an order of magnitude increase in sodium inventory. For this reason the liquid forced convection loop was rejected.

2.2.3.2 Direct Fired

As previously mentioned, the direct fired approach might be feasible for both the FBC and the low Btu gas heat sources. Because the low Btu gas heat source was rejected as an "A" concept, (advanced

gasifiers will be discussed under section 3. Advanced Concepts) the direct fired by low Btu gas concept was not investigated further. However, because the FBC was selected as the best state-of-the-art heat source, the FBC direct fired concept was investigated in some depth. It was concluded that the concept might be feasible and it might offer advantages over the other heat transport options. However, the FBC direct fired concept does not use state-of-the-art Stirling engine technology and was judged to have the highest technical risk of the heat transport options. Thus it was decided to consider it as a "B" advanced concept. See section 3.2.2 for further discussion.

2.2.3.3 Gas Convection Loop

An initial design investigation was carried out on the gas convection loop. This section will show that the gas convection loop is technically feasible and has some attractive features; however, it is inherently expensive compared with the heat pipe.

Approach

An analysis was generated which predicts the convection loop pumping power requirement as a percent of engine shaft power and as a function of fluid properties, operating parameters (temperatures and heat fluxes) and design parameters (temperature drops, flow areas, system pressure). A figure of merit for the convection loop fluid was defined from this analysis. There was insufficient time during this study to perform a complete design optimization; however, a reasonable design case was studied. A cost estimate was then performed on the basis of that design.

Analysis

The design equations expressing the heat transfer and pressure drop characteristics of the heater head heat exchanger were written

with emphasis on the gas convection loop side design parameters which control the heat transfer behavior of this component. The heat transfer and flow friction equations can conveniently be combined to yield the following single expression for the convection loop pumping power as a percentage of engine shaft power ($\%P_{HH}$) in terms of the various design parameters.

$$\%P_{HH} = \left(\frac{50}{\eta_p \eta_s} \right) \left(\frac{Pr^{2/3} R^2}{M^2 C_p^3} \right) \left(\frac{f}{j} \right) (q^2 T^2) \left(\frac{1}{\Delta T_{lm} \Delta T_{cl}^2 A_x^2 P^2} \right) \quad (1)$$

where:

η_p = efficiency of pump

η_s = efficiency of Stirling engine

Pr = Prandtl number of gas

R = universal gas constant

M = molecular weight of gas

C_p = specific heat of gas

f = friction factor

j = Chilton-Colburn factor = $\frac{h}{GC_p} Pr^{0.67}$

q = heat supplied to heater head (Btu/hr or W)

T = mean temperature of gas

ΔT_{lm} = log mean temperature difference between the convection loop gas and heater head tubes.

ΔT_{cl} = temperature drop of convection loop gas through heater head

A_x = flow area through heater head on the convection loop side

P = mean operating pressure of convection loop

Although most of the system pressure drop and pumping power is likely to be consumed in the heater head heat exchanger, some would also be required for the piping and the in-bed heat exchanger. The following equation gives the percent of engine shaft power consumed in pumping the convection loop gas through a straight duct (skin friction loss only - excludes fitting losses).

$$\%P_{\text{DUCT}} = \frac{14.2}{\eta_p \eta_s} \left(\frac{R^2 \mu^{0.2}}{M^2 C_p^{2.8}} \right) q^{1.8} T^2 \left(\frac{L}{D^{4.8} P^2 \Delta T_{cl}^{2.8}} \right) \quad (2)$$

where:

μ = absolute or dynamic viscosity

L = equivalent length of duct

D = hydraulic diameter of duct

Convection Loop Gas Selection

From equation 1, the pumping power consumed to pump the gas through the heater head is proportional to a term composed of fluid properties. This term may be called a figure of merit (FM) for the gas, where the lower the number, the better the gas.

$$FM_1 = \frac{Pr^{2/3}}{M^2 C_p^3} \quad (3)$$

From equation 2, a similar figure of merit may be defined as follows:

$$FM_2 = \frac{\mu^{0.2}}{M^2 C_p^{2.8}} \quad (4)$$

The values of these figures of merit are given in Figure 2-5 for several gases at 1500°F (815°C). Low values are desirable in order to minimize the pumping power.

The last two columns in Figure 2-5 give the ratio of the figure of merit for each gas to the figure of merit for helium. Assuming that heater head pumping power will dominate over ducting pumping power, FM_1/FM_{1He} is more informative than FM_2/FM_{2He} . From this, one can say that a convection loop which uses air would consume approximately one and one half times as much power as a loop using helium (all other design factors being equal). Similarly one using hydrogen would consume approximately one sixth the power consumed in the same system using helium.

From Figure 2-5, it may be seen that hydrogen is by far the best. However, because of safety considerations, it is not being proposed for use in the convection loop. Carbon dioxide is the next best followed by water (steam) and helium. Water is not being seriously considered because of potential corrosion problems. This leaves helium and carbon dioxide. Both of these gases seem viable and therefore deserve consideration. Helium has the advantage of being the probable working gas inside the Stirling engine. This means for any installation only one gas need be supplied if it is also used in the heat transport loop. Also, with helium in the convection loop, corrosion problems should be minimized. The advantages of carbon dioxide are performance and probably also cost. If CO_2 were used, then the economic penalty of accidental loss of fluid would be reduced. However, the corrosive aspects of using CO_2 need to be addressed. This should be more carefully investigated if CO_2 is

FIGURE 2-5: CONVECTION LOOP GAS SELECTION DATA

Gas	Pr	M	C _p BTU/LB°F	μ LB/HRFT	FM ₁	FM ₂	FM ₁ /FM _{1,He}	FM ₂ /FM _{2,He}
He	0.67	4	1.24	0.110	0.0251	0.0220	1.00	1.00
air	0.71	29	0.29	0.105	0.0388	0.0243	1.55	1.10
F ₂	0.70	2	3.60	0.052	0.0042	0.0038	0.17	0.17
N ₂	0.71	28	0.28	0.101	0.0462	0.028	1.84	1.29
CO ₂	0.71	44	0.30	0.100	0.0152	0.0095	0.61	0.43
H ₂ O	1.3	18	0.575	0.097	0.0193	0.0091	0.77	0.41

chosen over helium. For the conceptual design evaluation, all calculations were done assuming helium; however the results could easily be modified for CO₂.

Pumping Power and System Pressure

Equations 1 and 2 give the pumping power requirement for the heater head and the ducting respectively. No similar analysis has been done for the combustion heat exchanger although its power requirement should be less than the heater head since it need not be as compact. Figures 2-6 and 2-7 were generated using the following values for the constants in equations 1 and 2:

$$\eta_p = .5$$

$$\eta_s = .4$$

Helium was the fluid

$$\frac{f}{j} = 3 \quad (\text{generously low for heat transfer surfaces that would be employed here})$$

$$q = 4.5 \times 10^6 \frac{\text{Btu}}{\text{hr}} = 1.3 \text{ MW} \quad (\text{for 500 HP engine module}).$$

(This value for q is slightly higher than the final conceptual design value, but the difference does not materially change the results.)

T_{supply} is the temperature of the gas at the exit of the combustion heat exchanger. ΔT_{lm} and ΔT_{cl} were optimized* for minimum heater head pumping power at each supply temperature. The values for A_x were chosen on the basis of being the probable range of interest when more detailed investigation of the heater head heat exchanger commences.

* This is discussed further on page 2-45.

FIGURE 2-6: HEATER HEAD PUMPING POWER
FOR HELIUM LOOP

FIGURE 2-7: DUCT PUMPING POWER
FOR HELIUM LOOP

Figure 2-6 gives the pumping power (percent of engine shaft power) consumed in the heater head heat exchanger versus system pressure for various heater head flow areas and convection loop supply temperatures. Figure 2-7 gives the pumping power (percent of engine shaft power) consumed in the ducting versus duct hydraulic diameter for various pressures and supply temperatures assuming an equivalent duct length of 20 feet. The total pumping power requirement for the convection loop is the sum of these two contributions, plus the pumping power consumed in the combustion heat exchanger and other ducting losses.

Discussion

Figure 2-6 shows the power consumed to pump the gas through the heater head versus system pressure for various supply temperatures and flow areas. Because it is important to minimize heater head dead volume, the allowable surface area is limited. Thus dead volume considerations tend to increase the optimum pumping power. This contribution to total pumping power is expected to dominate over the others. If it is desired to keep the pumping power low for any set of design conditions, increasing the pressure sufficiently (possibly to impractically high levels) will accomplish the desired results. Furthermore, in order to keep this power requirement below 2.5% of engine output, the range of interest for pressure is probably 150-500 psi (10-34 atm). From Figure 2-7, in order to keep ducting losses reasonable at this pressure, the range of interest for duct diameter is .5 to 1 ft (15-30 cm). However, as pressures and duct diameters increase, stresses get higher. This implies thick walls and therefore high materials cost.

There is probably an optimum pressure which trades auxilliary power versus increased materials cost to arrive at minimum life-cycle cost. This optimum has not been determined. Rather this analysis was used to generate a reasonable design. While this design is probably not the exact optimum, it is sufficiently close to give meaningful

results. These results can then be used in a preliminary cost analysis to compare the gas convection loop to the other candidate heat transport systems.

Conceptual Design

A conceptual design for the gas convection loop is described below. The following methodology was used in the design: First, the supply temperature was chosen from fluidized bed and in-bed heat exchanger design considerations. A reasonable heater head flow area was chosen on the basis of Stirling engine heater head heat exchanger considerations. Next, an acceptable pumping power requirement was established. From the above assumptions and Figure 2-6, the required mean pressure can be determined. The ducts can then be sized from Figure 2-7. Wall thickness and material requirements can then be estimated on the basis of high temperature creep strength.

Supply Temperature (T_{supply}): The supply temperature chosen (1500°F, 815°C) is an optimistic estimate. Referring to equation 1, the mean gas temperature, T , appears in the numerator. This has the effect of increasing the pumping power as the supply temperature is increased. However, a 50°F (28°C) rise in temperature above 1450°F (788°C) or 1910°R (1061°K) is only a 2.5% change in temperature, thus this is a minor effect. The more significant effect of increasing the supply temperature is to increase ΔT_{lm} and ΔT_{cl} . Since these temperature drops appear in the denominator, and since their absolute value is small compared to the absolute supply temperature, increasing the supply temperature has the overall effect of significantly reducing the pumping power.

However, as the supply temperature is increased the log-mean temperature difference between the convection loop gas and the fluidized bed is reduced. This, of course, has the effect of reducing the heat flux to the in-bed heat exchanger. In the previous chapter,

Section 2.1.5, Conceptual Design of FBC, it was noted that the bed heat release per unit cross-sectional area is limited by the heat transfer to the tubes. The bed cross-sectional area shown in that section is based on the heat flux to heat pipes (to be discussed in the next section). At 1500°F (815°C) supply temperature and 1650°F (900°C) bed temperature, the heat flux to the helium in-bed heat exchanger is approximately one-half the heat flux to the heat pipes which were used in the conceptual design of the FBC. This is because the log-mean temperature difference is approximately one half the value for that pipes. Thus, for the same tube packing geometry, the area heat release rate of the bed with the helium loop must be half what it is for heat pipes. This requires half the fluidizing velocity, and thus even smaller particle size. In fact, it has not been established that such a bed could be built. This is why the feasibility of this supply temperature is questionable. Increasing the bed temperature is not a realistic solution since 1650°F is a practical limit for acceptable sulfur capture in a conventional AFBC.

It should be noted at this point that the in-bed heat transfer surface area for the helium loop system is twice that of the heat pipe system. Since the helium loop tubes must contain high pressures (versus the heat pipes which are under vacuum), the helium loop tubes require relatively heavy walls. Thus, the in-bed heat exchanger for the helium loop system will contain more than twice as much material as the in-bed heat exchanger. With 250 psi internal pressure, and with a heat exchanger wall temperature greater than 1500°F, the creep strength requirement will dictate the choice of materials. In fact, high nickel alloys will be required. Thus the helium loop requires a greater quantity of a more expensive material than the heat pipe.

Note also that in equation 1, ΔT_{C1} is squared in the denominator and ΔT_{1m} is to the first power in the denominator. If the heater head tube temperature is assumed to be a constant (which is approximately true) then a relationship between ΔT_{C1} and ΔT_{1m} can be established

for any supply temperature. It is then a simple matter to optimize (i.e., maximize) the product $\Delta T_{cl}^2 \Delta T_{lm}$ to obtain minimum pumping power. This was done for a supply temperature of 1500°F (815°C) and heater head temperature of 1328°F (720°C). Optimum values of ΔT_{cl} and ΔT_{lm} were found to be 145°F (81°C) and 78°F (43°C). However, if ΔT_{cl} is changed to 130°F (72°C), then the pumping power will only be increased marginally while the log mean temperature difference to the heater head heat exchanger will be increased from 78°F (43°C) to 93°F (51°C). This leads to a more favorable heater head heat exchanger design and thus, 92°F and 130°F were used for ΔT_{lm} and ΔT_{cl} respectively.

Reasonable Heater Head Flow Area: Referring again to equation 1, the heater head flow area on the convection loop side, A_x , is also an important design parameter. Increasing this area decreases the pumping power requirement linearly. The limit on an acceptable area is basically due to the limit on the length of heater head tubes in the Stirling engine. Assuming a similar heater head tube arrangement as the direct fired engine, (circular arrangement of U-tubes) except where extensive finning is utilized and the tubes are staggered to give a multiple pass arrangement, a reasonable upper limit to the flow area for the four cylinder module heater head is 2 ft² (1860 cm²). Heat exchanger analysis shows that 1.4 ft² (1300 cm²) is actually a more reasonable area from a heat transfer design point of view. Thus, 1.4 ft² will be used in the example system design.

Acceptable Pumping Power Requirement: This is a problem which requires some iteration. If the limit on allowable pumping power is too low, then excessive pressures are required. This of course drives up the cost. On the other hand, as the allowable power consumption increases, less and less is gained in the area of reduced pressure. This can be seen by the curves in Figure 2-6 which show that at higher pressures, small changes in power require large changes in pressure. At lower pressures, large increases in power result from small pressure changes. Curve number 2 in Figure 2-6, approximately fits the

assumptions made so far in this conceptual design process. Allowing 2.5% power requires a pressure of 240 psia. Allowing 1% power requires a pressure of 380 psia. Thus, the material requirements almost double for this difference in power. For this reason, 2.5% power to pump the gas through the heater head was selected as a reasonable value.

Ducting: Referring to Figure 2-7, curve number 1 is closest to the assumptions made in this conceptual design. At 1% power loss in ducting, the duct diameter required is estimated to be 0.87 feet (10.4 inches or 26.5 cm).

Manifolds and In-Bed Heat Exchanger: To simplify the estimating process, similar analyses were not done for manifolds and the in-bed heat exchanger. Rather, it was assumed that these losses would be on the same order of magnitude as the ducting losses. A 1.5% power requirement was assumed, bringing the total estimated power requirement to 5% of engine shaft power.

Cost Estimate

Based on the assumptions made in the conceptual design, it is possible to generate estimates for material requirements including type and quantity. Figure 2-8 gives the results of the conceptual design transformed into materials specifications and quantities. These materials specifications in turn form the basis of the cost estimate. The results of the cost estimate are given in Figure 2-9 for one 500 hp engine module. Explanation of each of the numbers in Figure 2-9 follows.

The cost of the fan with housing, motor, water cooled shaft, bearings and high temperature and pressure shaft seal is estimated at \$6000. This is based on a quote of \$3100 from a blower manufacturer for the fan, housing, shaft and bearings. However, the fan which was

FIGURE 2-8: HELIUM LOOP MATERIAL QUANTITIES
FOR 500 Hp MODULE

Heater Head

108 lbs (49 kg) of fin material 0.025" (0.6mm) thick.

Estimated price for high nickel alloy: \$5/lb

Finning operation: 268 ft/module (82 m/module) tubing

7 fins/in (2.8 fins/cm)

Estimated price: \$1000/module

Total cost chargeable to helium loop: \$1500/module

In-Bed Heat Exchanger

1310 ft (396 m) of 1-1/4 schedule 80 RA-800 H* pipe

1.66" O.D. x 0.191" Wall thickness, 3 lbs/ft

Quoted Price: \$15.53/ft (\$50.95/m)

Total = 1310 x \$15.53 = \$20,300

Ducting

Size: 10 ft x 12 in (Dia) x 0.375 in (thick)

3.05 m x 30.5 cm x 9.5 mm

Material: Incoloy 800 H plate at \$3.13/lb (\$6.89/kg)

Total: 490 lbs x \$3.13/lb = \$1500

* Alloy by Rolled Alloys which is similar to Incoloy 800H

FIGURE 2-9: HELIUM LOOP COST ESTIMATES
FOR 500 Hp MODULE

<u>Materials</u>	<u>COST PER MODULE</u>
1) Fan, Incl. Housing, Motor, Water Cooled Shaft and Union, Bearing, Shaft Seal, Guards, Etc..	\$ 6,000
2) Heater Head - Fin Material and Finning Operation Only - (Tubes Included in Engine Costs)	\$ 1,500
3) In-Bed Heat Exchanger Tubes	\$20,300
4) Ducting	\$ 1,500
5) Manifolds	<u>\$ 1,500</u>
SUM OF MATERIALS:	\$30,800
<u>Manufacturing</u>	
100% of Materials Costs Except for Item Number 1.	<u>\$24,800</u>
SUBTOTAL:	\$55,600
<u>Pumping Power</u>	
Penalty for Consumption of 5% of Engine Shaft Power (Only for Comparison with Heat Pipe)	<u>\$20,000</u>
TOTAL:	\$75,600
For 500 hp Module or 1180 kW of Heat Transferred	\$64/kW _{heat}

quoted is only suitable for temperatures up to 1300°F (700°C) and the housing is not a suitable pressure vessel. Furthermore, the additional cost of the pumped cooling loop, motor, shipping costs, and safety accessories such as guards, etc. must be added. Thus, twice the baseline quote seems like a generously low value.

The heater head cost shown includes only the cost of fin material and the cost of the finning operation. No allowance is made for the cost of the heater head tubes. This is done because the cost of the heater head tubes is omitted from the heat pipe cost, and is included in the engine cost. In this way, direct comparison between helium loop cost and heat pipe cost is possible. However, because the helium loop makes extensive use of fins on the heater tubes, the cost of the fin material (\$500) and the cost of the finning operation (\$1000) must be included. Thus, the total cost chargeable to the helium loop is \$1500.

The largest single contribution to convection loop cost is the in-bed heat exchanger (\$20,300). This is to be expected, as discussed previously under the sub-heading: Conceptual Design. The dollar value was determined as follows. The interior heat transfer coefficient (helium to tubes) was estimated to be 100 Btu/hr-ft²-°F (5.68 W/m²-°C) while the exterior heat transfer coefficient was estimated to be 72 Btu/hr-ft²-°F (4.09 W/m²-°C). With a bed temperature of 1650°F (900°C), a gas supply temperature of 1500°F (816°C) and a gas return temperature of 1370°F (743°C), the overall log-mean temperature difference is 208°F (116°C). Assuming 1-1/4" Schedule 80 RA 800H pipe is used, the total length of pipe required for adequate heat transfer area is estimated to be 1310 ft (399 m). At 2.997 lbs/ft the mass of tubing is estimated to be 3930 lbs (1781 kg). The cost was quoted at \$15.53/ft, (approximately \$4/lbs for high nickel alloy) leading to a total material cost of \$20,300.

The ducting materials cost was estimated by calculating the material contained in 10 ft (3 m) of 12 in. (30.5 cm) diameter duct. A wall thickness of 0.375 in (0.95 cm) is required for RA 800H to contain 250 psi for 30,000 hrs creep life. A cost for plate of \$3.13/lb (\$1.42/kg) was used to calculate the ducting materials cost of \$1500. Manifold materials were estimated to be approximately equal in cost to the ducting.

The cost of manufacturing was estimated to be 100% of materials cost. This is a number often quoted in industry for manufacturing equipment of similar complexity with tight quality control and in small quantities. Presumably, if the unit were manufactured in large quantities this cost would be lowered, however it is difficult to estimate how much lower. If, optimistically, the manufacturing cost was reduced to 50% of materials cost, then the total cost would be reduced by \$12,400.

The pumping power penalty of \$20,000 accounts for the power consumed by the helium loop. Although this is not a true capital cost, it is useful for comparison with the heat pipe, which does not consume any pumping power. In this way, direct comparison can be made between the helium loop and the heat pipe without having to develop full life cycle cost estimates of both systems.

The value \$20,000 is intended to represent the incremental cost of additional generation capacity and additional operating expense in order to generate the required power. Reference 22, page 25 indicates that a 5% increase in efficiency is worth about \$100/hp in initial costs. The helium loop consumes roughly 5% of engine shaft power, thus decreasing the effective engine efficiency by 2 percentage points (assuming 40% nominal efficiency). Thus, this power consumption could be applied as a \$40/hp penalty to the helium loop system. For a 500 hp module this translates into a \$20,000 penalty. (Equivalently, at \$1000/kW, 20 kW of pumping power is worth \$20,000.)

2.2.3.4 Sodium Heat Pipe

At the outset of the program, a heat pipe design investigation was initiated. Many ideas were generated and studied, and as a result of these studies the generic characteristics of the heat pipe system were determined for the Stirling application. A preliminary conceptual design, analogous to the design of the helium convection loop, was studied. Preliminary estimates for heat pipe cost were generated from this design. Based upon these initial studies, the heat pipe was selected as the best state-of-the-art heat transport system option. Further analysis ensued and a better design was developed. This design is presented in section 2.2.5 Conceptual Design of Heat Pipes.

This section of the report covers the generic characteristics of the sodium heat pipe and how they relate to the Stirling engine. The cost estimate is not discussed in detail here because it is discussed in Section 2.2.6. Thus, the factors involved in the decision making process will be presented, without the burden of details. The details are covered in sections to follow.

Temperature and Pressure

Sodium heat pipes and Stirling engines are well suited to each other. The current materials limitation of super alloys for use as heater head tubes limits the upper operating temperature to less than 900°C if any reasonable life is required. This is an ideal temperature range for sodium heat pipes.

Sodium has an atmospheric boiling point of 895°C; thus for all operating temperatures less than 895°C the heat pipe operates below atmospheric pressure. Hence, the envelope material need only be capable of withstanding an external pressure of 15 psi. This is important in keeping material costs down.

Likewise, the sodium heat pipe is well suited for use with a coal-fired FBC which is operated in the temperature range where the sulfur retention is high and NO_x emissions are low (750-900°C). This temperature range falls below the atmospheric boiling point of sodium as the case of the upper limit temperature of the Stirling engine heater head tubes. Accordingly, the heat pipe will operate at less than atmospheric pressure. Thus, the heat pipe envelope material should be selected to withstand the effects of corrosion and erosion within the FBC, and to prevent hydrogen permeation into the heat pipe. However, creep strength is not important.

Generic Advantages

The heat pipe has several unique features not found in other heat transport systems. They are:

- 1) The heat pipe is a heat flux transformer.
- 2) The heat pipe is essentially isothermal.
- 3) The heat pipe is a static system.

These features of the heat pipe are very important in the Stirling application. Some discussion on each of these features follows.

Heat Flux Transformer: This property of the sodium heat pipe is particularly important since this application requires the use of dirty fuels, such as coal. In the clean fuel fired engine, high heat fluxes in the combustion chamber are possible. This is due to the large log mean temperature difference between the combustion products and the heater head tubes. Thus, because high heat fluxes are possible, the internal volume of the heater head tubes may be kept low. This is critical to the power and efficiency of the Stirling engine.

When dirty fuels are to be used, high heat fluxes are not possible at the primary combustion gas heat transfer surfaces. If the flame temperature is high (to provide large temperature differences) then the tubes will become fouled with slag, which solidifies on and insulates the tubes. In the FBC, the slag is not a problem; however, the temperature is considerably lower. Thus, the heat flux to the in-bed tubes is much less than with clean fuel firing, and a heat flux transformer is required to obtain the desired high heat flux to the engine heater head tubes. In fact, the sodium heat pipe provides such high heat transfer coefficients on the outside of the heater tubes, that the inside of the tubes limits heat transfer. Thus, internal heater head volume is minimized and engine performance is optimized.

Isothermal Property: It was mentioned that the heat pipe is essentially isothermal. This means that even for high heat fluxes, the temperature drop through the heat pipe is small (from heat input to heat rejection end). In the Stirling-FBC application this is very important. With a Stirling heater head temperature of 720°C (1330°F) and an FBC temperature of 900°C (1650°F), the overall temperature difference is only 180°C (320°F). If the temperature drop through the heat transport system is large, then the available temperature difference from the FBC to the heat transport system is small. Thus, large surface areas would be required to transfer the heat. This, of course, means high cost. Thus, the low temperature drop property of the heat pipe serves to reduce system cost significantly.

Static System: This feature simply means that there are no moving parts: No wear, no friction, no lubrication nor cooling requirements and long life are results of this feature. The life of the heat pipe system is determined by corrosion and hydrogen diffusion. Furthermore, no auxiliary power is consumed. This is an important cost benefit when compared to the helium convection loop.

Other Advantages: As will be seen in section 2.2.5, Conceptual Design of Heat Pipes, the heat pipe can be designed to be redundant. Thus, with some degree of redundancy built into the system, very high reliability should result.

Generic Disadvantages

While the heat pipe has many advantages, there are a few disadvantages which must be taken into account with careful design. The areas of concern are safety, field serviceability, and effect on engine configuration. These are discussed in some detail below.

Safety: While the heat pipe is a hermetically sealed device, with sufficient abuse it is possible to cut through the steel walls and break the hermetic seal. Unfortunately, once exposed, the liquid metal inside the heat pipes is potentially explosive. There are, however, many potentially explosive chemicals in our daily lives which, with proper care, provide little danger and great benefit to society. Natural gas in our homes and automotive gasoline are only two examples. Thus, the heat pipes must be designed for safe handling. This is possible within the realm of state-of-the-art.

Furthermore, the heat transport system will be exposed to the corrosive environment of the FBC. It is important to ensure that when a heat pipe fails in the FBC, that no danger results. Fortunately, the heat pipe operates below atmospheric pressure and thus any leaks in the heat pipe envelope will first cause gas to leak into the heat pipe rather than spew liquid sodium into the bed. It is expected that after such a failure, the heat pipe will be raised to bed temperature (its heat flux to the engine will stop) and possibly slowly leak small quantities of sodium into the bed. This is not expected to be problematic in any way.

Field Serviceability: The heat pipe is a hermetically sealed device containing liquid sodium at pressures below atmospheric. Sophisticated equipment and extreme cleanliness is required to service the inside of the heat pipe. Such service techniques are not readily accomplished under field conditions. Accordingly, the heat pipe must be cheap enough and easy enough to change so that in the event of failure, a replacement heat pipe may be field installed and the defective heat pipe returned to the factory for service or disposal. This is important if the heat pipe is to be a practical working device to be serviced by the same level of personnel as is required for ordinary boiler or engine service.

Effect on Engine Configuration: Heat pipes can be designed to operate in any orientation, with or against gravity, and over horizontal distances. However, the cost of the heat pipe is a function of the requirements imposed upon it. From the point of view of reducing cost, it is desirable to have the heat pipe assisted by gravity (not working against gravity) and to have the heat pipe as short as possible. This can have an effect on engine configuration. For example, it is desirable to have the condensate run down off the heater head tubes back to the evaporator by gravity. This is most rapidly accomplished by placing the engine heater head below the cylinders and crankcase. Thus, in comparison to conventional practice, the engine is mounted upside-down. However, this is not expected to be a problem.

Cost Outlook

As previously mentioned, the detailed cost analysis will not be presented here, but rather is presented in Section 2.2.6 for the conceptual design described in Section 2.2.5. However, in order to understand the decision-making process which led to selection of the heat pipe, the costs must be considered. The FOB cost of heat pipes

from the factory is estimated to be approximately \$25,000 per engine module (1980 dollars) or approximately \$21.20 per kilowatt of heat transfer capability. See Section 2.2.6 and Appendix C for details of this estimate. This compares to \$75,600 per engine module for the helium convective loop (including a penalty for consumption of pumping power).

2.2.4 Selection

During the course of the discussions presented above, both the liquid forced convection loop and the direct-fired concepts were eliminated from consideration for the state-of-the-art engine. The liquid loop was rejected on the grounds of safety and the direct-fired concept was eliminated on the basis of technical risk - thus it is now part of the "advanced concepts" section of this report. That leaves the helium loop and the heat pipe.

As discussed in section 2.2.2 Selection Criteria, the final selection criteria reduces to life-cycle cost, safety, and technical risk. The heat pipe has roughly a 3 to 1 advantage over the helium loop in life-cycle cost. (This does not include maintenance cost - which is not expected to be less for the helium loop than for the heat pipes.) Thus, unless the helium loop is far superior in the other two categories, the heat pipe is the clear choice.

Safety

The question as to which is the safer system could probably be debated either way. True, the heat pipe utilizes liquid metals, however the helium loop utilizes hot high-pressure gas. Rupture of a duct or vessel in the helium loop is more likely (due to the higher stresses) and probably more dangerous (due to the high potential energy) than the possibility of fire or explosion due to contact of the sodium with water.

Technical Risk.

Both of these systems carry some technical risk. There are several unanswered technical questions in the conceptual design of the helium loop (FBC Tube layout, circulator design, etc.). While the heat pipe has been used for many different applications, and it has been produced in large quantities, there is no large-scale production of high-temperature heat pipes today. Furthermore, there is only limited experience using heat pipes in fluidized beds. However, while the helium loop is subject to shut-down at the smallest failure, the heat pipe system is designed for redundancy, and thus is more likely to give long trouble-free life as designed. On balance, the heat pipe is believed to have a smaller technical risk associated with it than does the helium loop.

Conclusion

The heat pipe is substantially less costly than the helium loop. Both systems have possible safety hazards and both carry some technical risk. However, not nearly enough difference can be seen in these areas to overcome the cost difference. Thus, the heat pipe is the clear choice.

2.2.5 Conceptual Design of Heat Pipes

The concept chosen for 1985 demonstration is illustrated in Figure 2-10. It is a two-stage gravity assisted heat pipe. This means that instead of using a single heat pipe to carry the heat from the FBC to the engine, two heat pipes are used in series. The primary heat pipes consist of an array of approximately 250 one inch diameter heat pipes distributed across the fluidized bed. They interface with (and are attached to) the secondary heat pipe via tapered joints. The condenser of the primary heat pipes supplies heat through the metallic interface to the secondary heat pipe evaporator. The sodium

FIGURE 2-10: TWO-STAGE, GRAVITY ASSISTED
HEAT PIPE CONCEPT

vapor thus generated in the secondary heat pipe condenses on the Stirling heater head tubes. Thus, the Stirling engine heater head tubes, the cylinder caps and the regenerator caps are a part of the hermetically sealed secondary heat pipe. The concept shown in Figure 2-10 utilizes cylinder caps and regenerator caps which are welded to a baseplate. This baseplate is bolted to the water jacket of the Stirling engine and is thus kept cool. In order to minimize the heat leak from the sodium vapor to this cold plate, inert non-condensable gas is introduced. Suitable insulation prevents convection currents and thus the plate is insulated. [This design (illustrated in Figure 2-10) does not allow precise alignment of the cylinder caps because they are welded to the baseplate. Alternative concepts for this interface were studied by United Stirling and are reported in Section 2.3.2, Engine Design for FBC With Sodium Heat Pipe.]

Before proceeding into a more detailed discussion of this concept, it is educational to review some of the considerations which led to this concept. In particular, the possible arrangements for one-stage heat pipes should be examined in order to understand the advantages of the two-stage concept.

Single-Stage Concepts

At the outset of the program, three alternative heat pipe concepts were examined. All three concepts used heat pipes which had the evaporator located in the fluidized bed and had the engine heater head tubes forming the condenser. Thus, they were single-stage heat pipe concepts. The basic difference between the three concepts was the number of heat pipes which would be required per engine module. One concept utilized one heat pipe per module, one utilized one heat pipe per engine cylinder (4 per module) and the third concept utilized one heat pipe per heater head tube (96 heat pipes per module). Some discussion of each follows.

One Heat Pipe per Module: This concept involves using a large container around the Stirling heater head which interfaces with the engine in a similar fashion as the secondary heat pipe in the two-stage concept. The evaporator in this concept consists of a large number of tubes or "fingers" which are welded to the large container. These fingers are dispersed throughout the fluidized bed. The sodium liquid which condenses on the heater head tubes must be wicked into each of the fingers. The liquid in these fingers is evaporated and the sodium vapor travels down the fingers and into the large container where it condenses on the heater head. From here the liquid is picked up by the wick for return to the fingers. The primary difference between this concept and the two-stage concept illustrated in Figure 2-10 is the fact that with this concept, the fingers are in communication with the large tank, whereas in the two-stage concept they are independent but are attached at a tapered metal interface.

The primary advantage of this concept (one heat pipe per module) is that the temperature of each and every heater head tube will be the same (i.e., no hot spots). This means that for a given material (with its associated temperature limit) the optimum engine performance may be achieved. Uniform heat input into the four cylinders is assured.

However, the drawbacks to this approach are numerous. First, the interface between the heat pipe and the engine (discussed previously and in section 2.3.2) has alignment and thermal stress problems which must be solved. Second, the wick structure required to distribute the liquid sodium evenly to each of the fingers is extremely complex. In fact, the individual fingers might even have to be joined at the liquid end in the fluidized bed to ensure that none of the fingers runs dry. Serviceability was discussed in section 2.2.3.4 and it was concluded that the best approach is to design for easy replaceability. This concept (one heat pipe per module) would involve replacing a large heavy container with many long appendages.

on each occurrence of failure in a single appendage. This would not be an easy task. Hence, this concept is very poor from a reliability point of view. Should any one weld fail, or should corrosion penetrate one of the fingers in the FBC, then the entire engine would have to be shut down for heat pipe replacement. It is likely that this would mean considerable downtime.

One Heat Pipe per Cylinder: This approach involves use of four heat pipes per four cylinder module. Each heat pipe would cover one cylinder and its associated heater tubes and would have many fingers branching off each of the main containers and extending into the FBC. This concept would ease the problems of engine interface somewhat, and might ease the wicking problem discussed in the one heat pipe per module concept. Furthermore, because each heat pipe would be roughly one fourth the size of the one heat pipe per module concept, replacement of failed heat pipes could be easier. However, unless the four heat pipes are cross-connected (which would make for more complicated assembly and disassembly) then the temperature uniformity which was guaranteed by the one heat pipe per module concept would be lost. Furthermore, the reliability problem of the one heat pipe per engine module approach is still a problem. The only difference here is that in the event of a failure, only one cylinder's heat pipe would be replaced. However, the entire engine module must still be shut down for that replacement.

One Heat Pipe per Heater Head Tube: This concept involves the use of 96 individual heat pipes. No branching of heat pipes (as in the previous examples) would be required, thus simplifying the wick structure. The heat pipe would most likely be configured in such a way that no thermal stress problems at the engine-heat pipe interface would result. Serviceability would most likely be better with this concept than the other two, except for possible accessibility problems. Reliability would be improved substantially over both of

the other two concepts, in that, should only one or two heat pipes fail, it would still be possible to run the engine at high load factor. The disadvantages with this concept are with regard to packaging and heat distribution. The arrangement of 96 heat pipes covering the heater tubes is not an easy task. Furthermore, these heat pipes would have to go from the engine to the FBC, and would have to be evenly distributed throughout the FBC. It is likely that each heat pipe would be different in length and bends, etc., although a pattern might be devised to provide more similarity of parts.

The heat distribution question could be problematic. Not only would it be possible to have non-uniform heat input to each cylinder, but also it would be possible to have uneven heat input to the various tubes of one cylinder. This is not desirable.

The conclusion to be drawn is that each of these single-stage heat pipe concepts leaves something to be desired. None of the three concepts provides the desired redundancy which was mentioned earlier in section 2.2.3.4. The following paragraphs explain how the two-stage concept solves the problems of the single-stage concepts while providing operational redundancy.

Two-Stage Heat Pipe Concept

This concept is illustrated in Figure 2-10. The introductory paragraph explained what it is and how it works. This section will explain why it is superior to other heat pipe options.

First, it is clear that if the 250 primary heat pipes are bolted to the secondary heat pipe and they are not in internal communication with it, then the failure of a single primary heat pipe will be almost unnoticeable by the engine. When a primary heat pipe fails, the secondary heat pipe will still be able to deliver sufficient heat at the required temperature, uniformly, to all of the heater head tubes.

Furthermore, as the number of primary heat pipes which have failed grows, the engine will still be able to run, but possibly at reduced load factor. This is very important from an overall system operation and reliability point of view. It should be possible to build primary heat pipes which on average will be able to last long enough so that primary heat pipe replacement can be performed on a scheduled shutdown basis. This is the desired operational redundancy which was described earlier. It is felt that this feature substantially reduces the technical risk (i.e., poor reliability) of the heat pipe system. Furthermore, because the secondary heat pipe is not exposed to the corrosive environment of the FBC, its life should be longer than that of the primary heat pipes.

The heat distribution problem of the one heat pipe per heater head tube concept has been eliminated. If one secondary heat pipe is used (as shown in Figure 2-10), then even heat distribution is guaranteed. If four secondary heat pipes per engine module are used (for manufacturing, assembly, and interface reasons) then uniform heat distribution among cylinders will rely on the heat flux uniformity of the four sections of the FBC module. Alternatively, the secondary heat pipes might be cross connected; or it might be possible to modulate the outputs of the four bed quadrants individually by controlling the air flow.

The wick structure required with this two-stage concept is minimal, and is not expensive. Furthermore, many identical small heat pipes constitute the primary array. By high volume production techniques, the cost of these heat pipes will be kept low. These factors, plus the high reliability and the fact that optimum engine performance is achievable, contribute to a favorable cost outlook.

This concept is also excellent from a safety point of view. The most likely failure will be of primary heat pipes in the FBC. Should a sodium leak develop, only about one tenth of one pound of sodium

would be released per primary heat pipe failure. Furthermore, because of the nature of the corrosive action and because the heat pipes operate at partial vacuum, small slow leaks are more likely than sudden release of liquid sodium.

The gravity assist design ensures that when the unit is started up, liquid will be available in the evaporator. Furthermore, preliminary analysis has shown that the start-up of the heat pipes can occur at a pace substantially faster than start-up of the FBC, and thus there are no anticipated start-up difficulties.

Further details of the primary and secondary heat pipes follow.

Heat Pipe Design Details

A conceptual design layout of the heat transport system is seen in Figure 2-11. It consists of a set of primary heat pipes, vertically oriented in the FBC, their adiabatic section going through the freeboard and their condenser extending into the secondary heat pipe. The primary heat pipe condensers are tapered and are mechanically held in place via a flange in a mating tapered section of the secondary heat pipe.

The primary heat pipe arrangement shown in Figure 2-11 is for a cylindrical fluidized bed 1 meter in diameter. After this layout was devised, the FBC cross section was changed to 1.1 meters square. It is intended that the number of heat pipes would not change but rather their spacing would be increased from 5.5 cm to 6.5 cm. It should be noted that a heat pipe "membrane wall" is under consideration. (Similar in concept to a water cooled membrane wall in conventional boilers.)

The details of the primary heat pipe and its thermal performance are listed in Figure 2-12. Figure 2-13 shows the details of the evaporator and seal off cover. This seal off cover would most likely

FIGURE 2-11: HEAT PIPE CONCEPTUAL
DESIGN LAYOUT

FIGURE 2-12:

PRIMARY HEAT PIPES

Number - 254

Spacing - 5.5 cm

Envelope - 2.54 cm OD x 0.25 cm wall

Material - 18 SR (Armco)

Evaporator Coating - .015 cm thick AL_2O_3/ALN

Evaporator - 100 cm

Adiabatic - 110 cm

Condenser - 50 cm

Condenser Taper - .05 cm in 50 cm

Interface Material - copper

Wicking - circumferential "V" grooves feed from spiral wound "L"
through utilizing gravity.

Thermal Performance -

1.18×10^6 watts total

4645 watts/primary heat pipe

Evaporator heat flux - 5.75 W/cm^2

Condenser heat flux - 11.64 W/cm^2

Coating ΔT 0.35°C

Evaporator Wall ΔT 2.30°C

Heat Pipe Vapor ΔT 7.30°C

Falling Film ΔT $.45^\circ\text{C}$

Condenser Wall ΔT 4.60°C

Heat Pipe Total ΔT 15.0°C

EVAPORATOR DETAIL

FIGURE 2-13:

TITLE PRIMARY HEAT PIPE EVAPORATOR			
NAME		ST'D. TOLERANCES	
DRAWN		DECIMAL	.xx = +.015 .xxx = +.005
APPROVED	CSME	DATE	4/1/60
SCALE	MATL	FRACTION + 1/64	
ASS'Y or BILL		ANGULAR	+ .003 PER. INCH OF LENGTH
THERMACORE, INC.		DWG. NO.	A-82-1

be attached in some way to the distributor plate in the FBC. This attachment must allow for thermal expansion of the heat pipes but prevent gross oscillations of the pipes which could otherwise be introduced by the bubbling action in the bed.

The condenser of the primary heat pipe and mating section of the secondary heat pipe is seen in Figure 2-14. The mechanical tapered interface will have an intermediate element of a thin sheet of soft copper (.01 cm) to increase the thermal conductance of the clamped interface. Thermacore's experience with joints of this type indicate a thermal resistance of $1/2$ to $2^{\circ}\text{C}/\text{W}/\text{cm}^2$. Thus, the mechanical joint will experience a nominal temperature drop of 11.5°C .

For easy servicing, the center to center spacing of the primary heat pipes should be at least two pipe diameters. Furthermore, an easily removable flange pattern is needed. Several flange concepts are shown in Figure 2-15. The first three flange designs require two bolts per heat pipe and removal of the bolts for heat pipe removal. Design 4 requires two bolts per heat pipe, but can be rotated out of position after loosening the nuts. Design 5 and 6 require only one bolt per heat pipe as neighboring heat pipes share bolts. Design 5 allows the heat pipe to be rotated out of position. Design 6 requires removal of the bolts or nuts for heat pipe removal.

The secondary heat pipe is a cylinder one meter in diameter by one meter high. The lower end is domed and has 254 protrusions, which accept the primary heat pipes and act as the secondary heat pipe evaporators. The engine interface is discussed in section 2.3. Internally there are supports to provide structural rigidity. The sodium vapor condenses on the heater head tubes and drips off by gravity, which feeds the wicks covering the evaporators. Figure 2-16

FIGURE 2-14: INTERFACE DETAIL

TITLE PRIMARY - SECONDARY HEAT PIPE JOINT				ST'D. TOLERANCES	
NAME				DECIMAL .xx = +.015 .xxx = +.005	
DRAWN			DATE	FRACTION + 1/64	
APPROVED OME			4/11/80	ANGULAR +.003 PER. INCH OF LENGTH	
SCALE		MATL		DWG. NO.	
ASS'Y or BILL				A-82-3	
THERMACORE, INC. HEAT TRANSFER SPECIALISTS

NO 1

NO 2

NO 3

NO 4

NO 5

NO 6

FIGURE 2-15: FLANGE CONCEPTS

FIGURE 2-16:

SECONDARY HEAT PIPE

Envelope - 1 m diameter x 1 m high

Material - 18 SR

Power - 1.18×10^6 watts

254 Evaporators

Heat flux - 11.64 W/cm^2

72 Condensers

Heat flux - 25 W/cm^2

Evaporator Wicking

1 layer 60 mesh

1 layer 200 mesh

Evaporator Wall ΔT - 4.3°C

Evaporator Wick ΔT - $.46^\circ\text{C}$

Vapor ΔT - $.25^\circ\text{C}$

Condensing Film ΔT - $.99^\circ\text{C}$

Total ΔT 6.0°C

FIGURE 2-17:

SYSTEM PERFORMANCE

Total Power - 1.18×10^6 watts

Primary Heat Pipe ΔT - 15°C

Interface ΔT - 11.5°C

Secondary Heat Pipe ΔT - 6.0°C

Total ΔT 32.5°C

lists the details of the secondary heat pipe. Figure 2-17 summarizes the temperature drop performance of the two-stage heat pipe design.

The secondary heat pipe will be internally insulated to reduce the heat leak through the heat pipe-engine interface (see Section 2.3.2). This internal insulation will be augmented by the use of a small amount of inert gas such as argon or helium which will increase the effectiveness of the insulation.

A mode of operation is anticipated where the FBC is at a high temperature and there is no heat transfer to the engine. This can occur in the case of a rapid emergency engine shutdown or in the case where the bed is approaching full temperature and the Stirling engine has not yet been started. The worst case is the emergency rapid engine shutdown, at which time heat transfer to the engine would stop, yet the bed continues to produce heat because of the unburnt coal in the bed. (Coal supply is also stopped.)

Several system scenarios can be envisioned. One might be "slumping" the bed. This means stopping all combustion air flow, and thus the heat pipes would be raised to bed temperature (1650°F). Another might involve by-passing the preheater and baghouse, thus dumping all heat to the atmosphere. In such a case the maximum attainable bed temperature will be less than the maximum allowable heat pipe temperature. The maximum allowable heat pipe temperature has not yet been determined, but is estimated at over 2000°F. (The limiting factors are the internal heat pipe pressure and the strength of the wall material.) In either case, there is no possibility of thermal runaway (as with electric heating) and thus no serious problems are anticipated.

Materials Selection

The choice of heat pipe envelope material for use with FBC-Stirling engines is a challenging task. Internally the heat pipe material must be compatible with sodium. Externally it must survive

the erosive, corrosive atmosphere of the FBC, and it must provide a reduced rate of hydrogen permeation into the heat pipe to prevent the heat pipe from suffering a performance penalty.

Figure 2-18 shows these material considerations along with two suggested candidate heat pipe materials, their nominal cost, composition and strength at elevated temperatures. The much better strength of Incoloy 800 is not needed in this application. Furthermore, to date, little hydrogen permeation data or sodium compatibility data is available with Incoloy 800. Thermacore has substantial data on the Armco 18-SR material and similar materials. A piece of 18-SR was tested at Argonne National Laboratory and showed a 300% reduced hydrogen permeation (23) compared to conventional stainless steel such as 304 stainless. A similar alloy (Allegheny Ludlum OR-1) was tested as the envelope of a sodium heat pipe for 1000 hours at 800°C in a gas flame and showed no hydrogen permeation nor did the heat pipe fail. This heat pipe is shown in Figure 2-19.

An 18-SR sodium heat pipe was also tested at 800°C in a gas flame for several hundred hours and showed no signs of hydrogen permeation. Subsequently, this heat pipe was operated in an FBC for several hundred hours with no sign of hydrogen permeation. However, the heat pipe was physically dropped into the bed where it stayed for an additional few hundred hours. Upon removal from the bed the heat pipe was tested and showed signs of hydrogen permeation (24). This permeation is attributed to the erosive and scrubbing action by the bed on the micro-thin oxide layer, which had been integrally formed on the 18-SR to produce the hydrogen permeation barrier. Upon removal of the barrier, hydrogen permeates the heat pipe wall. Thus, one concludes that an additional protective layer may be required to protect the thin hydrogen permeation barrier from erosion.

Tests at Argonne National Laboratory have shown that several ceramic coatings are capable of protecting the underlying metal

FIGURE 2-18:

HEAT PIPEMATERIAL SELECTION

<u>Desired Property</u>	<u>Material Considerations</u>
Low Cost	No Co, Mo, Ni, Minimum Cr
Oxidization Resistant	High (10-20% Cr) Low (1-2%) Al, Ti, Si
Low Hydrogen Permeation	Low (1-2%) Al, Ti, Si
Weldability	Ex. Low (<.05%) C, Low (< ½%) Al, Ti, Si
Sodium Compatibility	Ex. Low (<.05%) C, Low O ₂
High Strength	High (> .1%) C

CANDIDATE MATERIALS

<u>Alloy</u>	<u>Manufacture</u>	<u>Coils</u>	<u>Cost</u> <u>Tubing</u>
Incoloy 800	Huntington	\$3.32/lb	\$4.45/lb
18 SR	Armco	\$0.87/lb	\$1.86/lb

COMPOSITION

<u>Alloy</u>	<u>C</u>	<u>Mn</u>	<u>S</u>	<u>Si</u>	<u>Ni</u>	<u>Cr</u>	<u>Al</u>	<u>Ti</u>	<u>Fe</u>
Incoloy 800	.05	.75	.01	.5	32.5	21	.38	.38	46
18 SR	.05	.5	.01	1.0	.5	18	2	.5	77.5

STRENGTH

<u>Alloy</u>	<u>Temperature</u>	<u>UTS</u>	<u>0.2% YS</u>
Incoloy 800	650°C (1200°F)	358 MPa (52 ksi)	206 MPa (30 ksi)
Incoloy 800	816°C (1500°F)	172 MPa (25 ksi)	103 MPa (15 ksi)
18 SR	650°C (1200°F)	179 MPa (26 ksi)	152 MPa (22 ksi)
18 SR	816°C (1500°F)	75 MPa (11 ksi)	34 MPa (5 ksi)

FIGURE 2-19: 1000 HOUR LIFE TEST OF A GAS-FIRED
SODIUM HEAT PIPE AT 800°C SHOWING
NO HYDROGEN PERMEATION

substrate in coal gasification environments at 980-1070°C (1800-2000°F). These tests showed that several coatings on Incoloy 800 did not spall off during thermal cycling and indeed protected the base metal.

These coatings were aluminum oxide, spinel (magnesium aluminate), magnesium zirconate and 50% aluminum oxide/50% chromium oxide.

It is Thermacore's belief that the apparent good adherence of the ceramic coatings on Incoloy 800 was due to the thin layer of glassy oxide on the surface of the base metal, which is also the hydrogen permeation barrier. Thus, any of the materials capable of providing a hydrogen permeation barrier should also be a good substrate material for the application of ceramic coating, provided that the thermal expansion difference is not too great.

Thermacore has investigated potential coatings further and has identified Zirconium oxide and nickel aluminide along with application of several coatings in a graded structure as being possible candidates for coatings. Figure 2-20 presents the physical properties of these several coatings and the thermal expansion of the two suggested heat pipes materials from which a comparison can be made.

From Figure 2-20, Zirconia has a low thermal expansion which may prevent its successful use. Also, the thermal conductivity of Zirconia is too low to consider the use of a very thick coating. The 50/50 mixture of Aluminum/Chromium oxide has a thermal expansion which is probably too high. Also, the melting point of pure chromium oxide is quite low.

Thus, it appears that alumina, spinel, or the intermetallic nickel aluminide are several possible coatings which should be investigated further for potential use with Incoloy 800 or 18-SR heat pipes in an FBC.

FIGURE 2-20:

MECHANICAL PROPERTIES OF ABRASION

RESISTANT COATINGS

<u>Material</u>	<u>Melting Point</u>		<u>Thermal Expansion</u>		<u>Strength</u>		<u>Knoop Hardness</u>
	[°] C	([°] F)	[°] C	([°] F)	MPa	(ksi)	kg/mm ²
Aluminum Oxide	2072	(3762)	17.3 x 10 ⁻⁶	(9 x 10 ⁻⁶)	206	(30)	2260
Spinel	2135	(3875)	15.8 x 10 ⁻⁶	(8.8 x 10 ⁻⁶)	85	(12)	1000
Magnesium Zirconate	2120	(3848)					1000
50% Chromium Oxide- 50% Aluminum Oxide			21.6 x 10 ⁻⁶	(12 x 10 ⁻⁶)			
Zirconium Oxide	2766	(5010)	10 x 10 ⁻⁶	(5.6 x 10 ⁻⁶)			900
Nickel Aluminide	2038	(3700)	12.6 x 10 ⁻⁶	(7.0 x 10 ⁻⁶)	172	(25)	320-685
18 SR			14.4 x 10 ⁻⁶	(8 x 10 ⁻⁶)			
Incoloy 800			18 x 10 ⁻⁶	(10 x 10 ⁻⁶)			

Summary

The two-stage heat pipe has been selected as the best state-of-the-art heat transport option for 1985 demonstration. The preliminary investigation to date indicates that reliable, long life heat pipes may be built. The overall temperature drop of the heat transport system (from in-bed tubes to heater head tubes) is estimated to be approximately 33°C (60°F). The details of the system, including the tube layout, secondary heat pipe interface, etc. will require more work. Nonetheless, it is possible to design and build a heat pipe system which interfaces with the FBC and the Stirling engine in the manner described here and illustrated in other sections of this report.

2.2.6 Cost Estimate

Estimates for production costs of the two-stage heat pipe were based on the design which was detailed in the previous section. The quantities were based on a production volume suitable for 1000 engine modules per year (includes spares and scrap). The secondary heat pipe is estimated to cost \$16,500 and the primary array is estimated to cost \$8,500, or, for 254 heat pipes, \$33.50 each. The total heat transport cost is thus estimated to be \$25,000 per module or \$21.20 per kilowatt of heat transfer capability. These are FOB costs at the heat pipe factory. Appendix C gives the details of this cost estimate.

In comparing the primary heat pipe cost of roughly \$30 per tube to the estimate of about \$100/tube for the helium loop heat exchanger, four important differences stand out:

1. Since the heat pipes operate near atmospheric pressure, versus 10 to 30 atmospheres for the helium system, the wall thickness can be determined primarily by corrosion considerations, not on the basis of pressure loads. Thus the heat

pipe tubing weighs about 1/2 lb/ft versus about 3 lb/ft for the helium loop.

2. Because of its lower operating temperature and low stress level, a less expensive high-chrome alloy (~\$2/lb) can be used for the heat pipe instead of a high-nickel alloy (~\$5/lb) which must be used for the helium loop heat exchanger.
3. The large number of individual heat pipes (e.g., 250,000 per year) can benefit from mass production techniques as compared to the relatively small volume (e.g., 1000 per year) of helium heat exchangers.
4. A single failure in the helium loop will require shut down of the entire engine while the heat pipe has been designed to allow for up to 15% failure of heat pipes before requiring repairs. Thus, the cost of the helium loop manufacturing operation will be substantially higher than for heat pipes because of the requirement for 100% good products (welds, materials, etc.) compared to a slightly lower requirement for the heat pipes.

The cost estimate and personnel staffing is comparable to that found in the manufacture of evacuated devices (vacuum tubes) as is the type of equipment which is envisioned for the processing of heat pipe, i.e., a rotary processor. Recently, under DOE contract DE-AC04-77-CS04099, Cost Effective Solar Collector Using Heat Pipes, Thermacore performed a heat pipe cost analysis and preliminary design of a rotary processing unit. Appendix A contains a description of that processing unit. A similar rotary processor could be designed for the sodium heat pipes for the FBC-Stirling engine.

It is important to note that these costs assume a rather high production volume. The first few engines to be built would be hand fabricated since no production facility exist today which turns out a high volume rate of sodium heat pipes. These costs, after Phase II and III, in 1980 dollars, for the primary heat pipes are \$825 each, or \$209,000 for 254, and \$66,000 for the secondary heat pipe. Thus, the system cost is \$275,000 per engine module.

Mature Technology Projection

In order to project what a mature product will cost, it is typical to use a learning curve. Thermacore's personnel have considerable experience in large quantity volume production rates of vacuum devices, and smaller production quantities of heat pipes and heat pipe heat exchangers.

Experience shows that the labor content in terms of manhours is reduced by approximately 20% at the end of a period when the total quantity of identical items has been doubled. Likewise, during this time the material content will be reduced by approximately 5%. Thus, from this it is possible to estimate the cost of the mature product at any given time in the future based on some initial cost.

In addition to cost reduction by nature of a learning curve, the cost and performance of the product can be enhanced by ongoing development work. Increases in the reliability of the product will also fall out as part of the cost reduction of a learning curve. As technology continues to develop better materials, there may be quantum jumps in the reliability of the product due to materials change which would then be cause to start the learning curve type of estimating over with a new initial cost.

It is estimated that after five years production that factors will be such that a new cost structure will be generated. This will

come about because of better base material and/or coating materials and the possible need to go to a heat pipe system which is to work horizontal or vertically against gravity.

Based on an 80% labor and 95% materials learning curve, the estimated cost after five years is \$25.85 for the primary heat pipe (1980 dollars) and \$12,650 for the secondary heat pipe (1980 dollars) for a total of \$19,216 per engine module or \$16.28 per kilowatt of heat transfer capability.

Maintenance Costs

The actual maintenance costs of the heat pipe system are difficult to estimate because the system has not yet been built and tested. Nonetheless, for purposes of cost estimating, the design life is taken to be 32,000 hours for the secondary heat pipe and 16,000 hours for the primary heat pipe. As can be seen in Section 2.3.4.2 these intervals correspond to minor and major overhaul intervals of the engine. The cost of materials for these overhauls is taken as the learning curve mature technology production cost of the heat pipes. Labor is estimated at 80 manhours for primary heat pipe replacement and 80 manhours for secondary heat pipe replacement, at \$25/manhour. The total maintenance cost in 32,000 hours is estimated at \$31,800. Total power production is taken as 275 kW for 32,000 hours (i.e., 70% load factor) giving a total maintenance cost of \$0.0036/kWh or \$0.004/kWh rounded off.

It should be noted that the state-of-the-art Stirling heater head has a design life goal of 32,000 hours and a current projected life of 16,000 hours (See Section 2.3.4.2). Thus, in the near term the actual life of the secondary heat pipe may be less than 32,000 hours since the heater head is part of the hermetic secondary heat pipe. In estimating the cost of the replacement parts, it is assumed that the primary and secondary heat pipes will be returned to the

factory, where the primary heat pipes will be scrapped and the pressure parts (heater tubes, regenerator caps, and cylinders) of the secondary heat pipes will be replaced. It is assumed that the scrap and salvage value of the returned parts will enable the replacement parts to be sold at approximately original equipment cost.

2.3 ENGINE CONCEPTUAL DESIGN

This section covers the work performed at United Stirling on a conceptual design study for a large stationary Stirling engine for power generation in the 500 to 3000 hp size range. During the initial stages of this study, before the evaluation of various combustion and heat transport systems was completed, United Stirling's engine design efforts were concentrated on the conceptual design of a large liquid (or gas) fueled engine (Referred to as the direct fired engine). This was done for two reasons; first the design of a large Stirling engine capable of burning liquid and gaseous fuels derived from coal was included within the workscope. Second, it was believed that the basic engine structure would likely be the same for both liquid and solid fuel burning Stirlings - only the heater head and combustion systems would likely differ. The following section (2.3.1) describes this initial design work on the basic "liquid fueled" engine design. Most of the design features described in that section apply to both the liquid fueled and solid fueled engines. Engine thermodynamic performance estimates are also included in that section and are intended to serve as estimates for both the direct fired and the heat pipe heated engine. Then Section 2.3.2 describes the specific design features proposed for a Stirling engine fired by a fluidized bed combustor with a 2-stage sodium heat pipe system to transport heat from the FBC to the engine heater tubes. Section 2.3.2 represents work performed by United Stirling after the selection of the FBC and sodium heat pipe as the favored combustion and heat transport systems. It concentrates on the engine design features required for a heat pipe heater head to interface with the FBC.

2.3.1 Liquid/Gas Fueled Engine Design

2.3.1.1 Initial Selection of Engine Type

Based on previous experience with both single-acting and double-acting Stirling engines it was decided that a 4-cylinder double-acting mechanical arrangement similar to the USS P75 (4-275) engine would be the best choice, with the basic engine module providing 500 hp net power. This implies 125 hp per cylinder. Then, 1000-3000 hp units could be obtained by incorporating 2 - 6 basic engine modules.

The twin crank arrangement was selected over the V arrangement for the following reasons:

1. The twin crank, parallel cylinder arrangement allows complete engine balancing.
2. Parallel cylinders reduce manufacturing cost by simplifying machining. (V designs require machining operations at different angles.)
3. Better interface with heater head. This is true for the direct fired engine as well as the heat pipe heated engine.
4. More options in number and location of regenerator-cooler units.
5. Twin crankshafts should be less expensive to produce than a single crankshaft because they are shorter, more rigid and involve less complex machining.

Advantage #3 is nullified for the oil-fired case when the engine becomes so large as to require individual combustion chambers for each cylinder. However, at the 125 hp/cylinder level there is not enough justification for the additional cost and complexity of one combustion system per cylinder.

The primary disadvantage of the U-4 design compared to the V-4 is the gear loss between crankshafts and the possibility for gear noise. However, this is considered a small disadvantage compared to the advantages.

Thus the concept chosen for the basic engine module is a 4-cylinder twin-crank parallel cylinder arrangement with the four cylinders in a square cluster formation and with regenerators and coolers located on a circle outside the cylinders (See Figure 2-21). The multi-module engine could then be an in-line configuration with several modules placed end to end on a three-crank V-configuration with the center crank common to the basic engine modules.

2.3.1.2 Scaling

The selection of main engine dimensions was done by scaling from the USS P75, which is also based on a direct fired liquid fuel combustion system. Helium, rather than hydrogen, has been selected as the engine working fluid to avoid possible problems of hydrogen diffusion into a sodium heat pipe, and since helium would avoid any concern over the risk of flammability.

The power level for one module was chosen to be approximately 500 hp (368 kW) as a starting point for scale up of the P-75 engine. Using helium as the working fluid and choosing 900 rpm as the reference speed, it was found that a scaling factor of 2.3 would be suitable. Heater temperature was originally fixed at 760°C but was later decreased to 720°C because of the material chosen for the hot parts of the engine. Full load mean operating pressure was set at 15 MPa (2030) psia.

Figure 2-22 gives the results of scaling which were subsequently used in the analysis and design work.

FIGURE 2-21: P-75 CROSS SECTION

Figure 2-22: ENGINE SCALING RESULTS

Number of cylinders per module	4
Bore	200 mm
Stroke	110 mm
Swept volume per cylinder	3,455 cm ³
Number of heater tubes per cylinder	24
Outside diameter of heater tube	14.5 mm
Inside diameter of heater tube	10 mm
Heater tube length (Second row of tubes are finned)	850 mm
Number of regenerators per cylinder	4
Diameter of regenerator	120 mm
Height of regenerator	140 mm
Number of coolers per cylinder	4
Length of cooler	210 mm
Inner diameter of the heater cage	500 mm
Outer diameter of the heater cage	1,000 mm
Gap between heater tubes in first row	1.86 mm
Cooling water outlet temperature	50°C
Cooling water flow at 900 rpm	26 kg/s
Heater head temperature	720°C

Comments on Scaling

The dimensions shown in Figure 2-22 apply to the large scaled-up direct fired engine. The design details for the large heat pipe heated engine could not be determined because a complete Stirling cycle design optimization is required, which was beyond the scope of work for this contract. However, it is possible to comment on the direction that the parameters will most likely go for the large heat pipe heated engine when the optimization is performed.

The number of cylinders, their bore, stroke and swept volume will not substantially change. (Both engines would have the same block pistons and drive unit.) The number, diameter and length of heater tubes will change to some extent. This is because the internal heat flux will be limiting, and thus the external fins will be eliminated, and the surface area and dead volume in the heater head will decrease slightly. It appears that 12 regenerators per cylinder are more attractive for the heat pipe heated engine (See Section 2.3.2) and thus their diameter will change in order to maintain an essentially equal cross-sectional area per cylinder. The height of the regenerator will not substantially change. Accordingly, there will be twelve coolers per cylinder and their dimensions will change. Finally, the heater tube arrangement would change and thus the dimensions given for direct fired tube arrangement are not applicable to the heat pipe heated engine.

Even with all these changes which will occur when the complete optimization is performed, it is important to keep in mind that once the major parameters are determined they will be the same for both the direct fired and the heat pipe heated engine. Only the heater head will be different for the two engines. Thus, it is reasonable to suspect that no gross changes to the cycle will occur when the engine is optimized for heat pipe heat input.

When scaling up an engine to a much higher power output it is important to keep all void volumes to a minimum. Large void volumes at the cold side (cold connecting ducts between cooler and cylinder cold space) penalize power, and to a small degree efficiency. Void volumes at the hot side (manifolds between cylinder hot space and heater and between heater and regenerator) penalize both power and efficiency. A large void volume at the regenerator side is the most critical.

Based on the chosen concept, the following can be said about regenerator size and number: With decreasing numbers of regenerators, the regenerator diameter increases and so the regenerators will be moved away from the cylinders. This increases the heater cage outer diameter, requires longer cold connecting ducts and results in larger manifolds and void volumes between heater and regenerators. A large number of (smaller) regenerators increases the number of parts but makes it easier from a stress point of view to design the regenerator housing with its flat hot surface.

The selected number of regenerators (4) can be considered the best compromise between compactness and performance.

2.3.1.3 Performance Analysis

Performance calculations have been conducted for the engine module, and the results are shown in Figures 2-23 and 2-24. There are no auxiliary losses included and the efficiency is based on the heat supplied to the heater tubes.

In order to achieve 60 Hz frequency the speeds considered are 600, 720, 900 and 1200 rpm, corresponding to 12, 10, 8 and 6 poles on the generator. The final choice of speed must of course be based on economical factors where investment cost is balanced against fuel cost. (Power goes up and efficiency goes down with increased speed.)

ASSUMPTIONS:

1. No auxiliaries and no combustion losses
2. Heater temperature - 720°C
3. Cooler exit temperature - 50°C

FIGURE 2-23: BRAKE SHAFT POWER P vs. RPM

ASSUMPTIONS:

1. No auxiliaries and no combustion losses
2. Heater temperature - 720°C
3. Cooler exit temperature - 50°C

FIGURE 2-24: BRAKE THERMAL EFFICIENCY η vs RPM

If 900 rpm is chosen, the engine module power will be 615 hp (brake) with helium at full pressure (15 MPa) and after reduction for auxiliaries this will be more than enough for the required 500 hp.

At 900 rpm the efficiency increases from 39% to 45% when going from helium to hydrogen. Thus the factors which led to the selection of helium substantially penalize efficiency.

The very good part load efficiency (lower pressure) can be explained as follows: The relative increase in parasitic losses, like heat conduction from the hot end to the cold end, are compensated by the lower temperature drop between heat exchanger walls and working media at lower heat loads.

A parametric study was performed in order to identify the influence on performance due to a change in heater tube temperature and/or a change in cooling water temperature. The results are shown in Figures 2-25 and 2-26.

In order to show the importance of equal heat input to all four cycles in a double-acting engine, a calculation was performed where the heat input to one cycle is decreased and the resulting heater temperature is calculated (See Figure 2-27). From Figure 2-27 it can be seen that a 20% decrease in heat input to one cycle will cause a 300°C temperature decrease on the heater of that cycle. In that case, the power output from that cycle will only be 30% of the nominal power. The same thing also applies to a multi-module engine with one common heat source, where the heat input to each module is not temperature controlled. The physical explanation is that if the heat input to one module is cut off, then that module will act as a cooling machine driven by the other modules.

ASSUMPTIONS:

1. Working Fluid - Helium
2. Mean Pressure - 15 MPa
3. Speed - 900 RPM

FIGURE 2-25: INFLUENCE OF HEATER TEMPERATURE ON POWER AND EFFICIENCY

FIGURE 2-26: INFLUENCE OF COOLER
EXIT TEMPERATURE ON
POWER AND EFFICIENCY

FIGURE 2-27: COUPLING BETWEEN HEAT SUPPLY TO ONE CYCLE AND HEATER TEMPERATURE AND POWER OUTPUT

2.3.1.4 Hot Engine System Conceptual Design

Heater

The oil/gas direct fired heater consists of four identical quadrants with four regenerator housings in each quadrant (See Figure 2-28). The first combustion gas passage is formed by gaps of equal width along the inner involute shaped part of the heater tubes. The second combustion gas passage is between the straight outer parts provided with extended surface (fins) (See Figure 2-29). The inner tube ends are brazed or welded to a manifold which in turn is welded to the cylinder. The outer tube ends can either be connected to regenerator manifolds, or brazed or welded directly to the regenerator housings. In the latter case a certain portion of the heater tube will be ineffective in heat transfer due to the necessary tube bends, but the manifolds can be eliminated.

The cylinders are bolted to the cylinder block with bolts or studs extending through the water jacket down to an upper extension of the cross head liner, thus forming a closed pressure vessel transmitting only the net piston forces to the crankcase. The regenerator housings are connected to the duct plates below the coolers with long bolts or studs passing through the water jacket.

Regenerator

The regenerator could be of the same design as for the P75 engine, i.e., fine mesh stainless steel gauze layers, stacked in a thin cylindrical shell. A possibly cheaper manufacturing method, not yet tested, could be achieved by rolling the matrix material.

SECT. THRU PREHEATER

2-95

CRANKCASE APPROX. ENVELOPE

HEATER, TOP VIEW

FIGURE 2-28: DIRECT
FIRE HEATER HEAD -
PLAN VIEW

FIGURE 2-29: EXTERNAL
HEAT SYSTEM - ELEVATION

Gas Cooler

The gas cooler is similar to the P75 design, i.e., small diameter tubes joined to circular end plates. Here, the scaling rules cannot be used with any precision, and the details of tube size, length and number must await a design optimization.

Stress Analysis/Material Selection

Cylinder and regenerator housings for the P40 and P75 engines have previously been analyzed using a finite element stress calculation program. Based on these results an estimate of required thickness for this concept was made. Materials containing large amounts of strategic elements like cobalt were eliminated. The choice was then between high-strength nickel-base materials or lesser strength iron-base materials.

Designing for a required life of 30,000 hrs and using an allowable stress equal to $2/3$ of the creep-rupture stress, several candidate materials were examined for use in the hot cylinders and regenerator housings. Initial material selection studies considered Inconel 713C with an allowable stress of 265MPa and Climax Molybdenum XF-815 with an allowable stress of 100MPa for pressure vessel temperatures of 720°C (corresponding to a 760°C heater tube temperature) Inconel 718C, which is much easier to braze and weld than Inconel 713C, was finally selected as the preferred material, but the heater tube temperature had to be restricted to 720°C, which gives a pressure vessel temperature of about 675°C. At this temperature the allowable stress for Inconel 718C is 250MPa.

With Inconel 718C for cylinders and regenerator housings the following thicknesses apply:

Cylinder:	Cylindrical Part	8 mm thickness
	Rounded top	8 mm thickness
Regenerator housing:	Cylindrical part	6 mm at cold side
	Cylindrical part	12 mm at hot side
	Flat top	8 mm with some increase around holes

Inconel 718 sheet material was selected for the piston dome, which also operates at 675°C max. For the dome, local buckling of the cylindrical part is critical. With dome fitted with one or two radiation shields it seems possible to accept a thickness of 3 mm or slightly above. It should be noted that before final dimensions are decided, a more accurate stress calculation could give a different result than simple scaling. It should also be noted that a stronger material, even if more expensive, could pay dividends in less weight and smaller heat losses to the cooling water.

The heater head tubes operate at 720°C with an average internal pressure of 15MPa. Inconel 625 was selected as the primary material choice for heater tubes because it provides adequate creep strength under these operating conditions and it contains no cobalt. The second choice for heater tubes is Sandvik 12RN72 whose composition is 25Ni, 19Cr, 1.4Mo, and 0.5Ti.

2.3.1.5 Cold Engine System Conceptual Design

Cylinder Block

The cylinder block is divided into four equal quadrants (See Figure 2-30) for ease of manufacture. Also with one single cylinder block and four individual cross-head liners between the crankcase and the cylinder block, excessive stresses could be imposed on the cross head liner flanges due to liner length tolerance if the four liners were bolted between the crankcase and the cylinder block.

The cylinder extends down into the cylinder block with a wet liner guided by the cross head liner (See Figure 2-31). The cooling water enters the cylinder blocks at the center of the engine, flows around and cools the cylinder liner, and is then split up with one half of the cooling water in each quadrant flowing through two coolers (Figure 2-30). The cooling water exits into an outlet connecting two adjacent quadrants. In this way the cooling water is collected at four points around the periphery of the cylinder block.

With one common cylinder block (as in the USS P-75 engine), loads between cylinder and duct plates corresponding to the cold connecting duct areas are taken within the cylinder block. When splitting up the cylinder block into quadrants, it would be desirable to be able to remove one quadrant without interfering with the others, but this is not easy. A few possible alternatives have been examined and are described below.

Alt A (Fig. 2-32): This is the alternative that is shown on the detailed drawings. In this case the adjacent heater quadrant must be removed before the

FIGURE 2-30: CYLINDER BLOCK CROSS SECTION

FIGURE 2-31: COLD ENGINE SYSTEM CROSS SECTION

FIGURE 2-32: QUADRANT ARRANGEMENT

ALTERNATE A

cylinder block quadrant with its heater can be removed. The loads from four of the cold connecting ducts must be taken by some simple connection between adjacent cylinder block quadrants.

Alt B (Fig. 2-33):

In this alternative it is possible to remove one cylinder block quadrant with its heater without disturbing the adjacent heaters, but the cylinder block must be of an odd shape and again the loads from all four cold connecting ducts must be taken between adjacent cylinder block quadrants.

Alt C (Fig. 2-34):

In this case the loads from the cold connecting ducts are taken within the cylinder block quadrant, but the adjacent heater quadrant must be removed before removing the cylinder block quadrant, which also here has become an odd shape.

Alt D (Fig. 2-35):

This alternative contains the smallest cold connecting duct volume but requires an involute heater with 90° between the two tube ends, which complicates the tube manufacturing.

Alt E (Fig. 2-36):

This alternative does not have an involute heater as in alternatives A through D. The heater tubes extend radially from the cylinder manifold to the regenerator housings. By necessity the gap between the heater tubes

FIGURE 2-33: QUADRANT ARRANGEMENT
ALTERNATE B

FIGURE 2-34: QUADRANT ARRANGEMENT
ALTERNATE C

FIGURE 2-35: QUADRANT ARRANGEMENT
ALTERNATE D

FIGURE 2-36: QUADRANT ARRANGEMENT
ALTERNATE E

will vary but this could be handled by special surface extension. Each cylinder block quadrant carries its own heater head quadrant, making disassembly of one quadrant easy. The load from the cold connecting duct can be accomodated.

The main drawback is the much larger void volume on the cold side as compared to the other alternatives, but it is felt that this alternative is interesting enough to render further investigation.

Cross Head Liner

The cross head liner guides the cross head, connecting the piston rod and the connecting rod. The cross head liner guides the cylinder liner and acts as a support for the piston rod seal housing. Its lower flange is bolted to the crankcase and is loaded by only the net piston force. Engine disassembly can be done by removing the rod pin when the cross head is in its upper position, unbolting the cross head liner from the crankcase, leaving the connecting rod attached to the crankshaft.

Cold Connecting Duct

The cold connecting ducts are simple straight tubes connecting the duct plates with the cylinder cold space through corresponding holes in the upper parts of the cross head liners. When assembling the engine, the tube is pushed into the duct plate, and with the duct plate in place the tube is guided into its recess in the cross head

liner and locked in place by a retaining ring (See Figure 2-31).

Seals

First, it must be noted that there is no experience to date in scaling seals to a much larger size engine. Also there is no demonstration of service life as high as 30,000 hours for any one seal. United Stirling has concentrated their effort on development of sliding seals for the double acting engines. The roll-sock seal used on single-acting Stirling engines was abandoned several years ago, because these seals were very sensitive to inclusions of foreign matter during manufacture and to gas diffusion through the seal causing problems at pressure changes.

For long service life requirements it might be possible to change to clearance seals or gap seals for the piston rings and the piston rod cap seal. The result could be less friction but higher leakage. However, no in-house experience exists.

The most critical sealing problem is the piston rod seal. This seal must prevent the loss of working gas, and at the same time prevent crankcase oil from migrating into the cycle where it could cause fouling of the regenerator with decomposition products and contamination of the working gas. The proposed piston rod seal, a "pumping Leningrader seal", has proved to be a very effective seal both with regard to keeping the volume above the seal dry from oil and in sealing the working gas (39). More testing is needed however, since the rod diameter is so much larger than on seals currently in use. The seal housing can float in a radial direction and is intended to be bolted to the upper part of the cross head liner after the piston rod has been guided by the piston and cross head. This is to minimize any radial loads on the seal.

2.3.1.6 Air Preheater - Liquid Fueled Engine

With a direct fired liquid fuel combustion system the engine must incorporate an air preheater (See Figure 2-29). Given a regenerator pitch circle of 1,000 mm, which coincides with the outer heater cage diameter, the maximum cylinder block diameter will be approximately 1,200 mm. In order to space the engine modules as close as possible in a multi-module engine, it is desirable that the air preheater not exceed the 1,200 mm diameter. This means that the preheater matrix must be located above rather than around the heater head. This was considered acceptable since the engine height is not critical.

The preheater was scaled from the P75 preheater, which also operates at 720°C heater head temperature. Using formulas for heat transfer and pressure drop derived from tests of P75 preheater plates, the new preheater was scaled keeping the pressure drop the same as for the P75 preheater.

The scaling resulted in the following dimensions:

number of plates	1,000
matrix outer diameter	1,020 mm
matrix inner diameter	770 mm
matrix effective height	370 mm
preheater plate thickness	0.05 mm

2.3.2 Engine Design for FBC with Sodium Heat Pipe

2.3.2.1 Initial Considerations

When a Stirling engine is integrated with a fluidized bed

combustor via a sodium heat pipe, the engine can either be placed with its heater up along one side of the FBC (See Figure 2-37), or be mounted upside down on top of the FBC (See Figure 2-38).

In either case a two-stage gravity assisted heat pipe may be used. The primary heat pipe system consists of a number of small diameter heat pipes extending from their upper condensor ends vertically down into the fluidized bed, with their lower ends vertically unconstrained, transporting heat from the fluidized bed combustor to a secondary heat pipe evaporator. The secondary heat pipe, with the engine heater forming the condenser, can be made either as one heat pipe per engine module or as one heat pipe per heater head quadrant.

The twin crank parallel cylinder arrangement, U4, is used for the heat pipe concept, as it is for the direct oil-fired concept. An upside-down mounted engine requires a dry sump arrangement, with the oil drained off from the cross head liner at the main piston rod seal. The cylinder block may be split into four quadrants for ease of manufacture and installation, as is also the case with the direct oil-fired concept. In this way the block quadrants can be individually removed if one heat pipe per quadrant is chosen.

The most suitable multi-module approach is the in-line engine arrangement, preferably with the power output shaft located along one side of the engine modules, making it possible to remove each engine module easily, and requiring no change in module design if more modules are added. See Section 2.3.3 for more details.

With a direct oil-fired heater there are important limitations in the heater tube locations to fulfill heat transfer requirements. With a heat pipe there are no longer any restraints on the heater tube

FIGURE 2-37: SIDE MOUNTED ENGINE

RICARDO

FIG No: 4

DRG No: 577114

DATE: 1-4-80

UNITED STIRLING
LARGE ENGINE - 500 HP MODULE

CROSS SECTION
SINGLE MODULE FLUIDISED BED

FIGURE 2-38: TOP MOUNTED ENGINE

shape and location as long as enough tube length or tube surface area is maintained.

The alternative engine concepts considered most interesting for indirect heating via heat pipe are:

- i) Twelve regenerators per cylinder located on a circle around the cylinder.

The required cylinder-to-cylinder distance is approximately 500 mm.

- ii) Four regenerators per cylinder with the same cylinder/regenerator arrangement as for the direct oil-fired concept Alt. E.

The required cylinder-to-cylinder distance is approximately 350 mm.

The alternative engine-secondary heat pipe combination concepts which have been studied are:

Alternative	Reg/Cyl	Engine Position	Number of Secondary Heat Pipes per Engine Module	Figure Number
Alt. 1	12	Top mounted	4	2-39
Alt. 2	12	Top mounted	4	2-40
Alt. 3	12	Top mounted	1	2-41
Alt. 4	12	Side mounted	1	2-42
Alt. 5	4	Top mounted	4	2-43
Alt. 6	4	Top mounted	1	2-44
Alt. 7	4	Side mounted	1	N.A.

ONE HEAT PIPE PER HEATER HEAD QUADRANT
12 REG/CYL ENGINE

FIGURE 2-39: ALTERNATE 1

ONE HEAT PIPE PER HEATER HEAD QUADRANT
12 REG/CYL ENGINE

FIGURE 2-40: ALTERNATE 2

ONE HEAT PIPE PER ENGINE MODULE
12 REG/CYL ENGINE

FIGURE 2-41: ALTERNATE 3

ONE HEAT PIPE PER ENGINE MODULE
12 REG/CYL ENGINE

FIGURE 2-42: ALTERNATE 4

ONE HEAT PIPE PER HEATER HEAD QUADRANT

4 REG/CYL ENGINE

FIGURE 2-43: ALTERNATE 5

ONE HEAT PIPE PER ENGINE MODULE
4 REG/CYL ENGINE

FIGURE 2-44: ALTERNATE 6

2.3.2.2 Twelve Regenerator per Cylinder Concept

Hot Engine System Conceptual Design

In this concept, the required regenerator area for each cylinder has been split into twelve regenerators located on a circle around the cylinder (Fig. 2-45). The required cylinder to cylinder centerline distance is 500 mm, as compared to 350 mm for the direct oil-fired concept.

The heater tubes are bent in a hair-pin shape (Fig. 2-39) and extend radially from the cylinder to the regenerators. They are welded or brazed to cylinder manifolds and regenerator housings.

The regenerators and gas coolers are similar to the ones for the direct oil-fired 4 regenerator per cylinder concept, but of smaller diameter.

When a hot partition wall with bellows is used to seal off the secondary heat pipe from the main engine structure (Fig. 2-39), the cylinder load is taken by long studs through an inner clamping ring, which also carries part of the load on the regenerator housings. The other half of the regenerator housing loads is taken by a split clamping ring on the outer face of the regenerator housings (Fig. 2-45). A mounting plate, clamped between the clamping rings and the cylinder block, holds the heater head and the secondary heat pipe together. Each cylinder and regenerator housing carries an O-ring to seal off the cooling water.

When no partition wall and bellows are used, the cylinder and regenerator housings are brazed or EB-welded to a thick mounting plate, one for each heat pipe, to which also the secondary heat pipe

FIGURE 2-45: 12 REG/CYL ENGINE
WITH HEAT PIPE

structure is welded (Fig. 2-40 and 2-41). The studs taking the cylinder and regenerator housing loads are locked and sealed in this plate. Since the cylinder and regenerator housings are located relative to each other by the mounting plate, they require a fairly large radial clearance in the cylinder block. To seal off the cooling water, only one O-ring located outside the regenerator housings is needed between the mounting plate and the cylinder block quadrants.

Cold Engine System Conceptual Design

Cylinder Block: For ease of manufacture the cylinder block is split into four equal quadrants. Also with each block quadrant attached to an individual cross head liner no stresses are imposed when the liners are bolted to the crank case, unless some other restriction is added.

This is the case when one heat pipe per engine module is used, which requires one mounting plate covering all block quadrants. Since the mounting plate is not a weak structure, shims are required between the cross head liner flanges and the crankcase to take care of manufacturing tolerances. The same will be the case if one common engine block is chosen.

The cooling water enters the cylinder block at the center of the engine (Fig. 2-46), divides into two branches through the coolers with part of the water cooling the wet cylinder liner, and exits 180° from the inlet.

Cylinder Liner and Cross Head Liner: If the cross head liner and cylinder liner are offset, radial loads will be imposed on the piston rod seal. To increase service life, cross head and cylinder liners

FIGURE 2-46: CYLINDER BLOCK

should be integrated to have better control over bore eccentricities.

Cold Connecting Duct Plate: With the regenerators and gas coolers located around the cylinder, the cycle gas to and from the coolers must be collected in an annular duct that connects to the adjacent cylinder cold space. This duct must be designed so as to avoid a skewed gas distribution through the regenerators, otherwise the total regenerator efficiency will drop. Analysis shows that this requires a rather large duct cross sectional area, with a void volume that is much larger than that for the direct oil-fired Alt. A concept, where each regenerator and cooler carried its own cold connecting duct. However, the larger void volume can be compensated for by increasing the engine displacement slightly. The engine efficiency is affected very little.

The connection between the duct and the cylinder cold space (Figure 2-47), and the assembly procedure, is similar to that shown for the direct oil-fired concept (Figure 2-31). A gas pressure load, corresponding to the duct cross sectional area pushing the cylinder block quadrants apart, must be accommodated. Means for accomplishing this are not shown but could easily be handled by some connection between the quadrants, e.g. lugs and bolts.

If one heat pipe per engine module is chosen, one common cylinder block might be preferable, in this way taking care of the gas pressure loads from the cold connecting ducts.

The cylinder block or water jacket and the cold connecting duct plate could possibly be integrated, thereby eliminating some of the O-rings.

The 12 regenerator per cylinder arrangement can also be adapted to a direct fired liquid or gas fueled engine. Figure 2-48

FIGURE 2-47: COLD ENGINE SYSTEM, 12 REG/CYL ENGINE

FIGURE 2-48: 12 REG/CYL. DIRECT
FIRED CONCEPT

illustrates the manifolding and heater tube connections between cylinder and regenerators that would be required to form a single heater tube cage serving a 4 cylinder module in a manner similar to that currently employed in the USS P-75 engine. The heater tubes extend radially from the cylinder manifolds and end in the regenerator housings. The last portion of the tubes at the regenerator end is ineffective for heat transfer in order to seal off the combustion gases.

2.3.2.3 Four Regenerators Per Cylinder Concept

Among the direct oil-fired heater concepts, (See Section 2.3.-1.4) one heater head, Alt. E, was shown as a possible candidate. This alternative can also be used in combination with an indirect heat transport system such as a heat pipe since the heater tubes extend radially from cylinder manifolds to regenerator housings, and so do not cross over the adjacent cylinder block quadrants. (Fig. 2-49) Thus this alternate is suitable even if one heat pipe per cylinder is chosen as the best concept.

With this type of heater the cycle gas from one cylinder, after passing through the cooler, is collected in a manifold and is ducted to the cold space of an adjacent cylinder. The void volume of the cold connecting duct will be comparable to that of the 12 regenerators per cylinder concept. A gas pressure load corresponding to the duct cross sectional area must be accommodated. This can probably be done by bolting the duct pipe to the two adjacent cylinder block quadrants.

The attachment of the heater head and the secondary heat pipe follows the same procedure that was outlined for the Twelve Regenerator per Cylinder Concept. Also what was said under sub-headings Cylinder Block and under Cylinder Liner and Cross Head Liner applies here as well.

FIGURE 2-49: 4 REG/CYL ENGINE WITH HEAT PIPE

2.3.2.4 Secondary Heat Pipe Conceptual Design

Alt.- 1 Top Mounted 12 Reg./Cyl. Engine 1 One Heat Pipe per Heater Head Quadrant (Figure 2-39)

In this concept a partition wall, acting as a seal between the hot condenser space and the cylinder and regenerator housings is used to provide hermetic sealing of the secondary heat pipe chamber. Since the outer supporting heat pipe structure at the cylinder block end is at a lower temperature, a bellows must connect this structure with the partition wall to accomodate the differential thermal expansion.

A cold mounting plate covers the entire cylinder block quadrant. The heater head and the partition wall with its bellows are permanently joined together, with the cylinder clamping ring in place. Then the insulation, the outer clamping parts and the mounting plate with the heat pipe supporting structure between mounting plate and bellows are put in place. The mounting plate is attached to the cylinder and regenerator clamping rings with a number of screws. Finally the heat pipe shell and the bellows can be welded together. The heater head and the heat pipe can now be installed as one unit. All the bolts or studs taking the loads from the cylinder and regenerator housings have to be tightened from underneath the cylinder block.

Some bending loads will be imposed on the heater tubes at the regenerator because the tubes pass through and are sealed to the partition. However, it might be safer from sealing standpoint to incorporate a ring to which the partition wall is welded or brazed. Between this ring and the regenerator housing a larger number of smaller diameter tubes can be used, decreasing the tube stresses. The ineffective part of the tubes between the partition and regenerators decreases the power and efficiency only slightly.

The shape of the outer heat pipe structure can preferably change from a circular section at the bellows and cylinder block (Fig. 2-45) to a nearly square section at the primary heat pipe interface.

Comments:

With the design proposed in this concept, one heat pipe per engine module could also be arranged by using one common mounting plate and four bellows attached to a horizontal partition wall.

However, the mounting plate cannot be a weak structure and so the build-up of tolerances of the different parts between mounting plate and crankcase require some type of compensation, e.g. shims, in order not to impose high stresses on the crosshead liner flanges or the mounting plate.

Also, if the heat input to the many primary heat pipes in the fluidized bed is about equal, and if the heat transfer at all the interfaces between primary and secondary heat pipes is equal within certain limits, then the four cycles of the engine module should have the same heat input, and so four individual heat pipes, one for each heater head quadrant, can be accepted. From assembly and service standpoints it is the preferred choice.

Alt. 2 - Top Mounted 12 Reg./Cyl. Engine

One Heat Pipe per Heater Head Quadrant (Figure 2-40)

The clamping rings and mounting plate in Alt. 1 can be combined in a thicker plate to which the cylinder and regenerator housings are brazed or EB-welded. The secondary heat pipe structure is also welded

or brazed to this plate.

Since the sodium vapor cannot be allowed to condense on the cold faces at the mounting plate, and since the engine is mounted upside down, it is possible to introduce insulation and an inert gas between the condenser section and the cold face. This gas acts as a thermal barrier or insulation between the cold cylinder block and the sodium vapor.

This proposed concept is a simple approach of attaching the secondary heat pipe, eliminating both partition wall and bellows. The entire tube length is effective in heat transfer, and no additional thermal stresses are imposed on the tubes.

Alt. 3 - Top Mounted 12 Reg./Cyl. Engine

One Heat Pipe per Engine Module (Figure 2-41)

This concept is similar to Alt. 2 except that all cylinders and regenerator housings are sealed to one common mounting plate covering all four quadrants. If desired, one common cylinder block could be used.

Alt. 4 - Side Mounted 12 Reg./Cyl. Engine

One Heat Pipe per Engine Module (Figures 2-37 and 2-42)

Alts. 1 through 3 show the engine mounted on top of the FBC. However, it is possible to install the engine along one side of the FBC using one secondary heat pipe per engine module, if the engine heater is provided with a partition wall such that with the engine slightly tilted (Fig. 2-37), the condensate drains back to the primary/secondary heat pipe interface.

Parts of heater tubes pass through circular plates to which bellows are connected, the other end of the bellows being attached to the hot partition wall. Bellows also connect the cylinders to this wall.

As in Alt. 1 there is an ineffective part of the heater tubes just above the regenerators. Because of the small lateral stiffness of the bellows no additional bending stresses will be imposed on the heater tubes.

Alt. 5 - Top Mounted 4 Reg./Cyl. Engine

One Heat Pipe per Heater Head Quadrant (Figure 2-43)

This concept is based on the Alt. E concept for the direct oil-fired heater. The assembly of the heater head and heat pipe, and the installation, follows the same procedure as was outlined for Alt. 1.

Since the regenerators are not located symmetrically around the cylinders, as when 12 regenerators are used, bellows have to be connected between the partition wall and both cylinder and regenerator housings (Fig. 2-43); e.g., 5 bellows are required per quadrant. In this alternative the entire heater tube length is effective in heat transfer and no additional thermal stresses are imposed on the tubes.

Comment:

The comments under Alt. 1 apply also for Alt. 5.

Alt. 6 - Top Mounted 4 Reg./Cyl. Engine

One Heat Pipe per Engine Module (Figure 2-44)

In this concept the heater head and the secondary heat pipe are attached to a mounting plate in the same way as in Alt. 2 and 3, making it possible to use inert gas as a thermal barrier between the engine cold face and the condensor section, eliminating the partition wall and bellows.

Alt. 7 - Side Mounted 4 Reg./Cyl. Engine
One Heat Pipe per Engine Module

As for the 12 Reg./Cyl. Engine, Alt. 4, the 4 Reg./Cyl. engine also can be installed in an almost upright position on one side of the FBC.

A partition wall as for Alt. 4 (Fig. 2-42) has to be introduced, with the heater tubes connected (in clusters of six) to circular plates to which bellows are connected, the other end of the bellows being attached to the partition wall. From the lower face of the circular plates the tubes can spread out and connect to the regenerator housings.

2.3.2.5 Summary and Comments

Each of the alternative schemes described above has certain advantages and disadvantages. More detailed design investigations are required before making a final choice of the best design scheme. However, based on the work to date, United Stirling's preference is design alternate 2 followed in order of preference by alternates 1, 3, and 6. The reasons for this preference are summarized below.

Mounting Preference

The top-mounted (upside down) engine is favored over the side-mounted design. Although the side-mounted engine concept seems attractive from the viewpoint of improving accessibility for replacement of primary heat pipes, the secondary heat pipe structure is much larger, and thermal growth in both vertical and horizontal directions must be considered. Also, a large number of sealed bellows are required for the side mounted approach. Furthermore, the top mounted engine permits the use of the inert gas as a thermal barrier between the sodium vapor and the baseplate (which bolts to the water jacket). This eliminates the need for a hot hermetic seal, thereby

eliminating the possibility of introducing thermal stresses on hot engine parts and thereby eliminating the need for bellows.

With the engine mounted upside down there is space provided by the crosshead liners between the crankcase and the cylinder block to support the engine on the crankcase. If the power output shaft is placed alongside the engine modules, an individual module can more easily be removed for service. If for example a piston ring or a piston rod seal is to be replaced, then the bolts connecting the cross-head liners to the crankcase and the connecting rod pins are removed. The crankcase of that module is lifted up, and moved aside, and finally the nuts for the cylinder bolts are loosened. The crosshead liner can now be removed, followed by the piston and piston rod seal housing. However, provisions must be made so that the cylinder block can rest on some supporting structure when the crankcase is removed.

Engine Design Preference

The 12 regenerator per cylinder design is preferred for a heat pipe engine because it preserves symmetry, simplifying the arrangement of heater tubes between cylinder and regenerator caps; and it reduces the required wall thickness in regenerator housings, thereby reducing the conduction heat leak to the cold end.

Secondary Heat Pipe Preference

A secondary heat pipe employing one heat pipe per heater head quadrant is more attractive from manufacturing, assembly and service standpoints. Also, provided the heat input at the secondary heat pipe evaporating section does not differ too much, all four engine cycles will receive about the same heat input. One heat pipe per cylinder, together with individual cylinder block quadrants eliminates stresses

in flanges at the crankcase. The twelve regenerator/cylinder concept probably requires individual cylinder block castings anyway.

In order to use one secondary heat pipe per engine module, the use of a common cold mounting plate requires means to take care of manufacturing tolerances so as not to impose high stresses in the engine structure when bolting the crosshead liners to the crankcase. If one heat pipe per engine module becomes the number one choice, then it might be best to use one common cylinder block, in this way taking care of the gas pressure loads from the cold connecting ducts.

The overall approximate length of the secondary heat pipe structure and cylinder to cylinder distance is listed below for the various alternate designs. The number of bellows are added as an indication of the number of joints to be hermetically sealed. No bellows are needed when inert gas is introduced at the cylinder block cold face, (alternates 2, 3 and 6) but in that case cylinders and regenerator housings are seal welded or brazed to the mounting plate.

Alternative	Reg/Cyl	Engine Positions	Number of Heat Pipes per Engine Module	Cyl-Cyl dist.	Overall length Secondary Heat Pipe mm	Bellows per Engine Module
Alt. 1	12	Top	4	500	1100	4
Alt. 2	12	Top	4	500	1100	0
Alt. 3	12	Top	1	500	1100 square	0
Alt. 4	12	Side	1	500	or 1300 round	52
Alt. 5	4	Top	4	350	1350	20
Alt. 6	4	Top	1	350	1320	0
Alt. 7	4	Side	1	350	1320	20

Finally, a number of flat surfaces of the heat pipe structure must be checked for pressure loads. The pressure differential between ambient and heat pipe is low, one atmosphere, but the surfaces are very large. Possible shape limitations could influence the final choice of heat pipe.

2.3.3 Multi-Module Engine Concepts

Ricardo Consulting Engineers (England) was engaged by United Stirling to develop design concepts for the engine drive unit (crankcase assembly) and for multi-module arrangements suitable for use over the range of 500 to 3000 hp. The drive unit, or basic module, is based on the existing twin crank U-form drive as used in various United Stirling engines.

Due to the limited time and funds available, the investigation was used to emphasize possible problem areas which can be examined in more detail in a later phase of the project. The areas which need detailed attention are likely to be the coupling and de-clutching of the units in multi-module installations, and the balancing of each unit. Also problems associated with assembly and handling of large engines must be considered carefully in the design phase.

2.3.3.1 Introduction

Ricardo was engaged to undertake a preliminary concept of a 500 hp square-4 Stirling powered drive unit. These units must be suitable for use as a single module or coupled into various multi-module assemblies. Alternative fuel systems are being considered: these may be either liquid or gas, or the modules may be inverted and placed on a fluidized bed combustor. A further consideration in the preliminary design is the ease of module replacement for overhaul or maintenance in the multi-module installations.

This section summarizes the preliminary design schemes for the drive system module and indicates alternative configurations for the modules when coupled together.

2.3.3.2 Basic Module Concept

Due to the many possible variations and applications of the unit, and with strength and reliability a major requirement, a crankcase and bedplate assembly was chosen. With the square-4 form this concept will allow complete reversability of the crankshafts within the assembly which will be required in multi-module installations.

The crankshaft pins and journal sizes, (See Fig. 2-50) crankshaft, and connecting rod profiles are based on previous square-4, U-form design projects. The crankshaft will have bolt-on balance weights. As Fig. 2-51 shows the crankshafts are geared together, and the drive from a single module can be taken through a crankshaft. For a multi-module application an externally mounted drive shaft can be fitted, coupled to a crankshaft gear at any selected speed.

The basic crankcase assembly was designed to given credance to installation drawings, and no thought has been given to auxiliary drives such as oil pump, starter motor (if required) and the various Stirling engine system parts.

Although stress analysis work was not part of the initial phase of the project, sufficient knowledge has been gained by Ricardo to support the viability of the proposed design.

2.3.3.3 Module Balance

All of the U-form drive units recently designed for United

RICARDO

UNITED STIRLING

LARGE ENGINE - 500 HP MODULE

FIG NO: 2

DRG NO: 57712

DATE: 1-4-80

FIGURE 2-50: CRANKCASE
LONGITUDINAL SECTION

RICARDO

UNITED STIRLING
LARGE ENGINE - 500 HP MODULE

FIG Uo: 1
ORG Uo: ST711
DATE: 1-4-80

FIGURE 2-51: CRANKCASE CROSS SECTION

Stirling have a drive shaft mounted centrally in the crankcase above, and gear-driven by the two crankshafts. This causes the drive shaft to rotate in the opposite direction to the crankshafts and by adding balance weights to the front and rear of the drive shaft, the drive unit is in complete balance.

However, for a module of 500 hp it is impractical to incorporate the drive shaft inside the crankcase for the multi-module assemblies. It has been calculated that for a 6 module, in-line assembly producing 3,000 hp, and a generator speed of 1200 rpm, the common drive shaft must be a minimum diameter of 112 mm (Lloyds Register).

Figs. 2-52 and 2-53 show that for a module without this reverse rotation shaft, and with the necessary engine firing order required by U.S.S. and crankshaft balance weight positions, the resultant out of balance is a sideways oscillating motion of the unit. The magnitude of this movement has not been calculated, but is assumed to be of sufficient order to require remedial action, particularly when used as an inverted single module on a fluidized bed installation. Some methods of reducing or removing this unbalance are to fit two small balancing shafts inside the module (Fig. 2-54) or to mount the module in an opposed form (Fig. 2-55) and phase the units as a pair to neutralize the movement.

Ricardo therefore believes that some method of obtaining partial or full dynamic balance will be necessary for each unit or assembly, depending on the detailed overall configuration, fueling system, and engine installation.

2.3.3.4 Application of Modules

The unit has been designed for both single and multi-module

RICARDO

FIG No: 10
DRG No: 97673
DATE: 27-3-80

UNITED STIRLING - LARGE ENGINE 500 H.P.
OVERALL ENGINE BALANCE

SUMMARY OF BALANCE

PRIMARY FORCES - UNBALANCED

SECONDARY FORCES - BALANCED

PRIMARY MOMENTS - BALANCED

FIGURE 2-52: SUMMARY OF BALANCE

RICARDO

FIG No. 11

ORG No. 97674

DATE. 27-3-80

UNITED STIRLING - LARGE ENGINE 500 H.P.

OVERALL ENGINE BALANCE

PRIMARY FORCES - UNBALANCED IN THE FORM
OF A FLUCTUATING HORIZONTAL FORCE

FIGURE 2-53: OVERALL ENGINE BALANCE

RICARDO

FIG No: 12

DRG No: 57675

DATE: 27-3-80

UNITED STIRLING - LARGE ENGINE 500 H.P.
SOLUTION FOR 100% ENGINE BALANCE
2 BALANCER SHAFTS

C' BALANCE WT AT FRONT OF ENGINE
C' BALANCE WT AT REAR OF ENGINE

FIGURE 2-54: SOLUTION FOR COMPLETE ENGINE BALANCE

RICARDO

LIMITED STIRLING

LARGE ENGINE - 3000 HP ASSEMBLY

FIG NO: 6

ORG NO: 5776

DATE: 1-4-80

FIGURE 2-55: PLAN VIEW

SIX MODULES - OPPOSED - FLUIDISED BED

installations. The single module has been shown (Fig. 2-56) as a direct fired engine mounted with a 350 kW generator. An alternative assembly is an inverted module mounted on a fluidized bed combustor again driving a generator (Fig. 2-57). The multi-module arrangements are shown in Figs. 2-55, 2-58, 2-59, and 2-60 and compare the 6 module assembly of in-line and opposed units for liquid fuel and fluidized bed combustion.

The in-line modules are driven by a common external drive shaft, and by means of external bearing caps and overrun clutches will allow easy module shut-down for replacement or maintenance. The assembly of the drive shaft with gears, flexible coupling and overrun clutch, has not been investigated but is assumed to be a practical solution and torsionally stable.

The fluidized bed combustor has been shown in both 6 in-line form with a common drive shaft to a generator, and as 3 opposed pairs of modules coupled by two drive shafts to a common reduction gear box. The opposed pair of modules have the benefit of a smaller combustion bed area and possibly a balanced system.

An additional method of multi-module coupling is to drive the two units in an opposed form through a common shaft (Fig. 2-61). This will necessitate the module being turned through 180° to use the same engine firing sequence.

2.3.3.5 Conclusions

From knowledge gained during prior design and manufacture of various square U-4 form drive units, it is anticipated that this mechanical drive unit will pose no major problems with regard to stress levels. Areas that will require in-depth investigation

RICARDO

UNITED STIRLING
LARGE ENGINE - 500 HP MODULE

FIG NO. 3
DRG NO. 571/3
DATE: 1-4-80

FIGURE 2-56:
SINGLE MODULE WITH 350 KW GENERATOR

RICARDO

FIG No: 4

DRG No: 577114

DATE: 1-4-80

UNITED STIRLING
LARGE ENGINE - 500 HP MODULE

FIGURE 2-57:

SINGLE MODULE FLUIDISED BED

RICARDO

UNITED STIRLING

LARGE ENGINE - 3000 HP ASSEMBLY

FIGURE 5

DRG NO: 57715

DATE: 1-4-80

FIGURE 2-58: PLAN VIEW

6 MODULES - IN-LINE - FLUIDISED BED

RICARDO

UNITED STIRLING

LARGE ENGINE - 3000 HP ASSEMBLY

FIG NO: 7

DRG NO: 57117

DATE: 1-4-80

FIGURE 2-59: PLAN VIEW

SIX MODULES - IN-LINE - OIL FIRED

RICARDO

UNITED STIRLING
LARGE ENGINE - 3000 HP ASSEMBLY

FIG No: 8
DRG No: 57118
DATE: 1-4-80

FIGURE 2-60: PLAN VIEW

SIX MODULES - OPPOSED - OIL FIRED

RICARDO

FIG No. 9

ORG No: 57719

DATE: 1-4-80

LIMITED STIRLING

LARGE ENGINE- 500 HP MODULES

FIGURE 2-61: CROSS SECTION

TWO MODULES WITH COMMON DRIVESHAFT

include the coupling of multi-module assemblies, with a capability of de-clutching individual units, and the balancing of individual modules.

It must again be emphasized that due to limited time this report only covers an initial investigation into a large hp drive unit and its application. In the first stages of this investigation various factors proved that it was impractical to just "scale-up" the unit based on previous designs, and many detail areas (e.g. bearing lubrication system, production handling and assembly problems) will require investigation into the medium speed engine range (300 to 1000 hp) and development of its own design philosophy.

2.3.4 Engine Cost Estimate

The manufacturing costs of the large stationary Stirling engine were estimated by United Stirling with major reference to an existing engine design, the USS P-75 Stirling engine, for which there exists a substantial body of experience. Cost estimates were made for complete oil-fired engines in annual production quantities of 500, 1000 and 1500 units per year. The estimated factory costs for each 500 hp engine module are summarized below for these production quantities.

<u>Annual Production Rate</u>	<u>Factory Cost per 500 hp Module</u>		
	Sw Cr	U.S. Dollars [†]	\$/KW brake
500 units/yr	575,000	\$138,220	\$306/kW
1000 units/yr	440,000	105,770	\$234/kW
1500 units/yr	413,000	99,300	\$220/kW

* Based on 452kW brake shaftpower before auxiliaries (See Fig. 2-23 and 2-68)

† Based on the conversion rate of 4.16 SwCr = \$1.00.

Conversion from Swedish Crowns to U.S. dollars is based on an exchange rate of 4.16 Sw Cr per U.S. dollar. The final column shows the engine cost expressed as U.S. dollars per kilowatt gross engine output. These engine cost numbers are used in the overall system cost analysis described in Section 4.0. Although the engine cost estimates summarized above were developed for an oil-fired engine and include the liquid fuel combustion system and air preheater, they are also applied without modification for the coal-fired engine cost estimate in Section 4.0 where the FBC, and heat pipe costs are added to these engine costs. This can be justified by the fact that the oil-fired combustion system costs are a small fraction of the total engine costs as can be seen in Figure 2-63. Furthermore, the heater head cost used in the heat pipe cost estimate is thus the same as for the direct fired engine.

2.3.4.1 Details of Engine Production Cost Estimates

The following paragraphs describe the detailed assumptions and cost breakdowns that led to the engine cost estimates summarized above.

Assumptions

Cost estimates are based upon the following:

- 1 - The design of an engine that presently exists (P-75)
- 2 - Production facility layout shown in Fig. 2-62
- 3 - Direct labor costs, material costs, etc., as follows:
 - Material cost inclusive costs of components purchased
 - Direct labor costs (Swedish conditions)
 - Overhead on labor costs including costs for
 - Production Management and services
 - Reinvestment of Tools
 - Electricity
 - Rent of Facilities
 - Social Benefits
 - Interest and pay off of investments at 18%

Production areas

- A. Stores for received raw material.
- B. Stores for received purchased components.
- C. Heavy machining.
- D. Light machining.
- E. Stores for machined heavy components.
- F. Sub assembly.
- G. Stores for sub assembled parts.
- H. Final assembly.
- I. Testing.
- J. Packaging.
- K. Offices.

Production flow

- 1. Receiving raw materials.
- 2. Receiving of purchased components.
- 3. Machined heavy components to the stores.
- 4. Machined light components to the sub assembly area.
- 5. Purchased components to the sub assembly area.
- 6. Sub assembled parts to the store.
- 7. Sub assembled parts to the final assembly area.
- 8. Heavy machined components to the final assembly area.
- 9. Purchased components to the final assembly area.
- 10. Assembled engines to the test area.
- 11. Tested engines to the packaging area.
- 12. Delivery of engines.

FIGURE 2-62: FACILITY LAYOUT FOR MACHINING
AND ASSEMBLING 400 kW STIRLING ENGINES

This is to say that the estimated costs are comparable with the wholesale price of the engine and do not include selling costs.

- 4 - Costs for some Stirling components are to a certain degree based upon assumptions of future mass production technology

Uncertainties

Mass production technology for certain key components of the Stirling engine does not exist at this point in time. To some degree, however, there have been contacts with different machine suppliers regarding possibilities of manufacturing the Stirling components in mass production.

The material selection for components of the Stirling engine for future production is not yet completely defined. Therefore, this cost estimate is based on present material choices for the various engine components as described earlier.

Sensitivity Analysis

The summary of engine costs in Figure 2-63 shows an estimated manufacturing cost of 440,000 Sw Cr for a production quantity of 1,000 units/year.

Figure 2-64 shows how the costs will vary for production quantities of 500 and 1500 units/year.

FIGURE 2-63

SUMMARY OF ENGINE COSTS AT A PRODUCTION LEVEL OF 1,000 UNITS/YEAR

Component	Material & purch. parts Sw Cr	Machining time (hours)	Assembly time (hours)
-Drive unit	115,100	100	40
-Block and piston assembly	7,400	23	
-Regenerators & coolers	38,000	120	50
-Heater & preheater	33,000	220	165
-Air fuel system	4,500	15	15
-Power control	4,000	22	20
-Electronics	4,200		15
-Other auxiliaries and uncertain components	42,000	-	
-Final assembly			75
-Testing			80
	248,200	500	460

Summary

-Purchased materials and components	= 248,200 Sw Cr
-Machining costs 500 x 280 Sw Cr/hr	= 140,000 Sw Cr
-Assembly and testing costs	= <u>52,100 Sw Cr</u>

Summary	440,000 Sw Cr/unit
	or
	\$105,770/unit

FIGURE 2-64

SUMMARY OF ENGINE COSTS FOR VARIOUS ANNUAL PRODUCTION VOLUMES
(SENSITIVITY ANALYSIS)

Component	Material & purch. parts Sw Cr		Machining time (hours)		Assembly time (hours) (Sub assembly)	
	500/year	1500/year	500/year	1500/year	500/year	1500/year
-Drive unit	138,100	105,000	150	90	60	35
-Block and piston assembly	8,900	7,000	34	20		
-Regenerators and coolers	45,600	36,000	180	110	70	45
-Heater and preheater	38,500	31,000	320	200	240	150
-Air fuel system	5,400	4,300	20	10	18	12
-Power control	4,800	3,800	36	20	24	18
-Electronics	5,000	4,000			16	10
-Other auxiliaries and uncertain components	50,000	38,000	-			
-Final assembly					100	70
-Testing					120	60
	296,300	229,100	740	450	640	410
<u>Summary (Cost/unit at a production level of 500 unit/year;</u>					<u>1,500 units/year</u>	
-Purchased materials and components	= 296,300 Sw Cr		= 229,100 Sw Cr			
-Machining costs @ 280 Sw Cr/hr	= 207,200 Sw Cr		= 126,000 Sw Cr			
-Assembly and testing costs	= 70,400 Sw Cr		= 45,100 Sw Cr			
Summary	575,000 Sw Cr		400,200 Sw Cr			

2.3.4.2 Maintenance Costs

An analysis of scheduled maintenance and overhaul operations on the large Stirling engine was conducted so that these costs could be included in the overall system operating cost analysis (Section 4.0). This exercise was performed for a liquid fuel fired engine resulting in an O&M cost of 0.052 Sw Cr/KWH_{brake} or \$0.012/KWH_{brake}. When the maintenance costs associated with the combustion system are deleted, the corresponding O&M cost for the indirect fired engine (less FBC and heat pipe costs) are 0.037 Sw Cr/KWH_{brake} or \$0.0089/KWH_{brake}. Details of the analysis leading to these cost estimates follows:

Methodology

Estimated lifetime O&M costs for a 400 kW Stirling engine, operating under conditions as specified below, are based on essentially the same service intervals and maintenance operations as for a corresponding 75 kW-engine.

Spare part costs are related to a production volume of 1000 units per year. Manpower hour requirement is assumed to be 4 times that for a 75 kW module.

O&M costs/KWH for both directly and indirectly heated Stirling engines are based on 96,000 hours of operation at full power (452 kW_{brake}).

Application Data

Application: Prime mover for power generation set

Annual operating time: 8000 h

Max power generating capacity: 452 kW_{brake} at 900 rpm

Heat source: Coal fired fluidized bed with direct or indirect transfer of heat (heat pipe)

Engine Data

Output power: 452 kW_{brake} at 900 rpm

Load factor: 1.0

Working medium: Helium

MTBO, minor: 16,000 h

MTBO, major: 32,000 h

Engine life: 96,000 h

Maintenance Intervals, Component Life, Design Goal

General: Service intervals and life for the large 400 kW-engine are assumed to be the same as for the 75 kW-module. Only limited experience from maintenance and overhaul of 40-75 kW engines, running in laboratories and applications, exists so far. Projected component lives are based on current state-of-the-art. Maintenance intervals and component lifetimes as presented in the following are considered to be achievable in the near term period (e.g., before 1985). Specified design goals represent medium or long term development possibilities (1985-90). Discussions of O&M activities below refer to a direct heated Stirling engine.

External Combustion System

Preheater: The annular plate type recuperative preheater has a projected life of 8000 hr with a design goal of 16,000 hr. Regular cleaning from deposits (e.g., sulfur, ashes) is normally not considered necessary.

Combustor: Life of the combustor is primarily dependent on utilization of heat resistant materials. Present materials allow 8000

hours of operation before replacement. Design goals of 16,000 hours or more could, however probably be achieved with ceramic materials.

Heater: The heater head is designed for creep and thermal fatigue. Even tube temperature distribution and small variations of mean cycle pressure (constant engine load) will favorably affect heater life. Projections from current designs indicate more than 16,000 hours of operation before replacement. A design goal of 32,000 hours will require more heat/pressure resistant materials.

Acceptable limits for ash contents of the fuel to prevent deposits and degradation of heat transfer have not yet been established (direct-fired engine). However, removal of heater tube deposits should not normally be required.

Ignition System: The system is in operation only during the starter sequence. Replacement of the spark plug after 32,000 hours is foreseen.

Remaining components are designed for engine life.

Air/Fuel Control System: Combustor fan, fuel pump, atomizer compressor (if installed) will require rebuilding at 16,000 hour intervals. Cleaning of the fuel nozzle, turbulator, etc. are foreseen at 8,000 hour intervals. Remaining components, e.g., the K-jetronic system, the air control valve, are designed for engine life, requiring only minor adjustment and restoration at distant service intervals, e.g., 8,000-32,000 hr.

Closed Working Cycle System

Seals: Piston rod seals have the double function of preventing the

working medium from leaking into the crankcase and preventing lube oil from entering into the working cycle. Since degradation products from oil in the working cycle have a deteriorating effect both on cycle efficiency and life of the piston ring seals, only small amounts of oil are allowed to enter the cycle. This makes the piston rod seal a critical component to be replaced at 8,000 hour intervals from a preventive point of view. Recent progress in seal development indicates achievable design goals of 16,000 hours during the late eighties.

Similar intervals for replacement of piston ring seals (8,000 hours, design goal 16,000 hours) are foreseen.

Regenerator and Cooler: Both components are designed for engine life. Since regenerators and coolers operate in virtually clean helium gas, they will require a minimum of maintenance, i.e., cleaning at 32,000 hours.

Working Medium: A 10% reduction of engine torque due to working medium leakage is accepted, allowing 2,000 hours (design goal 4,000 hr) intervals between refills of helium.

Lube Oil System

A filter for separation of helium/lube oil mixtures inside the piston rod seal package, is designed for engine life with cleaning at 8,000 hour intervals. Corresponding intervals for exchange of lube oil and filter are sufficient, since the oil does not come into contact with hot combustion parts (no combustion products nor coking). Rebuild of lube oil pump is foreseen after 16,000 hours.

Power Control System

Piston and piston rod seals of the compressor are replaced at intervals corresponding to other helium seals (8,000 hr) and have a design goal of 16,000 hr. These intervals are also applicable to the check valves located in the engine. The electro-activated servo valve and the hydraulic pump are rebuilt at 16,000 hours. Design life is 96,000 hours.

Drive Mechanism

Inspection of cylinders, crank, bearings and connections for minor adjustments or - eventually - replacement, are foreseen at 16,000 hour intervals (minor overhaul).

A projected overhaul at 32,000 hours will include replacement of bearings and cylinder linings and restoration of pistons, crossheads, connecting rods, crankshaft, transmission gears, etc. All of them are designed for engine life.

Other Auxiliaries

Starter, generator, water pump are all designed for at least 96,000 hours life with a rebuild at 32,000 hours.

Scheduled Maintenance Activities - Summary

A survey of preventive component restorations and replacements at appropriate intervals is given in Figure 2-65. For an estimate of related man hour consumption, the O&M activities have been categorized into four groups (A-D) with increasing severity of operations involved. See Figure 2-66.

YEAR	0	1	2	3	4	5	6	7	8	9	10	11	12
HOURS OF OPERATION $\times 10^3$	0	8	16	24	32	40	48	56	64	72	80	88	96
A. Minor inspection	. A A A .	A A A .	A A A .	A A A .	A A A .	A A A .	A A A .	A A A .	A A A .	A A A .	A A A .	A A A .	A A A .
B. Major " "		B		B		B		B		B		B	
C. Minor overhaul (in-bed)			C				C				C		
D. Major overhaul (at workshop)					D				D				
External combustion system													
- Preheater, combustor		e	e	e	e	e	e	e	e	e	e	e	e
- Heater		c		c		c		c		c		c	
- Ignition system					e				e				
- Air/fuel control		c	c	c	r	c	c	c	r	c	c	c	c
- Combustor fan, fuel pump			r		r		r		r		r		
Closed working cycle system													
- Seals		e	e	e	e	e	e	e	e	e	e	e	e
- Heater			e		e		e		e		e		
- Regenerator, cooler					c				c				
Lube oil system													
- Oil filter			e		e		e		e		e		
- Oil trap			e		e		e		e		e		
- Oil pump			r		r		r		r		r		
Power control system													
- Seals, check valves		e	e	e	e	e	e	e	e	e	e	e	e
- Compressor, hydraulic pump			r		r		r		r		r		
Drive mechanism													
- Cylinder linings, bearings			r		e		r		e		r		
- Pistons			r		r		r		r		r		
- Crankshaft, connecting rods etc					r				r				
- Auxiliaries							r						

CODE: c = cleaning, adjustments r = restoration, rebuild e = exchange, replacement

Group A-D activities: see table II, page 7

FIGURE 2-65: MAINTENANCE INTERVALS

FIGURE 2-66: MANHOURL ESTIMATES FOR VARIOUS MAINTENANCE OPERATIONS

Group	Intervals, hours of operations	Operations	Man power requirement
A. Minor inspection	2000 hr	External inspection: -adjustments of auxiliaries, controls	8 hr
B. Major inspection	8000 hr	Major in-bed inspection: -inspection and adjustment of internal/external components -replacement of seals check valves, combustor, preheater -exchange of lube oil	40 hr
C. Minor overhaul	16000 hr	In-bed disassembly of engine: -inspection and replacement of components acc. to group B -replacement of heater -restoration of heater -restoration of cylinders, engine block, bearings -rebuild of auxiliaries	80 hr
D. Major overhaul	32000 hr	Major overhaul of the engine at workshop: -activities according to group B-C -complete restoration of the cold parts e.g. crank shaft, cylinder linings, transmission, auxiliaries	100 hr

Maintenance Costs

General: In order to allow for a comparison of both direct and indirect heated Stirling power generating plants, maintenance costs for oil/gas fired as well as heat pipe Stirling engines have been estimated in the following section.

The main difference between the two options is the external combustion system. Since only limited experience exists from heat pipe operations it is assumed that life of the heater remains the same for both systems.

Spare parts cost refer to a production volume of 1,000 units/year.

Manpower requirements for group A-D activities are based on results from investigations of 75 kW engines in identical applications, multiplied by a factor of 4.

Direct Heated Engines

Estimated O&M costs/KWH are based on the total engine life (96,000 hr).

Spare parts, material costs:

Sw Cr

- External combustion system		
(preheater (11),		
combustor (11)),	11 x 58,000	638,000

-	Closed working cycle system (heater (5), seals (11 sets)),	5 x 200,000	1,000,000
-	Power control system (compressor seals (11), check valves (11)),	11 x 35,000	385,000
-	Restoration of drive mechanism (4), auxiliaries (1)	4 x 25,000	100,000
-	Miscellaneous (lube oil, filter, helium, etc)		<u>15,000</u>

Total Material Cost: 2,138,000 Sw Cr

Working time cost:

Time consumption for O&M activities	Hours
36 minor inspections (group A) @ 8 hr	248
6 major inspections (group B) @ 40 hr	240
3 minor overhauls (group C) @ 80 hr	240
2 major overhauls (group D) @ 100 hr	<u>200</u>

Total time consumption 928 hr

Total working time cost: $928 \times 110 \text{ Sw Cr/hr} = 102,000 \text{ Sw Cr}$

Total O&M cost over the engine life-cycle: 2,240,000 Sw Cr

Total power generation over engine life cycle:

$$96,000 \times 452 = 43,390,000 \text{ KWH}_{\text{brake}}$$

O&M cost/KWH for a direct heated Stirling engine is 0.052

$$\text{Sw Cr/KWH} = 0.012 \text{ \$/KWH}_{\text{brake}}^*$$

* US\$ = 4.16 SW Cr

Indirect Heated (Heat Pipe) Engine

O&M costs are essentially the same as for a direct heated engine except for costs related to the external combustion system, which is excluded here.

	<u>Sw Cr</u>
Spare parts, material costs (2,138,000 - 638,000 Sw Cr):	1,500,000
Working time costs	102,000
	<hr/>
Total O&M Costs	1,602,000

$$\begin{aligned}\text{O\&M/KWH over the engine life cycle: } & \frac{1.602 \times 10^6}{43.39 \times 10^6} = 0.037 \text{ Sw-Cr/KWH}_{\text{brake}} \\ & = 0.0089 \text{ \$/KWH}_{\text{brake}}\end{aligned}$$

2.4 SYSTEM DESIGN AND ANALYSIS

The foregoing sections have described the selection and conceptual design of the major subsystems required for a complete coal-fired Stirling power system, e.g. the heat source or combustion system, the heat transport system and the Stirling engine. This section describes the conceptual design of a complete powerplant utilizing these major subsystems and presents estimated performance characteristics for this total system.

2.4.1 System Layout

Figure 2-67 is a pictorial of the Task 1A System Conceptual Design, showing the major elements required for an installation, physical arrangement, and their approximate size. The engine has three 500 hp modules driving a generator to produce electric power. Coal, limestone, and ash storage is sized for a 3 day supply (at 100% load factor). Not shown is the heat rejection unit, which may be a cooling tower located at the engine site, or connection to a nearby plant or building to provide useful hot water in a total energy installation.

The physical arrangement shown is selected for clarity rather than optimum space use. Considerations such as truck access, outdoor coal and limestone storage, etc. will influence the physical layout. The building is shown primarily to house the components which require maintenance, an office/control room, and sheltered space for the convenience of operating personnel.

Following the solid material flow, the coal and limestone (or dolomite) are transported from their respective silos by pneumatic lines to the small feed bunker. Screw conveyors transport the solids to the fluidized bed feed chutes, which gravity feed the bed with a mixture of coal and sorbent. On the other side of the bed, an ash extraction

1. Coal Storage	8. Forced Draft Fan
2. Sorbant Storage	9. Baghouse
3. Feed Hoppers	10. Ash Storage
4. Stirling Engine	11. Stack
5. FBC and Heat Transport	12. Induced Draft Fan
6. Reinjection Cyclone	13. Generator
7. Preheater	14. Office/Control Room

FIGURE 2-67: OVERALL PLANT SKETCH
1500 hp COAL FIRED
STIRLING ENGINE

line carries the ash to the ash storage unit, which in turn empties into trucks for disposal. Not shown is the means for filling the coal and sorbent silos.

The air flow path starts with the forced draft fan, cold air being heated by the preheater before entering the fluidized bed. The preheater is shown as a rotary type, with drive and ducting inside the enclosure. The combustion products exit the bed to a cyclone, which separates solid material for reinjection to the bed. The hot combustion products then flow through the preheater, to the baghouse for particulate collection, and finally to the induced draft fan for exhausting to the stack. Not shown is a small water-cooled heat exchanger prior to the baghouse, which may be required in order to protect the baghouse from high temperatures.

2.4.2 Heat, Mass, and Work Balance

The system flow diagram is given in Figure 2-68 for typical seasonal average conditions. For any specific installation the precise coal, sorbent, and ash flow rates will depend on:

- a) coal properties: heating value, ash content, sulfur content, moisture, etc.
- b) sorbent properties
- c) local requirements for SO₂ emissions
- d) fluidized bed sulfur capture properties: temperature, velocities, geometry, etc.

FIGURE 2-68: COAL FIRED STIRLING
FLOW DIAGRAM

- e) waste heat utilization, cooling water temperature, etc.

For the purposes of a conceptual design, typical FBC numbers were used to estimate flow requirements. The coal heating value was taken as 12,000 Btu/lb, and ash content 10%. The sorbent mass flow requirement based on typical coal sulfur content, sorbent properties, EPA emission requirements, and FBC characteristics is about one-half of coal mass flow rate. The ash flow rate is taken as 10% of the coal flow rate plus all of the sorbent flow rate (the substitution of sulfur for carbon in the sorbent does not appreciably change its mass). The flow figures are all nominal, suitable for conceptual design and rough cost estimating, but requiring more precise definition for a final design.

Shaft power from the engine is indicated as brake shaft power and net electric power based on 50°C engine cooling water outlet temperature and estimated auxiliary power requirements. The required auxiliaries are combustion air fans, cooling water pumps, cooling air fans, and coal sorbent and ash conveying motors. The auxiliary power requirement as a percentage of engine brake power is discussed in Section 2.4.4.

The heat to the cooling water is taken as the heat to the engine minus the shaft power output. Other heat losses from the engine were estimated to be below 2% and were thus ignored for the conceptual design.

2.4.3 Perspectives on the Use of Cooling Water

As can be seen in Section 4., System Cost Estimate, the coal-fired Stirling engine will not be cost competitive with utility power for electric power generation alone. What can make the system

economically attractive are possibilities for the use of waste heat. Unfortunately, the economics of waste heat utilization can become extremely complex due to the variety of economic and thermodynamic factors which must be taken into account when evaluating a specific application. Therefore, this report cannot define the best way to use the waste heat but can only examine the nature of the Stirling system in order to explain these thermodynamic and economic factors which must be taken into account.

Utilization Temperature

One of the most important factors when considering waste heat utilization is the temperature which is required. Typically, for most applications the higher the heat source temperature, the better. For all applications there will be a temperature below which heat is not usable. This cutoff temperature is a function of the application.

As already pointed out, the Stirling engine performance is a function of cooling water temperature (See Figure 2-26). As the cooling water temperature increases, the efficiency decreases and the power decreases. Furthermore, because of limitations in seal materials, the state-of-the-art Stirling engine is limited to a maximum cooling water outlet temperature of 70°C.

When the cost of power is calculated as a function of cooling water temperature, two factors must be considered. First, as the efficiency decreases, the cost of coal, sorbent and ash removal are increased per kilowatt of power generated. Second, as the maximum power decreases (for the same load factor) the capital charges (per kilowatt) and the maintenance costs of the engine (per kilowatt) increase. The second factor is roughly three times as significant as

the first. These factors were included in the model which was used to calculate the cost of power in Section 4.3.

Value of the Heat

In order to develop an economic model of waste heat utilization, some dollar value must be assigned to the waste heat. A common mistake is to value the heat at the cost of the fuel which would otherwise have to be burned to generate that heat. The cost of the fuel is the correct value to assign to the heat only if the equipment required to burn that fuel already exists and the repair and maintenance costs are small compared to the fuel cost. Otherwise, the capital charges and maintenance costs will be ignored.

For example, if the alternative to waste heat utilization is to burn coal in a separate FBC system, then the cost of heat produced from that boiler will be the sum of the fuel cost, other O & M costs, and capital charges. For a new installation, the cost of fuel will be roughly one third of the total. (20)

Furthermore, there are some significant engine direct costs or savings due to waste heat utilization. For example, the cost of electricity produced by the Stirling engine can be reduced when utilizing waste heat by reduction or elimination of cooling towers (thus reducing capital cost) and/or by reduction of the cooling power requirement (both the water pump and the cooling tower fans).

Other Possibilities

There are a few more effects which should be classified as first order effects, but which are only important in a portion of the potential market for the Stirling engine. The first deals with cases

where the utilization temperature is higher than is achievable with state-of-the-art engines.

Use of Heat Pumps: The current temperature limit for Stirling engine cooling water is 70°C due to state-of-the-art seal materials. Research and development in this area should provide materials with a higher temperature limit, but this is certainly in the category of advanced concepts. In the meanwhile, applications which require higher temperature heat could utilize heat pumps driven by the Stirling engine, which use the engine rejected heat at 70°C as the heat pump heat source. However, it should be noted that it can be shown by thermodynamic and economic arguments that it is usually not economical to drop the cooling water to a low temperature (maximum Stirling efficiency) and drive a heat pump which takes heat at the lower cooling water temperature and delivers heat at, say, 70°C. However, it might be economical to supplement the cooling water heat with a heat pump operating from an outside heat source if the heating demand is large. Essentially, the Stirling system becomes a heat activated heat pump in this case.

Use of Flue Gas Heat: There are two possibilities for the use of flue gas heat. The first possibility involves use of the flue gases beyond the preheater, which, under the assumptions of Figure 2-68 (System Flow Diagram), are at a temperature of roughly 500°F. Since this is too hot for the baghouse, the heat must be rejected, and can be classified as waste heat (assuming that detailed economic analysis shows that higher preheater effectiveness is not economical). Under these conditions, the flue gas heat could be used to increase the temperature of the cooling water a few degrees, or could be used in a separate heat transfer loop.

The other possibility for using flue gas heat is to reduce air preheat and generate steam. Strictly speaking, this concept no longer utilizes waste heat because the firing rate of the FBC must be increased proportionally. Nonetheless, this concept could prove to be economical in some applications, mainly because of reduced capital charges. In other words, if the bed is designed for less preheat, at a given temperature (and thus Stirling engine load), the firing rate of the bed is higher while the size of the heat exchanger remains the same. Thus, for essentially the same capital cost (except for the cost of the additional coal and sorbent storage which is required), the Stirling system can be made to produce more heat as needed. This would only likely be economical where a relatively high heat to electric demand ratio exists.

In conclusion, there are numerous factors which must be evaluated in a cogeneration application. Without the details of a specific application (particularly the heat and electric demand, the required utilization temperature, and the equipment already available), it is not possible to determine the best use of waste heat.

2.4.4 Auxiliary Power

There are four substantial auxiliary power requirements. They are the combustion air blowers, the cooling water pump, the cooling tower fans, and the coal and sorbent transport systems. The purpose of this section is to explain the assumptions and methodology used to estimate these power requirements.

Combustion Air

The combustion air would be supplied by a forced draft fan and an induced draft fan. The air must pass through the preheater, the

distributor plate, the bed, and then exhaust via cyclones, air preheater (and/or steam generators), baghouse, and finally leave through the stack. The volume flow of air may be estimated from the mass flows given in Figure 2-68 (Section 2.4.2, Heat and Mass Balance). The density may be estimated as that of air at the appropriate temperature. At full load the pressure drop across the fluidized bed (including distributor plate) is estimated to be 2.3 psi (16 kPa). The pressure drop through the entire flow path is estimated to be 4.5 psi (31 kPa). From this, the full load electrical power requirement is estimated to be 30 hp (22 kW) per engine module.

Cooling Water

The power required to pump the cooling water is primarily dictated by the method of heat rejection. The flow at full engine load is 26 kg/sec (417 gpm). This was determined in the scale up of the United Stirling engine module (Section 2.3.2). At this flow rate the pressure drop of the coolant through the engine is estimated to be 17 kPa (2.5 psi). The pressure drop through a Baltimore Aircoil Model 150-3 cooling tower at 26 kg/sec flow is approximately 60 ft head (26 psi, 180 kPa). Of course a larger, more expensive cooling tower could be used to reduce the pressure drop, however the cost-benefit trade off must then be re-examined. The full load power required to pump the water at 50% efficiency is estimated to be 12.5 hp (9.3 kW).

It is important to note that if the waste heat is utilized, then this power requirement might be decreased. Space heating, for example, may require a relatively low pressure drop. Furthermore,

its pumping power should no longer be chargeable to the engine.

Cooling Tower Fans

The cooling tower fan power requirement is difficult to estimate due to many site specific factors. The worst possible case for this power requirement would be if all of the engine rejected heat must be rejected by a dry closed-loop cooling tower. In this case, assuming an air-side temperature rise of 30°F (17°C), 1" wc air-side pressure drop (250 Pa), and 50% pumping efficiency, the power required is estimated to be 24 hp (18 kW).

However, in most installations the rejected engine heat would be utilized. If, for example, it were all utilized for space heating, then cooling tower fan power would be eliminated. Otherwise, only the fraction of heat not needed for heating would be rejected by the cooling tower. Furthermore, it is reasonable to expect that in most climates, wet or evaporative cooling would be utilized. This would have the effect of reducing the fan power requirement below that required for dry cooling. For these reasons the average full load cooling fan power requirement is assumed to be 5 hp (3.7 kW), or approximately 1% of engine shaft power.

Coal and Sorbent Transport Systems

The total power required for coal and sorbent and ash transport is estimated to be 0.5% of engine shaft power (2.5 hp at full load). This estimate is probably on the high (conservative) side. One commercially available unit (11) uses approximately 3 hp (2.3 kW) for coal and sorbent transport to fire a 15×10^6 Btu/hr boiler. This is roughly three times the firing rate of the FBC module for the Stirling

application and thus the Stirling application should consume one third the power, or about 1 hp (0.8 kW). Thus, the estimate of 2.5 hp (1.9 kW) should be conservative, but should allow for other miscellaneous loads not otherwise accounted for.

Summary

The total full load power requirement for auxiliaries is estimated to be 50 hp (37 kW). This is a sum of the following contributions:

- 1) Combustion air blowers: 30 hp (22 kW)
- 2) Cooling water pump: 12.5 hp (9.3 kW)
- 3) Cooling tower fans: 5 hp (3.7 kW)
- 4) Coal, ash & sorbent transport: 2.5 hp (1.9 kW)

Based on a brake engine shaft output of 452 kW and 95% generator efficiency, the total auxiliary power requirement is estimated at 8.6% of the gross electrical output. Alternatively, this can be expressed as 2.6% of the higher heating value of the coal. Since the bulk of the auxiliary power varies at least linearly with load, it should be conservative to assume that these percentages are independent of load.

2.4.5 System Performance

This section will complete the technical discussion of the proposed state-of-the-art concept with a discussion of plant efficiency, turndown and load following ability, and fuels flexibility.

Before proceeding, a few basic points must be considered. First, the Stirling-heat pipe-FBC system is fully modularized, in that for each engine module, there is a heat pipe system and an FBC module. Each FBC module is individually controlled to maintain a constant heater head temperature of 720°C for the associated engine module. The engine modules are geared to a common output shaft which

drives the generator. Thus, when considering system performance, it makes sense to consider the performance of a single Stirling-heat pipe-FBC module, and to scale the results for multi-module engines.

Plant Efficiency

All of the efficiency information presented in this section is derived from the Heat and Mass Balance (Figure 2-68) and the Influence of Cooling Water Exit Temperature on Power and Efficiency (Figure 2-26).

The coal pile to busbar efficiency (at $T_{\text{heater}} = 720^{\circ}\text{C}$, $\eta_{\text{generator}} = .95$, and all auxiliary losses included) versus cooling water exit temperature is given in Figure 2-69. At 80°C the plant electrical efficiency is 25%.

Because the plant performance can vary substantially in different applications, Figures 2-70, 2-71 and 2-72 present the state-of-the-art system performance under various circumstances. In Figure 2-70, the state-of-the-art cogeneration application is represented by cooling water exit temperature at 70°C (limited by current seals) heater temperature 720°C , and 95% generator efficiency. The state-of-the-art coal fired Stirling electric powerplant is represented in Figure 2-71 by reducing the cooling water outlet temperature to 30°C . These two cases are also presented in Figure 2-72 (first two columns). Furthermore, the nominal engine performance ($T_{\text{cooler}} = 50^{\circ}\text{C}$) is presented along with the state-of-the-art oil fired Stirling electric powerplant. For comparison, some advanced scenarios are presented which will be discussed further in Section 3.4.

* $T_{HH} = 720^{\circ}\text{C}$, $\eta_{gen} = .95$, auxiliary power losses included

FIGURE 2-69: COAL PILE TO BUSBAR EFFICIENCY
VERSUS COOLING WATER OUTLET TEMPERATURE

FIGURE 2-70: STATE-OF-THE-ART COGENERATION SYSTEM
HEAT BALANCE BY PERCENTAGE OF COAL HIGHER
HEAT VALUE

FIGURE 2-71: STATE-OF-THE-ART COAL FIRED
ELECTRIC POWERPLANT HEAT BALANCE
BY PERCENTAGE OF COAL HIGHER HEAT VALUE

FIGURE 2-72: STIRLING ENGINE SYSTEM PERFORMANCE
BREAKDOWN FOR VARIOUS APPLICATIONS

	<u>1A</u> <u>Cogeneration</u>	<u>1A</u> <u>Electric</u>	<u>1A</u> <u>Electric</u>	<u>1A</u> <u>Nominal</u>	<u>1B</u> <u>Cogeneration</u>	<u>1B</u> <u>Electric</u>	<u>XP-360</u>
1. Fuel	coal	coal	oil	coal	coal	coal	solar
2. Fuel Higher Heat Value	1.0	1.0	1.0	1.0	1.0	1.0	1.0
3. FEC/Heat Pipe Efficiency	.83	.83	.86	.83	.83	.83	1.0
4. Heat to Engine (2x3)	.83	.83	.86	.83	.83	.83	1.0
5. Heater Temperature (°C)	720	720	720	720	820	820	820
6. Cooler Exit Temperature (°C)	70	30	30	50	120	30	15
7. Engine Brake Efficiency	.359	.406	.406	.383	.330	.471	.589
8. Brake Shaft Power (4x7)	.298	.337	.349	.318	.274	.391	.589
9. Cooling Water Heat (4-8)	.532	.493	.511	.512	.556	.439	.411
10. Generator Efficiency	.95	.95	.95	.95	.95	.95	.95
11. Gross Electric Power (8x10)	.283	.320	.332	.302	.260	.371	.560
12. Auxiliary Power	.026	.026	.010	.026	.026	.026	2%
13. Net Electric Power (11-12)	.257	.294	.322	.276	.234	.345	.548
14. Cogeneration Efficiency (13+9)	.789	N/A	N/A	.788	.790	N/A	N/A

Note: XP-360 uses Hydrogen as the working fluid in the engine while all other cases assume Helium. Furthermore the XP-360 assumes a working fluid pressure of 11 MPa and engine speed of 600 RPM while all other cases assume 15 MPa and 900 RPM.

Turndown and Load Following

In order to assess the feasibility of the Stirling system in a particular application, it is important to determine whether or not the system has the turndown and load following capability needed for the application. The following analysis will examine some of the capabilities and limitations of the Stirling-FBC system.

The Stirling engine itself is capable of rapid load changes and a large turndown ratio. In fact, this characteristic of the engine is essential to the potentially largest market for Stirling engines - namely the automotive market. The capabilities of the FBC for rapid load changes and large turndown ratios are much more limited than those of the Stirling engine. Fluidized bed combustors have been reported to have characteristic response time constants for step changes in load in the range of 75 seconds to 720 seconds, depending on coal feed size (17). Turndown ratios of 2:1 are fairly commonplace, however improvements of up to 4:1 and higher are usually achieved by modularizing the bed or by defluidizing or slumping portions of the bed. The fluidized bed in the Stirling application will be no different. In fact, the Stirling FBC will most likely have a relatively small turndown ratio (2:1). The small turndown ratio is due to the fact that under full load operating conditions the bed heat release rate is limited by the heat flux to the heat pipes (See Section 2.1.5). Thus, the air flow cannot be reduced substantially before the bed no longer fluidizes (and mixes) properly.

It was noted that the FBC turndown ratio is limited by the need to reduce the airflow while maintaining fluidization. However, the bed can be operated at lower power levels, by operating with excessive combustion air or recirculated products of combustion. For example, if the air flow were maintained at the level required for good

fluidization and the coal input reduced below the level required to maintain stoichiometry of 20% excess air, then the bed temperature would drop and the heat flux to the heat pipes would drop below 50% of full power; however, the stack loss would be increased, resulting in lower heat input efficiency. Thus, in order to determine the lowest operating power level, a minimum heat input efficiency must be defined. This effect can be reduced through the use of exhaust gas recirculation (EGR).

The above discussion is valid when considering operation of a single FBC module at a fixed bed height. However, the Stirling FBC system would probably utilize bed height control in order to allow more rapid load following ability and would be modular in design, enabling greater turndown capability. For example, a 2000 hp engine would have four engine modules in line atop four FBC modules. A 50% power reduction could be achieved by reducing the fuel and air flow, resulting in a decreased bed temperature and heat flux. When operating in this mode (50-100% power) slow load changes could be followed simply by changing fuel and air flow. More rapid load following, if required, could be achieved by varying the bed depth and thus varying the active heat transfer area. While this bed depth control could provide more rapid response, it does inflict a small efficiency penalty due to extraction of bed material consisting of not completely spent sorbent and unburned carbon. (See Section 3.1.4 for an advanced concept which could improve this area.)

When turndown greater than 50% is required, it is possible to shut off one or more engine modules. Thus, if half of the modules were turned off and the other half ran at 50% load, total output would be only 25% of maximum. It should be noted, however, that the process of shutting down a module requires time. In order to protect the

structural integrity of the system and prevent possible corrosion problems in the bed, the air flow would be maintained for several minutes after the coal flow is stopped, in order to burn all of the coal in the bed and furthermore to reduce the bed temperature. (See Section 2.2.5 for further discussion.)

Fuels Flexibility

The state-of-the-art FBC module should be capable of burning oil, gas and all types of coal. However, the system should first be developed for coal. Once the operating and design characteristics are fully understood for one fuel, it will be possible to test the unit with alternate fuels such as wood, paper waste, peat, etc. For this reason, alternate fuels are discussed under the heading "Advanced Concepts", Section 3.1.2.

Conclusions

The overall plant performance has been characterized under conditions which reflect the most likely applications. Sufficient information has been presented to permit evaluation of system performance under off design and part load conditions as well. Furthermore, the limits of performance for state-of-the-art technology have been discussed. The next section presents some advanced concepts which could substantially improve system performance.

3. ADVANCED CONCEPTS

This section examines various advanced heat source, heat transport and engine concepts which could improve the cost, performance or reliability of the state-of-the-art system. As the emphasis is on concept generation and evaluation, the level of detail is somewhat less than in the previous section, State-of-the-Art Conceptual Designs.

3.1 HEAT SOURCE

3.1.1 Packaged FBC System

Given the chemical, thermal, and fluid mechanical constraints of the fluidized bed combustion process as presently understood, there is little which can be done to the process itself to reduce costs. However, in the mechanical implementation of the air, fuel, limestone, ash handling, and heat transfer equipment, there are opportunities for reducing costs.

The FBC system in Figure 2-3 is considered state-of-the-art. It is essentially a custom-engineered system, with the various components selected and designed for each particular installation. With sufficiently high demand, and reasonably homogeneous markets, the FBC could be designed as a standard integrated package.

One example of this approach is a commercially available boiler manufactured by the Johnston Boiler Company, as depicted in Figure 3-1. All of the essential elements of the FBC are contained in one integrated package - including mechanical dust collection. The package is designed so that it can be factory assembled and shipped

FIGURE 3-1: PACKAGED FBC CONCEPT
JOHNSTON BOILER COMPANY

either as one package or multiple modules which require a minimum of field work to install. On the basis of limited information, the "packaged" approach results in costs of the order of one half of engineered systems per unit firing rate. This is largely due to substantially reduced installation costs.

Referring to the system layout in Figure 2-67, a packaged Stirling FBC should contain all of the components shown inside the building. From the point of view of high production quantities, the packaged approach provides the means for substantial cost reductions by a manufacturer.

The remainder of the FBC support systems, including coal, limestone, and ash silos, and solids transport systems are less amenable to "packaging" - by virtue of their bulk. Referring to Figure 3-1, the size and configuration of those components will likely force them to remain non-packaged items. A purchaser of a coal fired Stirling will have site - specific requirements, and will want to make his own decision on how to store fuel at the engine site.

Suppliers of these types of material handling equipment do not see major cost reduction opportunities. The equipment is already mature in technology and production. With higher unit volume and standard sizes and configurations, there are opportunities for cost reduction through improvements in productivity, but cost reductions by factors of 2 or more are not likely.

In summary, advanced Stirling engine FBC systems would likely be "packaged" FBC units, which reduce both capital cost and installation cost. The rest of the necessary support equipment is not likely to change significantly in design but some reduction in cost should be possible by virtue of increased production quantities. Lastly, the

improvements described here are likely to result from the need for lower-cost industrial coal combustion equipment in general, i.e., steam boilers, not just for the Stirling engine.

3.1.2 Special Applications

There are many possibilities for application of the coal fired Stirling engine. Remote power generation and commercial/residential total energy systems are two such possibilities. Application of Stirling engines for total energy systems is covered in depth in reference 22. By comparison, it is beyond the scope of this study to cover the use of Stirling engines in various applications in much depth. This section is restricted to mention a few other possibilities, with limited analysis of what might be done in such cases. In particular, this section is concerned with changes that might be made in the heat source.

Emissions Changes

In Section 2.1.2 of this report, it was assumed that the use of "best available technology" with regard to emissions is necessary. That is probably true in the majority of applications. However, there may be cases, such as some remote power applications, where less stringent emissions regulations are in effect. The degree of control as well as what is controlled is very important. For example if less stringent sulphur oxides emissions standards are in effect, then the need for limestone or other sorbent may be eliminated. In such a case, any inert bed material could be used. Ash removal could involve screening the extracted bed material and re-using material of the correct particle size. This would eliminate the cost of limestone receiving, storage and handling and would reduce the cost of ash removal significantly, because the mass of material to be removed would be decreased.

Another possible cost-reducing emissions scenario might involve elimination of the baghouse for particulate clean up. Cyclone dust collectors would still be used to capture most particulates in order to maintain high carbon utilization. The elimination of the baghouse would reduce both capital and operating costs. Again, this depends entirely on the emissions criteria which must be met.

Temperature Effects

There might be applications where it would be possible to increase the temperature of the fluidized bed. For example, if sulphur capture efficiency is not important (because of low sulphur fuel, relaxed emissions standards or large supply of cheap limestone) then the bed temperature would most likely be limited by other considerations. The air preheater or distributor plate might be limiting for material reasons, or for example, the ash fusion temperature might be limiting. (Formation of ash agglomerates is undesirable.) In any case, a higher bed temperature might be possible in some circumstances. This could be used in many different money-saving ways.

One possibility would be to raise the bed temperature to compensate for failure of a portion of the primary heat pipes. As the primary heat pipes reach the end of their useful life, the bed temperature could be raised to increase the heat flux to the remaining heat pipes. In this way, the engine could be run longer at a higher load factor on a given set of primary heat pipes than could an engine which has a lower limit on bed temperature. This would reduce operating costs by decreasing downtime and labor requirements. Furthermore, the cost of the replacement heat pipes themselves would be reduced because of the longer average useful heat pipe life.

Another possibility would be to increase the temperature of the

Stirling heater head and thus increase engine efficiency. Increased efficiency reduces fuel costs, but more important is the increased power output for the same equipment. Thus, the \$/kW capital cost is reduced. This of course depends mostly on the ability of the engine to withstand a higher temperature. (See Section 3.3) If the engine could withstand higher heater head temperatures, then increased bed temperature could help provide the opportunity.

Alternative Fuels

One of the advantages of the FBC is the ability to burn a wide range of fuels in the same combustor. It is conceivable that once developed, the FBC - heat pipe - Stirling system will be capable of burning paper waste, wood chips, heavy oils, high sulphur oils, peat, or even tar sands. Each of the candidate fuels would have to be tested and the system developed for its use. Some fuels may require additions or modifications to the basic equipment.

For example, one problem with burning the tar sands directly is the high heat loss due to the heating of the inert solids. One solution to this problem if there is a simultaneous need for steam might be to utilize the heat in the rejected solids to generate steam. However, this means special equipment, custom designed for the application. Alternatively, some of the hot solids might be used to preheat the incoming tar sands. Again, this means custom designed equipment.

The economics of alternate fuels utilization probably would vary greatly. If there is a readily available cheap fuel and a need for power generation, the FBC Stirling system could provide an economical way to use the fuel. Furthermore, the economic outlook of such systems is strongly dependent on the price of conventional fuels. As

the price of oil and natural gas continues to rise, the economics of alternate fuels utilization will continue to improve.

Air Preheat

There may be applications where a significant steam demand exists. As was briefly pointed out in Section 2.1 and Section 2.4.3, it is possible to use the hot flue gases to generate that steam. The amount of air preheat could be varied according to the steam demand. If necessary, a steam superheater could be used in front of the air preheater. The important message here is that the system could be built to be flexible, and produce only the amount of steam that is needed. This feature could be useful in an application which has a constant electric load and a varying steam demand.

The economics of using flue gases to generate steam are likely to be favorable. This is because the bed firing rate must be increased as the flue gases are not used for preheat. Thus, more useful output results for the same basic equipment, and thus capital costs can be amortized at a lower per kilowatt price.

3.1.3 Low BTU Gas

The primary impetus for using low BTU gas in the Stirling application is elimination of the indirect heat transport system. The combustible gas would simply be piped from the gasifier to the combustor/heater head of the engine where it would be burned. The combustion gases would pass directly over the heater tubes.

To reiterate, the primary drawbacks in the conventional low Btu gas approach are as follows: Sulfur presents a major problem. It is too costly from a capital cost point of view and from an efficiency point of

view to remove the sulfur from the low Btu gas stream prior to combustion. This is because the sulfur is in the form of H_2S and by conventional techniques may only be removed at a low temperature. Thus, the gas must first be cooled. Even if this is done regeneratively (which means significant capital investment), system efficiency will be lowered. Instead, SO_x scrubbers would be used in the flue gas stream. These too are expensive and create a difficult disposal problem. Secondly, tube fouling of the heater head could be a problem due to particulates in the low Btu gas stream. This ash is heated upon gas combustion and deposits as slag on the heater head tubes. Thirdly, about 10% of the heating value of the coal is lost in converting from coal to gas. Combustion losses, stack losses, etc. further lower the efficiency. Fourthly, fuels flexibility is usually limited to a certain range of coal quality. Lastly, commercially available units have limited turndown capability and are currently available only in sizes too large for the Stirling application.

Gasification of coal in an atmospheric fluidized bed could possibly improve the outlook for low Btu gas in the Stirling application, particularly by improving the sulfur problem. The biggest question here would be with regard to the nature of the effluents and with the sulfur capture efficiency. In a fluidized bed combustor, the sulfur reaction proceeds as follows:

The product, calcium sulphate, ($CaSO_4$) is a dry stable solid with potential uses. (i.e., road bed material). In a fluidized bed gasifier, the sulfur is present as H_2S and the atmosphere is reducing, i.e., no free oxygen is available. The reaction is then likely to proceed along the following lines:

The product, CaS, would have to be discarded.

In any case, before the fluidized bed gasifier could run a Stirling engine, the problem of tube slagging would still have to be resolved. Three possible ways to deal with this problem are:

- 1) Clean all ash from the product stream before entering the combustor.
- 2) Allow for some tube fouling and provide means of cleaning the system.
- 3) A combination of the first two - likely to be the best solution.

The problem with cleaning the ash from the product stream is that it is very difficult to get all of the ash out. Small amounts will inevitably get by and foul the tubes over time. This problem is compounded by the fact that the ash removal must take place in the hot low Btu gas stream. Cooling of the gas poses an efficiency penalty. The proposed solutions of high temperature filtration in the coal fired gas turbine work currently underway (28) (utilizing combustion or gasification in pressurized fluidized beds) could apply in this case. However, the pressure drop across the filtering device would be more of a problem with the atmospheric pressure gas than with pressurized gas because of the higher velocity for equal mass flow.

The problem with allowing the heater head tubes to become fouled and then scheduling periodic cleaning is basically due to the insulating properties of the ash deposits. By insulating the tubes, lower combustion efficiency will result. Furthermore, temperature non-

uniformities could develop which would adversely affect engine life and performance.

Conclusion

If a workable direct fired system could be developed, the major gain would be the elimination of the sodium heat pipes. The drawbacks would be the cost of a hot filtering system and the cost of the combustion system with provisions for tube cleaning. Furthermore, disposal of the spent sulfur sorbant material could be troublesome.

3.1.4 Other Concepts

This section deals with ideas which were generated for improving the concept chosen for 1985 demonstration. This is not an exhaustive listing but rather the ideas are chosen as being potentially fruitful in the next ten years.

Extended Surface

One area of critical importance in the design of the fluidized bed is the heat flux to the in-bed heat pipes. It is important that the heat of combustion can be transferred to the tubes without requiring too close tube packing. Furthermore, the heat flux per unit tube area is limited by the combustion side heat transfer. In the present design, the bed temperature is pushed to the maximum from a sulfur capture point of view, in order to achieve the maximum possible heat flux per unit area to the tubes.

If extended surface could be used on the tubes to increase the effective tube area, then it might be possible to lower the temperature of the bed, thereby increasing sulfur capture efficiency

and improving sorbant utilization. Alternatively, fewer primary heat pipes could be used, thereby reducing cost.

Experiments should be performed to determine the effects of close tube spacing. Various extended surface configurations should be tried and their effects, if any, on allowable tube spacing should be determined. In this way, a more optimal array of heat pipes could be constructed.

Some extended surfaces to consider are: widely spaced circumferential grooves (similar to threads), longitudinal grooves or splines, longitudinal fins, and possibly pin-fins.

Alternate Elutriation Control

The question of freeboard height was discussed in Section 2.1.5. It was shown that for cost reasons, it would be desirable to decrease freeboard height. However, too short a freeboard can cause excessive particulate loading in the cyclone. This can cause inefficient carbon utilization. Thus, alternative means of elutriation control which would allow decreased freeboard height without causing excessive particulate loading in the cyclone are desirable. One possibility might be to increase the cross-sectional area of the freeboard in order to decrease the gas velocity. However, this would most likely only give limited improvement. Another possibility might be to place a screen or perforated plate in the freeboard to assure uniform gas velocity across the cross section and to block large particles which may have been sprayed vertically by a bubble reaching the surface. The lifetime of such a plate in the FBC environment might be a problem, and the array of vertical heat pipes which protrude through the plate might be a problem.

Alternate Feed Methods

Basically there are two ways to feed in the coal and sorbant. One method is to feed the coal down a chute onto the top of the bed. This is called overbed feeding. The advantages of overbed feeding are simplicity (and therefore cost), and greater coal dispersion per feed port. However, when a significant portion of the coal consists of fines, or the coal has a high volatile content, this method of feeding can be unsatisfactory, basically because fines and volatiles will be burned in the freeboard without ever passing through the fluidized bed. This can lead to emissions problems with both sulfur oxides and carbon monoxide, and can lead to efficiency problems due to incomplete combustion.

The other method of coal feeding, underbed feeding, involves injection of coal near the distributor plate, typically from a nozzle. The problem here is basically due to limited lateral dispersion of the coal and thus non-uniform stoichiometry in the bed. To overcome this problem, several feed ports are used in larger beds. The advantage is that all coal gets injected into the bed and thus comes into contact with limestone. Thus, with this type of feeding the FBC is less sensitive to the type of coal being burned. The principal disadvantage of underbed feeding is cost. The high preheat of the Stirling FBC further complicates underbed feeding. What is needed here is a low cost, reliable solution. It may be a cheaper underbed feed device, or it may be a method to assure rapid mixing of coal from overbed feed into the fluidized bed.

Alternate Ash Removal Methods

The problems with removal of spent bed material were briefly discussed in Section 2.1.6. The simplest and cheapest way to remove

bed material is by a fixed weir at the top of the bed. Alternatively, bed material can be removed from the bottom of the bed, but then an air lock is required because of the positive pressure at the bottom of the bed. Furthermore, with underbed ash removal, some sort of active control is not only possible but mandatory in order to maintain a desired bed height. Thus, the cost of underbed ash removal is higher compared to the fixed weir. An innovative solution needs to be developed.

The difficulty with both the coal feed problem and the ash removal problem is that it is difficult to devise a workable solution without intimate practical experience with a test bed. As more and more people gain experience with fluidized beds it is reasonable to expect that more and more innovative solutions to these practical problems will be devised.

Load Following Improvement

The possibility of utilizing bed depth control to allow more rapid load following was briefly discussed in Section 2.4.5, System Performance. It was mentioned that rapid changes in demand could be followed more quickly by changing the bed depth than could be achieved with bed temperature control at a fixed bed depth. For example, should the demand suddenly drop by 10%, then the bed depth could quickly be dropped by 10%. However, by disposing 10% of the active bed, substantial amounts of unburnt coal and unspent sorbent would be discarded.

However, if a separate control tank were utilized, then the extracted bed material could be saved. Then, rather than feeding in fresh coal and sorbent just after extraction of a portion of the bed, the material which had been removed would be fed back gradually into

the FBC. In this way, the response time of the bed could be improved with little to no efficiency penalty.

3.1.5 Conclusions

It is difficult to rank all of the concepts discussed in this section, since they are not all directly comparable. However, some overall priorities can be recommended. The first concept presented, Packaged FBC System, is expected to be critical to the success of the FBC - Stirling system. Large reductions in equipment costs and installation costs could be realized in volume production of a packaged system. Furthermore, it is expected that the packaged Stirling - FBC could help open up the smaller industrial boiler market for coal fired FBC steam boilers as well.

The advanced concepts discussed in Section 3.1.2, Special Applications, are concepts which could improve the economics of the Stirling FBC system in special applications. These concepts should be pursued as the need arises, i.e. if an application for the Stirling - FBC system is identified which permits utilization of one of these concepts.

Low Btu gasification of coal in an atmospheric fluidized bed (Section 3.1.3) should not be pursued for the Stirling engine alone. This technology is currently under development for other applications, and the Stirling could benefit from successful development of a gasifier for the other markets. We believe that the economic benefit of gasification is expected to be small for the Stirling application. At best, the heat transport system could be eliminated, but probably at the cost of greater complexity and/or higher operating costs.

Finally, the concepts discussed in Section 3.1.4 are not just

applicable to the Stirling, but apply to fluidized bed combustion in general. The impetus for their development is likely to come from broader needs than just this one.

3.2 HEAT TRANSPORT

The work on advanced heat transport concepts is divided into two sections. The first section, 3.2.1, Advanced Heat Pipes, presents ideas which were generated to improve upon the state-of-the-art heat pipe conceptual design. The second section, 3.2.2, Direct Fired by FBC, presents a totally different concept, essentially eliminating the heat transport system altogether.

3.2.1 Advanced Heat Pipes

Powder Metallurgy Wicks

The gravity-assisted two-stage heat pipe was determined to be the best heat transport option in the state-of-the-art conceptual design phase. However, until further design studies examine all of the details of this concept it is not possible to determine finally whether or not there is a better way. In light of this, it may be worthwhile to keep in mind the possibility of operating heat pipes over horizontal distances and even possibly against gravity. While it is likely that the chosen concept (gravity-assisted) is the most cost-effective, brief examination of non-gravity assist heat pipes is worthwhile as an alternate approach.

A heat pipe that operates against gravity has the evaporator above the condenser and the condensed working fluid (liquid sodium) is returned from the condenser to the evaporator by capillary action within a wick. The larger the liquid flow (the greater the heat flux) and the higher the gravity head, the more sophisticated a capillary wick is needed.

Sodium heat pipes similar in appearance to the primary heat pipe design have been operated against gravity. However, because of the sophistication required in the wick structure, their costs are relatively high and their reliability lower. This is not to say that improvements cannot be made, but development work needs to be carried out.

At the present time, Thermacore Inc. is involved in the development of high performance heat pipes capable of 10 kW/cm^2 of cross-sectional area. This high performance is achieved by the use of powder metallurgy wicks. These wicks have extraordinarily small capillary radii, which in turn produce the high capillary pressures necessary to transmit large flow rates against gravity. To date, powder metallurgy wicks have been made with copper, nickel, iron, and molybdenum. Figure 3-2 is a photograph of a lithium-molybdenum heat pipe at 1800°C . This 18" long heat pipe transfers 6 kW against gravity. Similar sodium heat pipes have been built and tested, and have the capability of being a primary heat pipe operating against gravity.

In order for powder metalurgy wick heat pipes to be considered for use in the Stirling application, their cost must be economically justified by their benefit; and secondly, fabrication techniques and equipment capable of manufacturing the required length of heat pipe must be developed.

Hydrogen Diffusion

No matter what type of heat pipe is used (gravity assisted or not) the following concept is proposed in order to extend the life of primary heat pipes when a two-stage design is employed.

FIGURE 3-2: MOLY-LITHIUM HEAT PIPE

Regardless of the material used for primary heat pipes, some finite hydrogen diffusion into the heat pipe will take place. If the hydrogen diffuses sufficiently rapidly into the heat pipe, then the heat pipe will be "poisoned" and will not function (The hydrogen collects at the condenser and forms an insulating layer). A possible solution to this problem would be to use material with very low resistance to hydrogen permeation at the condenser of the primary heat pipe and at the evaporator of the secondary heat pipe so that the hydrogen can diffuse from the primary heat pipes into the secondary. Once in the secondary heat pipe, the hydrogen would migrate to the same place that the inert gas (which was purposely introduced) resides. Because the volume of the secondary heat pipe is large, the amount of hydrogen which would enter it would not affect its performance. In this way, failure of heat pipes by hydrogen diffusion could be reduced or eliminated. Further design and experimentation is required to explore this idea.

3.2.2 Direct Fired By Fluidized Bed

The direct fired concept was considered for both the fluidized bed combustion and the low Btu gas systems. Because the fluidized bed was selected as the best combustion system, the focus of the direct fired work was on FBC. To date, no workable layout design exists, however some ideas have been generated which could lead to a viable conceptual design.

Technical Approach

The scaled-up (P-75 type) oil fired combustor was used as a baseline of comparison for potential direct fired by FBC design heater head designs. Unsuccessful attempts were made to develop

conceptual designs for a heater head compatible with the FBC for the scaled-up United Stirling U-4 engine. Alternate engine configuration schemes suitable for direct firing in an FBC were developed. Time and manpower limitations did not allow the technical feasibility of these alternate engine configurations to be studied in depth.

The following is a preliminary design analysis for a heater head direct fired by FBC, along with a description of some ideas regarding possible bed/heater/engine layouts. Figure 3-3 gives a list of the assumptions and nomenclature used in the analysis.

Scaling Results

The following geometry was defined as being suitable for direct oil-firing of the scaled up U-4 engine design. See Section 2.3.1.3.

$$\begin{aligned} N &= 24 \\ L &= 33.5 \text{ in} \quad (850 \text{ mm}) \\ D_o &= .571 \text{ in} \quad (14.5 \text{ mm}) \\ D_i &= .394 \text{ in} \quad (10 \text{ mm}) \\ V &= 97.8 \text{ in}^3 \quad (1602 \text{ cm}^3) \end{aligned}$$

Heater Head Comparative Analysis

$$V = \frac{D_i^2 L N}{4} \quad (5)$$

$$A_o = F D_o L N \quad (6)$$

$$P = \text{Internal Pressure Drop} = 4f (L/D_i) \frac{G^2}{2\rho} \quad (7)$$

FIGURE 3-3: ASSUMPTIONS AND NOMENCLATURE

Temperature of bed = $T_{BED} = 1650^{\circ}F = 890^{\circ}C$

Temperature of heater head tubes = $T_{HH} = 1330^{\circ}F = 720^{\circ}C$

Bare tube exterior film coefficient = $h_o = 67 \frac{\text{Btu}}{\text{hr-ft}^2\text{-}^{\circ}F} = 382 \frac{\text{W}}{\text{m}^2\text{-}^{\circ}C} *$

Length of tubes = L

Inside diameter of tubes = D_i

Outside diameter of tubes = D_o

Number of tubes per engine cylinder = N

External heat transfer area per cylinder = A_o

Heater head dead volume = V

Heat flux enhancement factor = $F \leq 6.0^{**}$

Heat input to 4 cylinder module = $4.0 \times 10^6 \text{ Btu/hr} = 1.2 \text{ MW}$

* This is smaller than for the heat pipe conceptual design. While the heat flux per unit is greater for direct fired than for the heat pipe, the heat transfer coefficient is smaller due to a decreased equivalent radiation heat transfer coefficient. See Section 2.1.5.

** The effective heat transfer area may be increased by the addition of fins. However, as the fin area increases, the heat transfer film coefficient decreases. Thus, there is a limit to the amount of heat flux improvement that can be achieved by the addition of fins. (40)(41)(29)

where f is the friction factor, ρ is the fluid density and G is the mass flow rate per unit flow area. From the above, it can be shown that:

$$P = \frac{L}{N^2 D_i^5} \quad (8)$$

Therefore, if the value of $L/N^2 D_i^5$ is held constant for all designs, the internal flow and pressure drop characteristics will be approximately the same for all designs.

Potential Direct Fired Heater Head Designs

Figure 3-4 summarizes calculations for several potential heater head designs for direct firing in a fluidized bed combustor. The principal design variables are tube size, number of tubes and effective area enhancement factor, F , due to external fins. Combinations of D_i , L and N have been selected so that the internal P is nearly the same for all cases. Actual tube arrangement is not important in this conceptual analysis. The problems of tube arrangement will be discussed later.

Comments on Analysis

Several observations may be made from the results shown in Figure 3-4. First, (see design case 4) by using bare tubes approximately equal in diameter to those of the baseline case (scaled-up design), both the number of tubes and the tube length must be increased substantially, thereby resulting in a much greater void volume. This is clearly undesirable.

However, the desired results (e.g. $V/V_{\text{Baseline}} = 1$ and

FIGURE 3-4: POTENTIAL DIRECT FIRED HEATER HEADS
FOR A FLUIDIZED BED COMBUSTOR

<u>DESIGN VARIABLE</u>		<u>OIL-FIRED BASELINE CASE</u>	<u>DIRECT FIRED FBC</u>			
			<u>1</u>	<u>2</u>	<u>3</u>	<u>4</u>
F		N/A	2.5	4.0	2.5	1.0
D _o	in (cm)	0.57 (1.45)	0.75 (1.91)	0.75 (1.91)	0.625 (1.59)	0.625 (1.59)
D _i	in (cm)	0.394 (1.00)	0.394 (1.00)	0.394 (1.00)	0.31 (0.79)	0.31 (0.79)
L	in (cm)	33.5 (85)	40.7 (103)	29.7 (75.4)	34.2 (86.9)	65.8 (167)
N		24	28	24	40	52
Effective A _o	ft ² (cm ²)	N/A	46.6 (43300)	46.6 (43300)	46.6 (43300)	46.6 (43300)
V/V _{Baseline Case}		1.0	1.42	0.89	1.05	2.64
ΔP/ΔP _{Baseline Case}		1.0	0.89	0.89	1.22	1.39

$\Delta P/\Delta P_{\text{baseline}} = 1$) can be achieved by using extended surface (fintubes) in the fluidized bed as illustrated by cases 1, 2 and 3.

Several design trade-offs may be identified. First, there is the trade-off between void volume and design risk, where void volume decreases with increasing F, but with increasing F there is increasing difficulty in designing the finned heater head tubes with confidence. Further work to identify the performance and limitations of finned tubes in fluidized beds is necessary to assess the feasibility of these concepts.

Second, there is a trade-off between higher numbers of smaller diameter tubes and higher void volumes. The assembly of higher numbers of smaller tubes gets more complicated the more tubes there are. Reliability also probably decreases. However, the void volume is important from a Stirling cycle efficiency point of view, so a compromise is required. Both of the cases shown are considered reasonable.

Although not shown in the figure, there is a trade-off between numbers of tubes and internal flow pressure drops. It was decided not to examine the nature of this trade-off, but rather to attempt to keep the pressure drop of the proposed designs as close as possible to the pressure drop of the "baseline case" (scaled-up) design.

Lastly, there is a trade-off between void volume and the tube length. Depending on as-yet-unidentified combustor dimensional requirements, longer tubes may be desirable in spite of higher void volume in order to fit the heater head into the FBC completely and uniformly.

The conclusion to be drawn from this analysis is that, at least

theoretically, it is possible to design a direct FBC fired heater head which has a pressure drop and internal volume comparable to conventional (oil fired) Stirling heater heads. This can be seen from Figure 3-4. Case 1, for example, has roughly equal pressure drop characteristics (0.89 times baseline case) and has an internal volume 1.42 times the internal volume of the conventional "baseline case". Case 1, however, is only feasible if it is possible to utilize fins to enhance the effective heat transfer area by a factor of 2.5. This has been reported in the literature (29). Another problem critical to determining the feasibility of the concept is one of tube arrangement.

Tube Layout

Upon studying potential heater tube arrangements for the U-4 engine, it became obvious that the problem of arranging many tubes of the same length with a reasonable heater head dead volume is difficult if the tubes are geometrically arranged to fill the bed. Conversely, if the tube banks are arranged to satisfy engine packaging criteria, then there will be flow channeling problems in the bed. For example, the present state-of-the-art conventional Stirling circular heater head design is not suitable for the FBC because of the potential for flow channeling problems.

Figure 3-5 shows a design scheme which is probably geometrically compatible with the bed; however, it requires inordinate amounts of header volume. Large header volume sacrifices both engine power and efficiency and is clearly undesirable.

Alternate Engine Configurations

Attempts have been made to generate engine module designs which

FIGURE 3-5: DIRECT FIRED U-4 ENGINE

are more amenable to direct firing in an FBC than the present U-4 arrangement. Figures 3-6 and 3-7 illustrate two of the resulting ideas. Yet another idea is presented in Section 3.3.2. These arrangements require some basic changes to the engine crankcase and cylinder layout. There are potential design problems with cold side void volumes between cylinders and coolers, etc. These potential problem areas have not yet been assessed adequately.

Conculusions

The direct fired fluidized bed could turn out to be the best technical approach for the future coal fired Stirling engine. It appears to have the lowest cost potential and minimum safety hazard potential of any alternative because it offers simplicity, and it eliminates the need for heat pipes. Furthermore the direct fired approach makes for less weight and bulk in a packaged system. These considerations are important in the packaged FBC system (See Section 3.1.1). However, today it carries a high technical risk because it requires an engine design which is not state-of-the-art and which has not been studied in sufficient detail to assess its ultimate feasibility. (See Section 3.3.2 for another direct-fired engine concept.)

FIGURE 3-6: DIRECT FIRED WITH
TWO REGENERATORS
PER CYLINDER

FIGURE 3-7: DIRECT FIRED WITH
FOUR REGENERATORS
PER CYLINDER

3.3 ADVANCED ENGINE CONCEPTS AND COMPONENTS

3.3.1 Introduction

Advanced concepts can be treated both at engine system and at component level. Advanced concepts at the engine system level will likely deviate significantly from the simple, upscaled 4-275 (P-75) concept. This will be the result of the combined effect of the much larger size and the introduction of new design ideas. A detailed presentation of such concepts will therefore require a significant amount of analysis and is outside the scope of work for Phase I. The discussion in Section 3.3.2 below serves the purpose of identifying ideas which merit further study but is not detailed enough to permit ranking of concepts.

At the component level there are many ideas for advanced concepts. Examples will be given in Section 3.3.3. Some of these ideas do not drastically change the basic engine system concept and can therefore be incorporated into an existing design at a later stage, once they have been properly evaluated in separate component testing. Others require more major configuration changes. An example of an idea generated at component level but requiring a different overall arrangement is the omission of separate balancing shafts in the drive units (Figures 2-55 and 2-60). The resulting opposed engine concept has potential to make the drive units lighter and cheaper but carries a slightly higher risk.

3.3.2 Advanced Engine Concepts

In a direct fired engine the heater tubes embedded in the

fluidized bed may be looked upon as the technologically most difficult part of the system. It is important to find a simple and straightforward design which could give promise of low cost and long engine life. The simplest way to place the tubes in the bed is to arrange a bundle of straight tubes horizontally so that they lead straight across the bed from one wall to the opposite side. This suggests the idea of placing a cylinder on one side of the bed and the regenerator belonging to the same cycle on the opposite side. For the next cycle the cylinder and the regenerator change sides, which means that for the engine as a whole, half of the cylinders are on one side and half of the cylinders are on the other side.

It is possible to design both double-acting engines and single-acting engines based on this idea. A double-acting engine would need to have at least eight cylinders (two modules) to avoid carrying large torque variations from one side of the engine to the other. A single-acting engine could be operated as a single module but would still have at least eight cylinders, as a single-acting engine requires two cylinders per cycle. To avoid carrying the output from one side of an array of modules to the other side, it might be advantageous to use two alternators, one for each side. In this case the only connection between the two sides would be a synchronization device, for example a chain drive, which does not carry any permanent torque loads. In this way mechanical losses can be minimized.

The mechanical design of each side of one engine module would differ from the 4-275 concept and also from the alternate designs presented in this study. It will of course require a considerable amount of work to produce even a conceptual design drawing, but in principle the design procedure seems fairly straightforward. As far as power and efficiency are concerned the results shown in Section 2.3.1.3 can be used as a first approximation. In the absence of a

design study, it is difficult to say how large dead volumes on the cold side would be. However, the dead volumes on the hot side would be small. Thus the total dead volume should be no larger than for the state-of-the-art conceptual designs presented in chapter 2, and the performance should be at least as good.

The ideas presented above allow a completely symmetric fluidized bed with a uniform packaging of tubes in a vertical cross section. However, there must not be any vertical temperature gradient in the bed, which would adversely affect the uniformity of heat transfer.

Another specific difficulty is the thermal expansion of the tubes. It may be necessary to allow the two sides of the engine modules to move relative to each other in order to absorb this expansion. Alternatively, some bends in the tubes could accommodate the thermal expansion.

It is suggested that in parallel to a conceptual study of these ideas, heat transfer tests and thermal cycling of horizontal tubes in a fluidized bed should be conducted to get an early indication of any practical difficulties using this idea as well as data on the corrosive behaviour of tubes in this environment.

See Section 3.2.2 for other direct-fired concepts.

3.3.3 Advanced Engine Components

A number of objectives can be identified to guide engine component development efforts. Some of the most important of these are:

1. Reduction of heat losses and leakage by introduction of ceramic heat dams
2. Reduction of costs by use of cheaper heat-resistant material in the hot end of the engine
3. Reduction of mechanical friction losses by improved design of seals and bearings
4. Increase of cooling water temperature capability for total energy applications by development of new seal materials and/or a separate cooling water loop for the seals.
5. Reduction of facility down-time as well as O&M costs by improving lifetimes of wear sensitive Stirling engine specific parts - such as piston rings and main seal
6. Increase of thermal efficiency by increase of temperature in the hot end of the engine.

Introduction of ceramic heat dams, which do not themselves absorb structural loads, may be seen as the first step towards a ceramic engine. A ceramic piston dome will decrease the internal heat leakage between the hot and cold spaces in a cylinder. Ceramic inserts in the cylinder tops would reduce heat losses through radiation. They would also reduce the temperature on the inside wall of the cylinder and may in that way facilitate the use of cheaper stainless steel type metals in the pressure vessel.

Significant increases of operating temperature will be possible only if ceramics are used extensively in the heater. An all-ceramic heater has the potential of increasing thermal efficiency as well as reducing the cost of material. It may also alleviate the corrosion

problems in the fluidized bed and is thus very attractive in this application. There is, however, little to no experience with ceramic materials in Stirling engines, and until suitable designs have been produced and the producibility of a ceramic heater for a large engine has been analyzed, such a heater must be regarded as a high risk component.

The advanced concept ideas touched upon in Section 3.3.2 seem to be equally suitable for ceramic materials as for metal materials. It may well be that this concept should be pursued as an all ceramic engine from the beginning. Such an engine should at least be included as a design alternative in further conceptual studies.

The modern Stirling engine has not yet reached maturity because of limited production and experience. This will of course also be true for the first generation of large Stirling engines. There are many small component changes that will likely take place in succeeding generations. Such changes will improve the Stirling cycle efficiency and power density by reducing working gas and cooling water flow losses as well as improving heat transfer and flow and temperature distributions inside the engine. They will also contribute to longer engine life and make the engine cost slowly approach that of a comparable Diesel engine.

3.4 ADVANCED SYSTEM CONCEPTS

In Section 2.4.5 (Figure 2-72) three advanced systems were compared against state-of-the-art systems. A brief discussion of each follows. Before proceeding it is important to note that the advanced systems which are discussed here take no credit for improved component designs which could lead to improved system performance, but rather merely reflect higher heater and/or cooler temperature capability.

IB Cogeneration

The "advanced" cogeneration example is different from the state-of-the-art in two important ways. First, the heater temperature is raised to 820°C. From Figure 2-25, this increases engine brake thermal efficiency by a factor 1.12 (compared to 720°C). Secondly, the cooler temperature was increased to 120°C. From Figure 2-26 this decreases the engine efficiency by a factor of 0.77 (compared to 50°C).

The higher heater temperature improves engine performance and is thus desirable for advanced engines. However, the higher cooler temperature drops the efficiency to a greater extent than the higher heater temperature increases efficiency for a resulting decrease in engine efficiency from the nominal state-of-the-art ($T_{HH} = 720^{\circ}\text{C}$, $T_C = 50^{\circ}\text{C}$). However, it is anticipated that the increased cooler temperature capability will widen the market for Stirling engines by allowing the engine to be used in cogeneration applications which require 120°C heat. However, it will still be advantageous to run the cooler temperature as low as the application allows.

IB Electric

The advanced coal fired Stirling electric powerplant simply takes advantage of a low cooler temperature (30°C) and an increased heater temperature (820°C). The brake thermal efficiency of the engine is then 0.47 (see Figures 2-25 and 2-26). With all losses and auxiliaries included, the coal pile to busbar efficiency becomes 0.34. Lower cooler temperatures and component improvements could increase this figure still more.

XP-360

The XP-360 represents the case where the engine is optimized for high efficiency. This could be, for example, a "low" solar power tower driving an irrigation system where the irrigation water is used to cool the engine. In such a case, the heat input efficiency is essentially 1.0. The heater temperature is taken to be 820°C and the cooler 15°C. Hydrogen (not Helium) is the working fluid. Engine speed is 600 RPM and mean pressure is 11 MPa. The brake thermal efficiency is then 0.59! Assuming a 95% efficient generator and 2% of engine power for auxiliaries (controls, oil pump, etc.) the "solar energy to busbar" efficiency is 0.55.

Because these numbers are derivable from state-of-the-art experience (requiring only higher heater head temperature) it is reasonable to expect that such performance is possible in the near term (1985) provided sufficient development effort is expended.

4. SYSTEM COST ESTIMATE

In this section the equipment, operating and maintenance costs for the various sub-systems are combined to provide a preliminary estimate of Stirling system costs. In Section 4.1, the equipment costs of the state-of-the-art coal-fired engine are estimated. Two estimates are presented. One assumes the use of a current technology, "one-of-a-kind" heat source sub-system. The other assumes the development of a "packaged", pre-engineered FBC heat source which provides substantial reductions in equipment and installation costs.

In Section 4.2 the equipment, installation, operation, maintenance, and capital costs are combined to produce an estimate for the costs of producing electric power from a coal-fired Stirling engine. Finally, in Section 4.3, the influences on system cost of certain application dependent parameters, such as waste heat utilization, are examined.

Since the cost estimates in this report are based upon a conceptual design, a certain degree of approximation is involved in all of the cost figures. The reader is referred to the sections detailing the various sub-systems for information regarding source and methodology of cost figures. Moreover, it should be understood that this composite cost estimate uses different methodologies for the various sub-systems. Specifically, the cost estimates for the engine and heat pipes are based upon manufacturer's price FOB factory at a production volume of 1,000 modules (400 kW) per year. On the other hand, some cost estimates for the heat source sub-system

are based on vendor quotes, and thus refer to end user cost. The O&M costs are based on users' costs. These differences in cost basis should be borne in mind when considering the estimated cost of power produced by the system.

Finally, it should be noted that the cost estimates in this section are for the most part based upon present state-of-the-art technology. No estimates are presented for the costs of a mature engine system which would be based upon an advanced technology. While it is reasonable to assume that a mature technology coal-fired Stirling engine would be less expensive, it is felt that any estimates of cost at this time would be highly speculative.

4.1 EQUIPMENT COSTS

The cost estimate in this section assumes the use of current state-of-the-art components, that is, the designs discussed in Section 2. The cost elements, as shown in Figure 4-1, are presented in terms of dollars per kW of net electric output, based upon a net electrical output of 392 kW per module after system auxiliary requirements are met (See Figure 2-68).

Two separate estimates are presented in Figure 4-1. The first assumes the use of current state-of-the-art, low production volume, custom engineered heat sources. The second estimate assumes the availability of a volume produced, pre-engineered, packaged FBC subsystem.

Engine

The engine cost of \$270/kW is based upon the United Stirling cost estimate at a production level of 1,000/year, as described in Section 2.3.4.1. The cost of the generator, controls, heat recovery

FIGURE 4-1: ESTIMATED COST OF A COAL-FIRED STIRLING ENGINE

	Low-Volume Production Heat Source \$/kW(e) _{net}	High-Volume Production Heat Source \$/kW(e) _{net}
Engine	270	270
Generator, Controls, Heat- Recovery, Installation	<u>195</u> 465	<u>195</u> 465
Heat Source	221	129
Heat Transport	64	49
Installation	<u>221</u> <u>506</u>	<u>65</u> <u>243</u>
SUBTOTAL	971	708
Heat Source Auxiliaries	256	161
Installation	<u>256</u> <u>512</u>	<u>81</u> <u>242</u>
TOTAL	1483	950

equipment and installation are taken from reference 22 for a diesel engine. That estimate of \$150/kW has been increased to \$195/kW to allow for 30% inflation from 1977 to 1980. The resulting cost of \$465/kW compares favorably with present day installed costs of diesel total energy systems, which range from \$500/kW to \$650/kW (25). As another point of comparison, allowing 30% for inflation between 1977 and 1980, reference 22 reports a cost of \$537/kW for a 400 kW diesel total energy system. Thus references 22 and 25 are in agreement. The Stirling cost, of course, does not yet include the cost of the heat source, but it is representative of what an oil-fired version would cost.

Heat Source and Heat Transport

The heat source cost estimate is based upon the costs shown in Section 2.1.7. To convert from heat source heat output to net electrical output, the flow chart shown in Figure 2-68 is used; i.e., 392 kW(e) are produced from 1180 kW. The \$221/kW estimate is an average of the "A" and "C" FBC modules described in section 2.1.7. These represent custom-engineered, field-assembled, "one-of-a-kind" systems. This compares with \$129/kW, which is the average of the "B" and "D" versions, and which are assumed to represent serially-produced, pre-engineered, pre-packaged FBC combustion systems. It should be noted that since these cost estimates are based upon vendor quotes, they are not directly comparable with the other (FOB factory) costs.

The costs of the heat transport system are taken from Section 2.2.6. The \$64 figure represents near-term volume production of the two-stage heat pipe at \$25,000 per module. The \$49 figure in the

second column is based upon the estimate of \$19,200 per module, which was given as the "learning curve" estimate in Section 2.2.6.

Installation cost is assumed to differ markedly between the two heat sources. The \$221 installation cost is based upon 100% of the equipment cost of the heat source, since this would be a custom-engineered, field-erected system. The \$65 cost estimate for the volume produced system is based upon 50% of heat source equipment costs, and is based upon a standardized, pre-engineered, factory-assembled system.

The costs up to this point are representative of a coal fired Stirling engine installed in a plant that is already burning coal, so that substantial additional costs for coal and ash handling equipment are not necessary. At this level, the costs of the state-of-the-art Stirling engine system are from about 40% to 100% higher than for a diesel system.

Heat Source Auxiliaries

Where coal, sorbent and ash storage and handling equipment and a baghouse are required to complete the engine installation, the installed cost of the heat source is roughly doubled. The costs of \$512 and \$242 per kW are taken from the same sources as, and are based upon the same assumptions as the preceding heat source estimates. Again, the pre-packaged, volume produced system is expected to produce significant savings in installed costs.

The final totals of \$1,483 per kW (net electric) for the "one-of-a-kind" coal-fired system and \$950 per kW (net electric) for the volume-produced engine result in installed costs which are roughly

equivalent oil-fired diesel.

This example illustrates the high cost of burning coal compared to oil (or gas). As will be seen in Section 4.2, the sizeable difference in capital cost is compensated by the significantly lower operating cost of the Stirling engine. Obviously, however, the Stirling engine would benefit greatly from reductions in coal combustion equipment costs, as would all other industrial coal combustion applications.

4.2 POWER PRODUCTION COSTS

In this section the equipment cost estimates of the previous section are combined with operation and maintenance costs and capital charges in order to develop an estimate of the cost of power produced by a coal fired Stirling engine. Thus, costs are based upon the use of state-of-the-art components as discussed in Section 2.

The O&M estimates assume 8,000 operating hours per year at an average of 70% load factor. Thus, the annual electric power production of a 392 kW (net electric) system would be 2,200,000 KWH.

Engine O&M costs are taken from Section 2.3.4.2, adjusted for losses and 70% load factor. Here it should be emphasized that service and maintenance intervals are based upon current projected life of engine components, not their design goals. Typically, the design goals are about twice the life currently projected. Assuming that the design goals could be met without significant increases in material costs, mature technology engines could have significantly lower O&M costs.

O&M costs for coal combustion equipment are quite difficult to estimate as they are highly dependent upon the nature of the fuel being burned and the process conditions. Here, we have assumed the O&M costs (not including fuel, sorbent, or ash removal) to amount to 10% of equipment costs per year.

The cost of the heat transport O&M is based upon a service life of 16,000 hours for the primary heat pipes and 32,000 hours for the secondary heat pipes. Because the secondary heat pipe must be

replaced whenever the heater head tubes require replacement (since they are a hermetically sealed unit) the secondary heat pipes would be replaced every 16,000 hours in the near-term. However, when the heater design goal of 32,000 hours is reached, then this service interval should apply to the secondary heat pipe as well. O&M costs for heat source auxiliaries and for the electric generator, controls, and heat recovery equipment are estimated to be 10% of equipment cost, which is estimated to be \$150/kW.

Operating costs are based upon the use of coal at \$1.65/10⁶ Btu (27); limestone at \$6.50 per ton (26) and a 2:1 ratio of coal to limestone. Ash removal is estimated to cost \$7.50/ton (26). Referring to the flow diagram of Figure 2-68 the coal requirement is 1.03 lb/KWH, or a cost of \$.02/KWH. Similarly, the limestone requirement is 0.52 lb/KWH resulting in a cost of \$.002/KWH. Assuming that the coal contains 10% ash, the total weight of coal ash plus sorbent (which must be disposed of) amounts to 0.6 lb/lb of coal, or 0.62 lb/KWH. This results in an ash disposal cost of \$.002/KWH.

The resulting total O&M cost of \$.05/KWH (Figure 4-2) compares with \$.08/KWH for a Diesel generator, as detailed in Figure 4-3. It is this favorable comparison that enables the Stirling engine to be competitive with the Diesel despite its higher capital costs.

The overall cost of power from a coal-fired Stirling engine is shown in Figure 4-4. Capital costs (high volume estimates in Figure 4-1) are based upon a capital charge rate of 25%, which is intended to include taxes and insurance as well. It is seen that at \$.043/KWH the capital costs are of the same order as the O&M costs, with a resultant total cost of power of \$.093/KWH.

**FIGURE 4-2: ESTIMATED OPERATION AND MAINTENANCE COSTS
OF A COAL-FIRED STIRLING ENGINE**

	<u>\$/KWH</u>
Engine	.015
Heat Source	.002
Heat Transport	.004
Heat Source Auxiliaries	.003
Generator, Controls & Heat Recovery	<u>.002</u>
Subtotal, Non-fuel	.026
Coal @ \$1.65/10 ⁶ Btu (27)	.020
Sorbent @ \$6.50/ton (26), 2:1 coal-to-limestone	.002
Ash Disposal @ \$7.50/ton (26), all sorbent plus 10% of coal	<u>.002</u>
Subtotal, Fuel	<u>.024</u>
<u>Total</u>	<u>.050</u>

BASIS: 8000 hrs/yr, 70% load factor

FIGURE 4-3: ESTIMATED DIESEL O&M COST

	<u>\$/KWH</u>
Repairs & Maintenance (25)	.012
Fuel Oil @ \$.90/Gal (27), 0.077 Gal/KWH (25)	<u>.069</u>
Total Diesel O&M Cost	.081

FIGURE 4-4: ESTIMATED ELECTRIC POWER COSTS OF A
COAL-FIRED STIRLING ENGINE

(\$/KWH)

	<u>CAPITAL</u>	<u>O&M</u>	<u>TOTAL</u>
Engine, Generator, Controls, Heat Recovery	.021	.017	.038
Heat Source	.009	.002	.011
Heat Transport	.002	.004	.006
Heat Source Auxiliaries	<u>.011</u>	<u>.003</u>	<u>.014</u>
Sub-total, Non-fuel Costs	.043	.026	.069
Coal	-	.020	.020
Sorbent	-	.002	.002
Ash Disposal	-	<u>.002</u>	<u>.002</u>
Sub-total, Fuel Costs		.024	.024
TOTAL COST	.043	.050	.093

BASIS: 8000 hrs/yr, 70% load factor, 25% capital charge rate, high
volume heat source cost estimate

Assuming the installed cost of a comparable diesel engine to be approximately \$550/kW, the corresponding capital costs amount to \$.025/KWH. When the O&M costs from Figure 4-3 are added to this amount, the comparable cost of power from the diesel engine is \$.106/KWH, or approximately 14% higher than that of the Stirling.

It is important, however, to note the differences in the make-up of the Stirling engine power costs versus the diesel costs. While most of the diesel costs are attributable to the cost of fuel, only about 20% of the Stirling costs are due to fuel. Conversely, almost half of the Stirling engine costs result from capital costs, versus only about 25% with the diesel. Thus, the Stirling engine will be most sensitive to changes in equipment costs, and will gain greater competitive advantage relative to the diesel as fuel costs continue to rise.

Up to this point, system costs have been considered independent of the application, and the only useful product that has been recognized has been the engine shaft power (or electric output). However, it is clear that the most attractive applications for the Stirling engine will be those that can make good use of the available waste heat or have a low cost supply of difficult to burn fuel. This section examines the influences of some of these application dependent considerations.

While it is obvious that waste heat utilization improves the efficiency of fuel utilization, it is not so obvious that such waste heat utilization will always be economical, nor is it obvious how to calculate the economic benefits. As discussed in Section 2.4.3, the costs and benefits of waste heat utilization depend on a great many factors which need to be evaluated for the specific application in order to form conclusions regarding its value. However, once the application has been identified, in general it is possible to put a value on useable waste heat. Once this has been done, the effect of waste heat on the cost of power produced by the plant can be determined.

There are three significant factors that enter into the cost formula:

Operating Costs

Since waste heat utilization generally requires elevating the cooling water temperature above that which would normally be used,

the efficiency of the power plant is reduced (See Figure 2-26). This has a direct effect on operating costs, since more fuel must now be burned per unit of power output. However, plant efficiency has only a small effect on plant capital and maintenance costs.

Capital and Maintenance Costs

Raising the cooling water temperature of a Stirling engine reduces the rated power output (See Figure 2-26). Thus, if the same engine is to be operated with a higher cooling water temperature, its cost per kilowatt is increased proportionally (for the same load factor). Similarly, assuming that maintenance costs do not directly depend on the quantity of power produced, then the maintenance costs of a given engine will increase in terms of dollars/KWH.

Value of Waste Heat

The net cost of power produced by the Stirling engine will be reduced by the value of the useable waste heat. In valuing the waste heat, consideration must be paid to all of the avoided costs, that is, capital, operating and maintenance.

The power production costs taking into account waste heat utilization can be represented by the following equation:

$$\text{COST} = 0.069 \times P_{50}/P + 0.024 \times \eta_{50}/\eta - 0.0034 \times U \times V \times (1 - \eta)/\eta \text{ (\$/KWH)}$$

where: U = Utilization Factor, $0 \leq U \leq 1$
 V = Value of Waste Heat, $\$/10^6\text{Btu}$
 P = Rated Power, kW
 P_{50} = Rated power at 50°C , kW
 η = Net Engine Efficiency
 η_{50} = Net Engine Efficiency at 50°C

The coefficient 0.069 is taken from Figure 4-4, and represents the capital and non-fuel O&M costs. The coefficient 0.024 represents the fueling costs. The final term in the equation represents the credit due to useable waste heat. The term $(1-\eta)/\eta$ is a close approximation to the total quantity of waste heat available per unit of power output, and neglects differences between shaft power and electric power.

This equation is plotted in Figure 4-5 for various $U \times V$ products. For reference, $V = \$2/10^6$ Btu typifies burning coal in an 80% efficient boiler where only the fuel costs are considered; i.e., the boiler is fully depreciated and requires negligible maintenance. The case $V = \$8/10^6$ Btu corresponds to distillate fuel oil at \$.90/gallon being burnt in an 80% efficient boiler, where again only the fuel costs are recognized. Conversely, $V = 8$ could correspond to the total cost of energy from a coal-fired boiler recognizing all capital, operation and maintenance costs. It is seen that the latter two cases bring the cost of electricity produced by a coal-fired Stirling down to the range of utility electricity prices, i.e., \$.05 - .06/KWH.

It is also important to recognize that it is not just the value of waste heat that is important, but also its utilization. For instance, in Figure 4-5 if the alternative fuel is oil but the utilization factor is only 25%, then the $U \times V$ product is only 2.

It is apparent that if the value of waste heat is high enough, it should be possible to reduce the cost of electric power to zero. While this may appear paradoxical, it stems from the fact that the alternative energy source may be considerably more expensive to own and operate than the Stirling engine. An extreme example might be if the alternative was electric heating from the utility at \$.05/KWH (\$14.65/ 10^6 Btu).

FIGURE 4-5

ESTIMATED COST OF ELECTRICITY VERSUS
COOLING WATER EXIT TEMPERATURE AS A FUNC-
TION OF WASTE HEAT VALUE AND UTILIZ-
ATION FACTOR

Also, we should note that if the Stirling fuel is "free", e.g., industrial or agricultural waste, garbage, or perhaps biomass, and assuming the same plant costs, then the power costs would be reduced by approximately \$.02/KWH from Figure 4-4. This, of course, assumes that fuel handling and preparation costs do not rise above those for coal.

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APPENDIX A

HEAT PIPE ROTARY PROCESSING UNIT

The attractiveness of heat pipes for use in many applications is often predicated on the ability to produce the product by automated or semi-automated volume manufacturing techniques which provide cost effectiveness in the market place. The intent of this design study is to show by means of a simplified manufacturing model, the practicality of manufacturing heat pipes by volume at competitive labor costs.

Discussion

The ability to produce cost effective heat pipes in volume has been an obstacle for many applications. For this reason, effort has been devoted to the design of a multiple station rotary processing unit in which repeatable processing can be attained with a minimum of direct labor effort. The design is based on 10 minutes of heat pipe processing and an adequate number of stations in which to achieve a practical rotary index time. The loading and unloading function is the only station where direct labor is required. A total of 40 stations with a 15 second rotary index time is practical and results in a machine capable of producing approximately 375,000 heat pipes per year for a single eight-hour shift with 85% efficiency and 5% scrap.

A simplified model utilizing five stations has been selected for evaluation of the various technologies required.

The five-station design is seen in Figure A-1 and A-2. A flow chart of this concept is shown in Figure A-3. The index time for this device would be 2 minutes per station giving the 10 minutes of required processing time for each heat pipe. Thus, all station functions and operations are similar to the ultimate 40 station unit. It is thus obvious that the simplified (5) station unit will provide adequate data to properly evaluate the concept.

The multiple station processing unit design has been developed based upon the following:

- a. Use of precleaned and vacuum-fired tubing as the heat pipe vessel;
- b. use of a pinched, welded, cut-off closure;
- c. one end of pipe closed after vacuum firing;
- d. utilization of a rotary vacuum seal concept similar to rotary vacuum tube exhaust machines;
- e. a processing time of 10 minutes per heat pipe, and
- f. an adequate number of stations to achieve a practical index time for the particular heat pipe being processed.

Since the basic operation can be described by the (5) functional stations, each performing an operation pertinent to the processing of the heat pipe, these stations and functions are described in detail.

Station #1 - Unload/Load

This station is the start/finish point of the processing operation. As the unit indexes from the #5 station to the #1 station, the completed heat pipe is removed, the exhaust

Figure A-1

ROTARY PROCESSING UNIT

- STATION 1 LOAD & UNLOAD
 STATION 2 EVACUATE & HEAT (VACUUM PUMP 1)
 STATION 3 EVACUATE & COOL (VACUUM PUMP 2)
 STATION 4 EVACUATE, FILL & WRENCH OFF } PUMP 3
 STATION 5 EVACUATE & WELD

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Figure A-3

FUNDAMENTAL FIVE STATION FLOW CHART

appendage removed and the next heat pipe inserted for start of processing. This involves removal of the clamp which provides power for resistance heating of the pipe, opening of the vacuum quick-disconnect, inserting of the next heat pipe into the vacuum fitting and attachment of the heating clamps.

The proper design and selection of components will permit a cycle time of fifteen seconds to be adequate for these operations. If necessary, the heating clamp attachment can be automated. Immediately upon indexing from station #1 to station #2, evacuation begins and heat-up is initiated.

Station #2 - Evacuation and Bake-Out

The use of a rotary vacuum seal permits evacuation of the heat pipes with a minimum of external control valves and minimum vacuum plenum length.

Heating is accomplished by resistance means with power introduced through slip-rings placed around the rotary unit. A back-out of 400 C is assumed and power input clamps would be located an adequate distance from the vacuum seal to provide the necessary thermal isolation.

The 2 minutes at temperature is adequate based on the use of previously out-gassed components. In the 40 station processing unit, additional heating stations are added until the index time times the number of stations equals the desired time at temperature.

Station #3 - Cool-Down

Thermal cool-down of the heat pipe is required to keep the working fluid vapor pressure low upon insertion. The method of cooling can be adjusted as required in order to achieve a reasonable time. Again, additional stations are utilized.

Station #4 - Fluid Insertion/End Closure

The repeatable insertion of an accurate quantity of working fluid is required if consistent quality heat pipes are to be produced by high volume processing techniques. A fixed volume, pneumatic controlled metering pump enables accurate deployment of a fixed amount of working fluid in each heat pipe. Thus, operator judgment is eliminated and "look alike" should result. Adjustment can be made to accommodate varying heat pipe volumes and different working fluids.

Following fluid insertion, the heat pipe end toward the vacuum connector is "pinched" together by means of pneumatically controlled clamping jaws. While this operation does not provide a vacuum tight seal at the "pinch" area, it does prepare the area for welding performed at station #5.

Station #5 - Closure Welding

Final vacuum closure of the heat pipe is accomplished passing high current through the "pinched" area prepared at station #4. This is done by use of a stud-type spot welder

which applies high current and pressure to the weld area. The small section of heat pipe between the weld area and the vacuum fitting is clipped off when the completed heat pipe is removed at station #1.

Summary

A design of a prototype machine for the manufacture of heat pipes by high volume processing techniques is described. Practical processing technology along with innovative design provides manufacturing capability which makes heat pipes attractive for high volume, low cost applications.

APPENDIX B: PRESSURIZED FLUIDIZED BED COMBUSTION

Pressurized fluidized bed combustion has received considerable attention, particularly for running gas turbines in combined cycle systems. The benefits of PFBC are mainly a high bed heat release rate, low NO_x emissions and controllable SO_x emissions. Furthermore, if adequate particulate clean-up equipment could be built, then gas turbines could be run on coal.

While the gas turbine is suitable for PFBC operation, the Stirling engine is not. In the Stirling application, if the power required to compress the gas is not recovered in a hot gas expander, then a sizeable fraction of the Stirling power output would be lost. Thus, a "turbocharger" concept would be required. A free standing single shaft compressor/turbine package could be installed, and with some design effort could be self-sustaining.

The benefit of this arrangement would be the elimination of the 5% of Stirling gross power output lost to combustion air fans, and possible size reduction in hardware. However, the pressurized concept greatly complicates sealing, coal and limestone feeding and ash removal. Furthermore, there is the capital and maintenance costs of the turbine/compressor and there is the cost of a hot gas particulate clean-up device (in order to protect the turbine). Last but not least, because the Stirling FBC is limited in heat release rate by the heat transfer to the in-bed heat pipes (see Section 2.1.5) it would probably be the case that the Stirling application could not use a smaller FBC and thus would not benefit from pressurization.

Accordingly, PFBC does not warrant further study for the Stirling application.

APPENDIX C

HEAT PIPE COST ANALYSIS

A detailed cost analysis for volume production of the primary heat pipes was performed. Cost analyses have been performed in the past for similar heat pipes, and Thermacore drew upon their experience for this analysis. More detail follows.

Primary Heat Pipes

A 32,400 ft² plant with five production lines is envisioned. This plant will work three shifts a day, 5 days a week, 50 weeks a year. The capacity of the plant is 360,000 heat pipes per year or one heat pipe every five minutes from each production line.

In the cost analysis, a daily plant rate of 1050 heat pipes is used which requires one heat pipe every six minutes for 21 hours. This gives a yearly production rate of 262,500 in order to get a net production of 254,000 good heat pipes.

Table C.1 lists the indirect and direct labor required and their salary range. Table C.2 lists the necessary equipment and capital cost. Table C.3 delineates the generation of the overhead rate and Table C.4 shows the calculation of the unit cost.

The cost of one primary heat pipe is \$30.53. Thus, the cost of 254 heat pipes for a 500 hp module is \$7,755 (1979 dollars). Applying 10% inflation allowance to reach the cost in 1980 dollars gives a total cost of \$8,530 for the primary array.

TABLE C.1

LABOR COSTS

<u>Indirect</u>		<u>Direct</u>
1. Plant Manager	\$50K	3-Operating Foreman
2. Asst. Plant Manager	35K	4-Shipping and Receiving
3. Sr. Engineer	40K	3-Inspectors
4. Jr. Engineer	25K	2-Degreasers
5. Jr. Engineer	25K	2-Parts Maker
6. Jr. Engineer	25K	3-Vacuum Furnace Operators
7. Supervisor	35K	3-Electron Beam Welder Operators
8. Supervisor	35K	3-Rotary Processor Operators
9. Supervisor	35K	3-Coating Machine Operators
10. Secretary	15K	3-Warehouse Men
11. Secretary	10K	3-Testers
12. Purchasing Agent	30K	3-Maintenance Men
13. Buyer	20K	3-General Help
14. Personnel	30K	<u>3-Hydrogen Furnace Operators</u>
15. Comptroller	35K	41-Direct Laborers @ average \$7/hr.
16. Accountant	25K	or \$14,560/yr.
17. Clerk	10K	
<hr/> Total		<hr/> Total Direct Labor = \$596,960

TABLE C.2

EQUIPMENT

1. Vacuum Furnace 15' x 4' dia. x 10 ⁻⁶ TORR	5 @ \$180K	\$ 900K
2. Hydrogen Furnace	5 @ \$80K	400K
3. Electron Beam Welder	5 @ \$400K	2,000K
4. Plasma Arc Sprayers	5 @ \$50K	250K
5. Wick Formers	1 @ \$100K	100K
6. Grooving Machine	1 @ \$50K	50K
7. Punch Press and Dies	1 @ \$50K	50K
8. Parts Degreaser	1 @ \$25K	25K
9. Conveyor System and Carts		50K
10. Test Stand	5 @ \$80K	400K
11. Welders	3 @ \$5K	15K
12. Fork Lift	2 @ \$5K	10K
13. <u>Miscellaneous Equipment</u>		<u>100K</u>

Total Equipment

\$4,350K

15% Depreciation

\$653K

TABLE C.3.

OVERHEAD RATE

Rent \$64K

2,400 ft² Office @ \$2.50/ft² = \$ 6,000

25,000 ft. Factory @ \$2.00/ft² = 50,000

5,000 ft. Warehouse @ \$1.50/ft² = 7,500

\$63,500

Utilities 150K

Indirect Labor 480K

Benefits 30% all labor 323K

Unapplied 10% direct labnr 60K

Insurance 25K

Office Equipment Depreciation and Miscellaneous 25K

Factory Equipment Depreciation 653K

Total Indirect Costs \$1,716K

Overhead = $\frac{\text{Indirect Costs}}{\text{Direct Costs}} = \frac{\$1,716K}{597K}$

Overhead = 287%

TABLE C.4

UNIT COST

Primary Heat Pipe

18 SR	3.45 lbs @	\$1.74/lb average	\$6.00
Sodium	0.11 lbs @	\$19/lb	2.00
Coating	0.28 lbs @	\$13.50/lb	3.38
Totals	3.84 lbs @	\$2.96/lb	\$11.38
Material Scrap - 6%			.68

Total \$12.06

Labor Costs = $\frac{\text{Total Direct Labor}}{\text{Net Good Heat Pipes}}$

Direct Labor = \$596,960/254,000

Direct Labor = \$2.35/heat pipe

Direct Labor = \$2.35

Overhead 287% = 6.74

Subtotal = \$9.09

Material Costs = 12.06

Subtotal = \$21.15

G & A - 10% = 2.11

Subtotal = \$23.26

Warranty 5% = 1.16

Subtotal = \$24.42

25% Profit = 6.11

Total \$30.53

Heat Pipe Cost = \$30.53

254 Heat Pipes = \$7,755 (1979 Dollars)

Secondary Heat Pipe

The calculation of the per unit cost is shown in Table C.5. For simplicity the same overhead rate is applied as for primary heat pipes, and the labor estimates are "best guess". The cost of a secondary heat pipe is \$15,000 (1979 dollars). Applying 10% inflation allowance gives a 1980 cost of \$16,500. Thus, the total heat transport cost is \$25,030 per module, or \$25,000 rounded off. At 1180 kW of heat transfer capability this cost can be expressed as \$21.20 per kilowatt of heat transfer capability.

TABLE C.5

UNIT COST

Secondary Heat Pipe

18 SR	2150 lbs @ \$1.83/lb average	\$3935
Sodium	40 lbs @ \$19/lb	\$ 760
Wick	60 lbs	\$1020
	120 ft ² 60 mesh \$2.50/ft ²	
	120 ft ² 200 mesh \$6.00/ft ²	
Totals	2250 lbs	\$5715
Material Scrap - 6%		\$ 345
Total For Materials		\$6060

Direct Labor = 120 hours

Direct Labor Rate = \$7.00/hr

Direct Labor Total = \$1120

Overhead 287% = \$3215

Subtotal = \$4335

Material Costs = \$6060

Subtotal = \$10395

G & A - 10% = \$1040

Subtotal = \$11435

Warranty 5% = \$ 570

Subtotal = \$12005

25% Profit = \$3000

Total \$15000 (1979 DOLLARS)