110
8/26/80 T.S.

DR-1733

ENERGY

DOE/ET/15207-T1

NO-H-K-T-H-02

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

First Quarterly Report

January 7, 1980

MASTER

Work Performed Under Contract No. AC02-79ET15207

Advanced Mechanical Technology, Inc.
Newton, Massachusetts

U. S. DEPARTMENT OF ENERGY

Division of Electric Energy Systems

DISCLAIMER

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

DISCLAIMER

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

DISCLAIMER

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

This report has been reproduced directly from the best available copy.

Available from the National Technical Information Service, U. S. Department of Commerce, Springfield, Virginia 22161.

Price: Paper Copy \$8.00
Microfiche \$3.50

January 7, 1980

FIRST QUARTERLY REPORT

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

CONTRACT NO. DE-AC02-79ET15207

Submitted to:

Component Technology Division
Argonne National Laboratory
Building 330
9700 South Cass Avenue
Argonne, Illinois 60439

Submitted by:

Advanced Mechanical Technology, Inc.
141 California Street
Newton, Massachusetts 02158

TABLE OF CONTENTS

	<u>Page</u>
Table of Contents	ii
List of Figures	iii
List of Tables	iv
1.0 Introduction and Summary	1-1
1.1 Project Background and Objectives	1-1
1.2 Technical Approach	1-1
1.3 Project Status	1-3
2.0 Heat Source	2-1
2.1 Initial Choices	2-1
2.2 Selection Criteria	2-2
2.3 Discussion	2-5
2.4 Tentative Selection	2-13
References	2-15
3.0 Heat Transport	3-1
3.1 Initial Choices	3-1
3.2 Selection Criteria	3-1
3.3 Discussion	3-2
3.3.1 Liquid Forced Convection	3-2
3.3.2 Direct Fired By Fluidized Bed	3-3
3.3.3 Gas Convection Loop	3-13
3.3.4 Sodium Heat Pipe	3-21
4.0 Engine Conceptual Design	4-1

LIST OF FIGURES

<u>Figure</u>		<u>Page</u>
1	Coal Combustion Comparison	2-6
2	Direct Fired Heater Head	3-9
3	Direct Fired With Two Regenerators per Cylinder	3-10
4	Direct Fired With Long Dimension Horizontal	3-11
5	Direct Fired With Long Dimension Vertical	3-12
6	Heater Head Pumping Power	3-18
7	Duct Pumping Power	3-19
8	Heat Pipe Diameter for Various Evaporator Radial Heat Fluxes	3-26
9	13 kW Sodium Heat Pipe	3-27
10	Required Heat Pipe Diameter	3-28
11	U-4 Automotive Engine	4-2
12	Shaft Power	4-6
13	Efficiency	4-7
14	Influence of Heater Temperature on Power and Efficiency	4-9
15	Influence of Cooler Temperature on Power and Efficiency	4-10
16	Coupling Between Heat Supply to One Cycle and Heater Temperature and Power Output	4-11
17	Crankcase, Approximate Envelope	4-13
18	External Heat System	4-14
19	Cylinder Block	4-18
20	Cold Engine System	4-19
21	Alternate A	4-21
22	Alternate B	4-22
23	Alternate C	4-24
24	Alternate D	4-25
25	Alternate E	4-26
26	Alternate Arrangement	4-31
27	Heat Pipe Alternative	4-33

LIST OF TABLES

<u>Table</u>		<u>Page</u>
1	Assumptions and Nomenclature	3-4
2	Potential Direct Fired Heater Head	3-6
3	Convection Loop Gas Selection Data	3-16
4	Heat Pipe System Specifications and Requirements	3-22
5	Heat Pipe Specifications	3-29
6	Wick Specification and Performance	3-31
7	Heat Pipe Material Selection	3-33
8	Mechanical Properties of Abrasion Resistant Coatings	3-36
9	Individual Heat Pipe Cost Based on 12,000 per Year	3-39
10	Engine Scaling Results	4-4

1.0 INTRODUCTION AND SUMMARY

1.1 Project Background and Objectives

This project is 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 comprises 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 is 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 for commercialization. The conceptual designs include a heat transport system for integrating the engine heater head with such energy sources as conventional oil/gas combustors, fluidized bed and other coal combustors, and combustors using coal-derived liquid fuels, and low/medium BTU gases.

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

1.2 Technical Approach

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; and the "B" designs which involve advanced technology and therefore would represent a greater risk for demonstration by 1985, but which may offer advantages in terms of lower cost, better performance, or higher reliability. The majority of the effort in Phase I is to be devoted to the "A" designs.

The basic technical approach has been to base the large engine design on a "scale-up" of the state-of-the-art United Stirling P-75 engine. This is a "U-4" type which has 4 cylinders in a square array connected to twin crankshafts. In its P-40 and P-75 versions, nominally 40 kW and 75 kW, respectively, the U-4 concept has achieved a high state of development. Fortunately, Stirling engines may readily be scaled so extrapolation of the P-75 design is expected to yield meaningful results. To take advantage of the economies of mass production, commonality of parts is to be maximized by using multiples of a basic 500 horsepower module to build the 500 to 3000 horsepower engine line.

The large engine will be similar to present day automotive Stirling engines in most aspects other than the heat input system. Because of the complexity and larger bulk of most solid fuel combustion systems, it is highly likely that the combustor will be more remote from the engine than the present highly integrated oil/gas combustor. Accordingly, the approach to be taken for the design of the heating system involves the design of a heat source sub-system along with a heat transport sub-system.

Thus, the overall conceptual design is broken down along the major lines of heat source, heat transport, engine module, and multi-module design, as detailed below.

Heat Source

Pulverized

Stoker

Cyclone

Fluidized Bed

Low-BTU Gas

Heat Transport

Forced Convection Liquid

Forced Convection Gas

Heat Pipe

Direct Fired (Oil/Gas and Coal)

Engine Module

Scaled-Up P-75

Multi-Module

Vee

In-Line

Star

Regarding the conceptual design of the heat source, it should be pointed out that it is not the purpose of the present study to duplicate previous or current conceptual designs of combustion systems. Rather, the objective is to evaluate current and emerging technologies to determine which method of combustion is best suited to the stationary Stirling engine. Since it would not be practical in the time-frame and scope of Phase I to examine combustion systems for all fuel types, it has been decided to concentrate the conceptual design on combustors capable of burning coal as the primary fuel, the reasoning being that this would be the most widespread fuel type, and that a combustor capable of burning coal would be amenable to firing a number of other fuels as well.

1.3 Project Status

Currently, the leading choice for the solid fuel combustion system is the atmospheric fluidized bed, with low BTU gasification still a viable alternative. Both systems will continue to be evaluated further, but with greater emphasis on FBC.

To date, there appears no clear choice among the heat pipe, forced convection gas loop, or direct firing as the prime candidate for the heat transport sub-system. Conceptual design and analysis will continue on all three sub-systems.

United Stirling is continuing with the conceptual design of the 500 horsepower engine module based upon a scale-up of their P-75 engine. Multi-module configuration studies are being delayed until a clearer choice appears among the candidate heat transport sub-systems, as this choice can have a decided effect on the appropriate engine configuration.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

2.0 HEAT SOURCE

2.1 Initial Choices

Examining the state-of-the-art in coal combustion technology, five choices for the heat source "firebox" were selected for evaluation. All five are systems which are, in general, 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 historically.
- c) Cyclone - Utilizing a very intense vortex chamber, cyclone combustors achieved a 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 subtask 1A evaluation.

2.2 Selection Criteria

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

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 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 1/2 of the national standard. Oxides of nitrogen, however, are uncontrolled. The 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". Each combustion system is evaluated 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 criterion is primarily intended toward the use of a wide range of coals. A secondary consideration is the capability of burning biomass. 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. Thus, typically an installation is tailored to burning a specific coal type. Combustion systems which are relatively insensitive to coal properties are therefore desirable in the design of a coal installation in particularly when one considers "mass producing" a single design.

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 accommodating multi-engine installations are most desirable.

Efficiency of the combustion system is of prime importance as a screening criterion rather than a ranking criterion. 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. 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 limits the transfer of heat from the source to temperatures of about 1500⁰F and higher. Thus, the remaining heat energy in the combustion gases must be recovered by an air preheater.

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.

2.3 Discussion

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

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 useful 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

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 5) Baghouse 6) Ash & Limestone Disposal	gas oil coal(any) wood garbage	Meets all existing Federal Standards. Local standards might be prohibitive. Spent limestone disposal might be small problem.	3.5:1	Combustion >95% Total preheat is possible.	Beds suitable for 500-3000 Hp are proven. Conducive to modular approach.	15×10^6 ETU/hr $\$1.5 \times 10^6$ for total system. (Rough estimate for one-of-a-kind)	1) <u>AUXILIARIES</u> - 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) Might be direct or indirect heat transport.
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 sulphur 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> $45-90 \times 10^6$ BTU/hr depending on the coal.	<u>RILEY:</u> Gasifier \$950,000 Wet Scrubber \$250,000	1) Physically large. 2) Heater tube slugging & fouling.
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, possibly wood	NO _x problematic Wet scrubber required. Acid sludge disposal problem.	Large ratio	Combustion: High Little to no preheat (450°F max) Must make steam	Our sizes available in steam generation. However Nothing available at our temperature.	Low except wet scrubbers usually expensive	Inability to preheat above 450°F and tube slugging rule out the stoker. NO _x is a problem and wet scrubbers are a problem.
CYCLONE COMBUSTION	Same as stoker except: no stoker 1) Cyclone combustor 2) Coal Preparation?	coals	NO _x prohibitive Wet scrubber required Acid sludge disposal problem	?	?	large	prohibitive	Advanced cyclone gasifiers might be considered for LB concepts.
PULVERIZED COMBUSTION	Same as cyclone except no cyclone 1) Pulverized combustor 2) Coal preparation OR Different coal handling with safety equipment	coals	NO _x problematic wet scrubber required Acid sludge disposal problem	?	?	utility only	prohibitive	

FIGURE 1: COAL COMBUSTION COMPARISON

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 boilers. In the Stirling application, however, approximately one-half 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 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 is prohibitively low.

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 of the order of \$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 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 the more efficient ash and dust collection and the costs of pulverization. It is 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 emission standards 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

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 nearly as easily as natural gas. 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 direct coal fired systems. Sulfur, present in the low BTU gas as H_2S , appears as SO_2 in the stack and requires scrubbers. 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. At least on the surface it appears that the Stirling may 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.

Fuels flexibility is limited. The gasifier is usually set up for specific coals, although it is modifiable for other coals.

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).

Efficiency is acceptably high so long as the gas can be burned "dirty" (i.e. direct from gasifier with no sulfur clean-up and/or gas cooling before combustion). Full air preheat may be utilized.

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 the gas combustion system with a tube cleaning system and stack clean-up equipment.

Fluidized Bed Combustion

Although not widely accepted as yet, FBC is commercially available and rapidly gaining favor as a practical coal combustion technique.

For the Stirling conceptual design, the FBC (atmospheric) is considered state-of-the-art, as units are now commercially available.

Emissions are a solid plus in favor of FBC. With the appropriate sorbent material in the bed, SO_2 is easily controlled. With the low bed temperatures, oxides of nitrogen are not a problem. A hot gas cyclone and cold gas baghouse are required for particulate control.

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

Size flexibility appears good. FBC boilers are commercially available in the heat rates required by the 500 to 3000 hp Stirling (11) and are under serious development for utility size installations. Scaling of FBC systems is typically by bed cross-sectional area.

Efficiency is acceptably high when particulate recycle is used. Full air preheat is possible.

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

An important characteristic not included among the five previously

described 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 heat exchanger slag is not readily cleanable, and thus is critical to the heat exchanger design. It is the practice in conventional boilers to radiantly cool the combustion products below the ash-fusion temperature before passing the gases over convective heat transfer surfaces. Thus, the slag does not have the chance to solidify on heat exchanger surface. In the intended application, however, this practice may not be practical due to the high working gas temperature which is only a few hundred degrees below typical coal ash-fusion temperatures. For the Stirling application then, either the molten ash would have to be removed from the flue gases to prevent heat exchanger slagging, otherwise the primary heat exchanger would have to rely predominantly on radiative heat transfer, and the hot gases would then be convectively cooled in a combustion gas preheater.

FBC is an exception to this generalization 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.

2.4 Tentative Selection

Based upon the evaluations to date, fluidized bed combustion appears to be 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 turn out to be optimistic. In addition, it is not yet clear whether FBC may have corrosion/erosion problems, 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 not unreasonable costs for multi-engine installations, and this approach makes possible a relatively simple direct fired heater head, thus simplifying the engine design.

These two prime candidates will continue to be evaluated in further detail, while no further evaluations are planned for the other combustion systems.

REFERENCES

1. Federal Register, September 19, 1978
2. Personal Communication with Boston EPA Office
3. Seibold, P. F. and Bessette, R. D., "Latest Equipment and Design For New Coal Fired Plants and The Conversion of Existing Plants to Coal Firing," Riley Stoker Corporation Technical Reprint 762-H.
4. Williams, R. J., "Coal and Refuse Combustion For Efficient Utilization", Riley Stoker Corporation Technical Report 772-H.
5. Steam-Its Generation and Use, 37th Edition, The Babcock & Wilcox Company, New York, N.Y. 1963, P. 17-1.
6. Baker, T.K., "Options For Burning Coal in Industrial Boilers," Coal Technology '78, Vol. 1, P. 189.
7. Riley Stoker Corporation, Publication 8-76AP-10MD, "Atrita Pulverizers."
8. Ctvrtnicek, T.E. and Rusek, S.J., "Cyclone Boilers - Their NO_x Emissions and Population," Proceedings of The Second Stationary Source Combustion Symposium, Vol. 1, EPA-600/7-77-073a, July 1977.
9. Personal Communication with Riley Stoker Corporation.
10. Keed, R.R., "Fluidized-Bed Combustion For Small Users of Coal", Coal Technology '78, Vol. 1, P. 172.
11. Johnston Boiler Company, Form No. FF-2-6-78.
12. Farmer, M.H., Proceedings of Fluidized Bed Combustion Technology Exchange Workshop, CONF-770447-P-1,2, April 1977.

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

3.0 HEAT TRANSPORT

3.1 Initial Choices

The initial approach was to evaluate 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 fluidized bed combustor (FBC) and the low BTU gasification process seem to have this potential for direct firing.

3.2 Selection Criteria

The decision as to which system is the best is not an easy one to make. Each of the systems has its own set of problems, advantages and disadvantages. Furthermore, this difficulty is compounded by the fact that the choice of heat transport device can have a decided effect on the engine configuration, especially when multi-module engines are considered. 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 intangible or qualitative points for comparison. We need to know how to rank the relative importance of these, so as to be able to come up with a final ranking of the systems we are comparing. Looking back at the list, each factor 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 require-

ment can be reflected as an incremental cost of additional electrical generation capacity plus the operating cost of the fuel required to generate that electricity. Field serviceability affects repair and maintenance cost. Heater head void volume must be considered as an incremental cost because of its effect on engine specific power. The effect on engine configuration is only important insofar as it affects materials and manufacturing cost, unless we find some other effect 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).

In conclusion, the final selection will hopefully boil down to life-cycle cost, safety problem potential, and technical risk. These three factors will then have to be considered in a somewhat subjective manner, unless one of the heat transport systems is clearly the best.

3.3 Discussion

3.3.1 Liquid Forced Convection

For any liquid forced 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 liquid metals are extremely corrosive and difficult to handle. For example, small quantities of liquid sodium and water mixed together in air will explode spontaneously. The larger the sodium inventory, the greater the danger. For this reason we reject the liquid forced convection loop. We feel that while this system might have some small advantages over the sodium heat pipe, (the liquid sodium loop would probably be field serviceable and the heat pipe is definitely not) the advantages are trivial compared to the principal disadvantage, which is an order of magnitude increase in sodium inventory.

3.3.2 Direct Fired By Fluidized Bed

The direct fired engine was considered for both the fluidized bed combustion (FBC) and the low BTU gas concepts. Because the fluidized bed was selected as the best combustion concept, 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 defined as a baseline of comparison for potential direct fired FBC designs. Unsuccessful attempts were made to conceptually design a heater head compatible with the FBC for the scaled-up United Stirling U-4 engine. Alternate engine configurations suitable for direct firing in an FBC were developed and are now being studied and evaluated.

The following is a preliminary design analysis for an FBC direct fired heater head, along with a description of some tentative ideas regarding possible bed/heater/engine layouts.

TABLE 1
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 = $h_o = 50 \frac{BTU}{hrft^2 F} = 285 \frac{W}{m^2 C}$

Length of tubes = L

Inside diameter of tubes = D_i

Outside diameter of tubes = D_o

Number of tubes per 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 = 4.5×10^6 BTU/HR = 1.3 MW**

* 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.

** Preliminary estimate subject to refinement

Scaling Results

The following geometry was defined as being suitable for oil-firing the scaled-up U=4 design.

$$\begin{aligned} N &= 24 \\ L &= 33.5 \text{ in} \quad (850 \text{ mm}) \\ D_o &= .571 \text{ in} \quad (14.5 \text{ mm}) \\ D_i &= .394 \text{ in} \quad (10 \text{ mm}) \\ A_o &= 10.0 \text{ ft}^2 \quad (9293 \text{ cm}^2) \\ V &= 97.8 \text{ in}^3 \quad (1602 \text{ cm}^3) \end{aligned}$$

Heater Head Comparative Analysis

$$V = \frac{\pi D_i^2 L N}{4} \quad (1)$$

$$A_o = F \pi D_o L N \quad (2)$$

$$\Delta P = \text{Internal Pressure Drop} = 4f (L/D_i) \frac{G^2}{2\rho} \quad (3)$$

where f is the friction factor, ρ is the fluid density and G is the mass flow rate per unit flow area.

If K is some constant:

$$\Delta P \approx K \frac{L}{N^2 D_i^5} \quad (4)$$

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

Table 2 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 ΔP is nearly the same for all cases.

TABLE 2: Potential Direct Fired Heater Heads

DESIGN NAME	BASELINE CASE	1	2	3	4
		—	—	—	—
F	1	2.5	4.0	2.5	4.0
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)	61.4 (156)	38.4 (97.5)	46.9 (119)	32.2 (81.8)
N	24	28	28	44	40
Effective A_o ft ² (cm ²)	10.0 (9293)	70.3 (65300)	70.3 (65300)	70.3 (65300)	70.3 (65300)
$V/V_{\text{Baseline Case}}$	1.0	2.14	1.34	1.59	0.99
$\Delta P/\Delta P_{\text{Baseline Case}}$	1.0	1.35	0.84	1.39	1.15

Comments on Design

From Table 2 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. Further work is underway at AMTI to identify the limitations of finned tubes in fluidized beds.

Second, there is a trade-off between higher numbers of smaller diameter tubes and higher void volumes. In the past, United Stirling must have made a similar trade-off in their P-40 engine design, and therefore AMTI is looking for guidance from United Stirling on this problem.

Although not shown in the table, there is a trade-off between numbers of tubes and internal flow pressure drops. Currently, 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 tube length. Depending on as-yet-unidentified combustor dimensional constraints, longer tubes may be desirable in spite of higher void volume.

Fluidized Bed Constraints

The assumptions of a bed temperature of 1650°F (890°C) and heat transfer film coefficient on bare tubes of $50 \frac{\text{BTU}}{\text{hrft}^2 \text{F}}$ ($285 \frac{\text{W}}{\text{m}^2 \text{C}}$) are consistent with published values for an atmospheric pressure bed. These assumptions will have to be reviewed in detail. The question of tube finning needs to be addressed further, especially how the heat transfer coefficient decreases with added fins. The possibility of pressurizing the bed in order to increase the heat transfer coefficient and decrease the size of the bed needs to be addressed. Last but not least, there is a definite geometric design problem. The author understands that it is necessary to keep the length of every heater head tube the same in order to prevent flow non-uniformities. Furthermore, if the tubes are to be finned, then the tubes should be horizontally oriented

in the bed, with their fins perpendicular to the axis of the tube. In addition, in order to prevent flow channeling problems in the bed, the heater head tubes should be evenly spaced and symmetrically distributed throughout the bed.

Upon studying potential heater tube arrangements for the U-4 engine, it became obvious that the problem of maintaining a constant heater tube length with a reasonable header dead volume is difficult if the tubes are geometrically arranged to fit the bed. Conversely, the problem could be viewed from the other reference frame. In other words, 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 circular heater head design is not suitable because of flow channeling problems. Another design illustrated in Figure 2, shows a design which is geometrically compatible with the bed; however, it requires inordinate amounts of header volume.

Alternate Engine Configurations

Attempts have been made to generate engine module designs which are more amenable to direct firing in an FBC than the present U-4 arrangement. Figures 3, 4, and 5 illustrate a few of the resulting ideas. These configurations are currently being evaluated jointly by USS and AMTI.

Conclusions

The direct fired fluidized bed could turn out to be the best technical approach. It appears to have the lowest cost potential, and minimum safety hazard potential of any alternative. However, it may be the highest technical risk approach of all. If the technical risk appears to be too great for demonstration by 1985, then the direct fired fluidized bed may well be reported under task 1B, advanced combustion and heat transport systems.

FIGURE 2: DIRECT FIRED HEATER HEAD

NOTES:

1. Either vertical crank V-4 or U-4
2. Top and bottom cylinders have longer cold side connections than middle two

FIGURE 3: DIRECT FIRED WITH TWO REGENERATORS PER CYLINDER

NAME FIG. 4: DIRECT FIRED WITH
LONG DIMENSION HORIZONTAL

A ADVANCED MECHANICAL TECHNOLOGY, INC. 141 CALIFORNIA STREET • NEWTON, MASSACHUSETTS 02158		☐				
		☐				
UNLESS OTHERWISE SPECIFIED		DATE	REVISION	ECN	APPD	
X ¹ XX ² XXX ³ L ⁴ ✓		DATE 11-1-79	PROJECT NO.	JN		
MATERIAL		CHECKED	NEXT ASSY NO.			
NAME		APPROVED	SHEET NO. 1	NO OF SHEETS 1		
FIG 5: DIRECT FIRED WITH LONG DIMENSION VERTICAL		APPROVED	DRAWING NUMBER		REV	
		SCALE				

3.3.3 Gas Convection Loop

To date, it appears that the high pressure gas convection loop is technically feasible. However, it may turn out that the gas loop is the least attractive from a cost point of view because of two factors. First, the pumping power consumed may be significant with respect to engine shaft power. Secondly, in order to keep the pumping power and size within acceptable limits the system pressure may have to be rather high. This of course means the ducting and all heat exchanger tubes would be expensive at best and dangerous at worst.

Approach

An analysis was generated which predicts the convection loop pumping power 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. Some estimates were generated for pumping power as a function of the variables and some preliminary conclusions were drawn.

Analysis

The end result of the analysis was the following equation which gives the percent of engine shaft power consumed in pumping the convection loop gas through the heater head heat exchanger:

$$\% P_{HH} = \left(\frac{50}{\eta_p \eta_s} \right) \left(\frac{Pr^{2/3} \bar{R}^2}{M^2 C_p^3} \right) \left(\frac{f}{j} \right) (q^2 T^2) \left(\frac{1}{\Delta T_{lm} \Delta T_{cl}^2 A_x^2 p^2} \right) \quad (5)$$

where:

η_p = efficiency of pump

η_s = efficiency of Stirling engine

Pr = Prandtl number

$\bar{R}$ = universal gas constant

M = molecular weight of gas

C_p = specific heat of gas

f = friction factor

j = Chilton - Colburn factor = $\frac{h}{G C_p} \times Pr^{2/3}$

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

T = mean temperature of gas

ΔT_{lm} = log mean temperature drop 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 would probably be consumed in the heater head heat exchanger, some would also be required for the piping. 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)

$$\% P_{DUCT} = \frac{14.2}{\eta_p \eta_s} \left(\frac{\bar{R}^2 \mu^{0.2}}{M^2 C_p^{2.8}} \right) q^{1.8} T^2 \left(\frac{L}{D^{4.8} p^2 \Delta T_{cl}^{2.8}} \right) \quad (6)$$

where:

μ = absolute or dynamic viscosity

L = equivalent length of duct

D = hydraulic diameter of duct

Convection Loop Gas Selection

From equation 5, 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.

$$FM_1 = \frac{Pr^{2/3}}{M^2 C_p^3} \quad (7)$$

From equation 6, a similar figure of merit may be defined as follows:

$$FM_2 = \frac{\mu^{0.2}}{M^2 C_p^{2.8}} \quad (8)$$

The values of these figures of merit are given in Table 3 for several gases at 1500°F (815°C). Low values are desirable in order to minimize the pumping power.

The last two columns in Table 3 give the ratio of the figure of merit for each gas to the figure of merit for helium. Presuming that heater head pumping power is more significant than 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 as a loop using helium (all other design factors being equal). Similarly one using hydrogen would consume approximately one sixth to one seventh the power consumed in the same system using helium.

From the table, it may be seen that hydrogen is by far the best. However, because of safety considerations, we are not considering the possibility of using it in the convection loop. Carbon dioxide is the next best followed by water and helium. Water is not being seriously considered because of potential corrosion. 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. This means for any installation only one gas need be supplied. 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. Intuitively CO_2 seems safe, but this should be more carefully investigated if CO_2 is chosen over helium. For the time being, all calculations are being done assuming helium, however modifying the results for CO_2 is trivial.

TABLE 3: 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
H 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

Pumping Power and System Pressure

Equations 5 and 6 give the power requirement for the heater head and the ducting respectively. No similar work has yet been done for the combustion heat exchanger. Figures 6 and 7 were generated using the following values for the constants in equations 5 and 6.

$$\eta_p = .5$$

$$\eta_s = .4$$

Helium was the fluid

$$\frac{f}{j} = 3$$

$$q = 4.5 \times 10^6 \frac{\text{BTU}}{\text{hr}} = 1.3 \text{ MW}$$

T_{SUPPLY} is the temperature of the gas at the exit 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.

Figure 6 gives the pumping power (percent of shaft power) consumed in the heater head heat exchanger versus system pressure for various heater head flow areas and convection loop supply temperatures. Figure 7 gives the pumping power (percent of shaft power) consumed in the ducting versus duct hydraulic diameter for various pressures and supply temperatures. 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 6 shows the power consumed to pump the gas through the heater head versus system pressure for various supply temperatures and flow areas.

Because we wish to minimize heater head dead volume, we expect that this contribution to total pumping power will dominate the others. This figure points out several factors. If we wish to keep the pumping power low for any set of design conditions, increasing the pressure sufficiently will accomplish the desired results. Furthermore, in order to keep this power requirement below 1%, the range of interest for pressure is probably 200-500 psi (14-34 atm). From figure 7, at this pressure in order to keep ducting losses below 0.25%, the range of interest for duct diameter is 9-14 in (23-36 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 auxiliary power versus increased materials cost to arrive at minimum life-cycle cost. This optimum has not yet been determined.

Conclusions

The helium convective loop looks technically feasible. Further work needs to be done on the specifics of the heater head heat exchanger and the combustion heat exchanger. Cost estimates need to be generated for the system components. The trade-off between initial cost and parasitic power needs to be addressed and then life-cycle cost for the entire heat transport system needs to be assembled.

3.3.4 Sodium Heat Pipe

This section covers work carried out by Thermacore Incorporated as consultants to Advanced Mechanical Technology Incorporated. The work is in support of AMTI's contract with Argonne National Laboratory for the Design and Development of Stirling Engines for Stationary Power Generation Application in the 500-3000 horsepower range.

Initial discussions pointed out several integration problems. These include the need for temperature uniformity of all of the heat pipes feeding a single engine, the need for field serviceability of the heat pipes and/or engine heater head, and the minimum required spacing between the heater head tubes. Most of these problems are not inherent to the heat pipe but are interface requirements imposed upon the heat pipe by the Stirling engine.

The baseline heater head design is based on an engine which was designed to accommodate direct firing of liquid fuel. However, if the heat pipe is to be considered a viable system for the integration of a Fluidized Bed Combustor (FBC) and Stirling engine, a fresh look at a new heater-head design is required.

Technical Discussion

The initial evaluation and preliminary design of sodium heat pipes for integration of FBC's with Stirling engines is based on the specifications and requirements in Table 4. Although there is considerable overlapping in the several areas of heat pipe system design they can be generally categorized as thermal performance, materials, and cost.

Table 4

HEAT PIPE SYSTEM

SPECIFICATIONS AND REQUIREMENTS

A. Fluidized Bed

1. Size: 2 m x 2 m x required height
2. Controlled Bed Temperature: 890°C (1635°F)
3. Heat Transfer Rates: 2-5 W/cm² (6000-16000 BTU/hr-ft²)
4. Environment: erosive, corrosive, source of nascent hydrogen
5. Maximum Heat Exchanger Temperature: 890°C (1635°F)

B. Stirling Engine

1. Size: 2000 horsepower @ 40%
2. Type: twin crank 4 cylinder module - 16 cylinders
3. Thermal Power: 235 kW/cylinder (800 KBTUh/cylinder)
4. Bore: 18.24 cm (7.18 inches)
5. Head Flange Size: 50-60 cm (20-24 inches)

C. Heater-Head Tubes

1. Number: 18/cylinder
2. Length: 112 cm (44 inches)
3. Diameter: 1.27 cm (0.5 inches)
4. Wall Thickness: 0.127 cm (0.05 inches)
5. Minimum Spacing: 2.67 cm (1.05 inches) (one heat pipe/heater tube)
6. Temperature: 720°C (1325°F)

D. Implied Heat Pipe Requirements

1. Environmental: erosion, corrosion and hydrogen permeation protection
2. Maximum Temperature: 815°C (1500°F)
3. Allowable Heat Pipe ΔT : 55°C (100°F)
4. Minimum Temperature: 720°C (1325°F)
5. Thermal Power: 13 kW/heater tube (45 KBTUh)

heater tube, a certain degree of insensitivity is achieved. For example, should one pipe fail, the rest of the engine would be only slightly affected, and could continue to run. Thus, one heat pipe per heater tube appears to be the best design.

Additionally, if either the heat pipe and/or heater tube fails there must be a method by which the unit can be field repaired. Since the heat pipe is a vacuum sealed device, in most applications it could only be repaired in the factory. Thus, heat pipe failure can best be dealt with by designing a system which is composed of inexpensive components which are easy to replace. Again, one heat pipe per heater tube makes the most sense.

Conversely, this concept leaves open the possibility of heater head temperature non-uniformities due to maldistribution of heat input to the heat pipes. Additionally, it is not yet clear whether this would be a least cost approach. Nonetheless, in order to conceptually design a heat pipe system, and then prepare a cost estimate for the system, it was assumed that a geometric heater head design could be devised to allow for one heat pipe per heater tube.

Heat Pipe Design

The preliminary heat pipe design was established by determining the limiting factor in the system. This was found to be the FBC heat flux, which determined the heat pipe OD for a fixed bed length and total power per heat pipe. Then, based on the heat pipe diameter

In order to evaluate the thermal performance of a sodium heat pipe, the power throughput, physical size, and materials of construction must be defined. In order to make initial performance calculations, the envelope material was assumed to be of the general class: sodium compatible, heat resistant and super alloy.

From the specifications given in Table 4, several different heat pipe sizes and power ranges can be determined. If the need for temperature uniformity from cylinder to cylinder is stressed, then a network of interconnecting heat pipes which communicate at both the liquid and vapor ends could be envisioned. The total power of the network would be that of the entire engine, nominally 3,760 kW (12,800,000 BTU/hr). The wisdom of such a design is questionable since the failure of one component in the network, such as one heater tube, causes the entire system to fail. If possible, one would like a system which is insensitive to little problems rather than one which fails completely because of a single tube failure.

A system comprised of one heat pipe to supply the heat to a four cylinder module is another possible design, as is one heat pipe per cylinder. Each of these designs is also limited by the fact that a single tube failure shuts down one module or one cylinder and thus the entire engine.

On the contrary, if one heat pipe supplies heat to only one

and assumed length, vapor pressure drop was calculated and the capillary structure was specified to achieve the required capillary pressure.

Figure 8 shows the relationship between total heat transferred per heat pipe versus heat pipe OD (for a 2 meter wide FBC) for different FBC heat fluxes. The maximum heat flux obtainable in an FBC with the evaporator at about 750°C is approximately $5\text{W}/\text{cm}^2$, which for a 12 kW heat pipe implies a 1.66" OD evaporator ($1.66" = 1 \frac{1}{2}$ IPS).

From system considerations, a 1.8 meter (6') adiabatic section was chosen, along with a 44" developed length condenser, which has the heater tube penetrating it in two places. Thus a preliminary heat pipe envelope design was established as seen in Figure 9. More design details are given in Table 5.

To determine the capillary structure required for the heat pipe, it was assumed that operation was in the horizontal plane and numerous computer runs were made. The results of these calculations are seen in Figure 10 which shows the heat pipe capability versus heat pipe OD for a 2 meter evaporator. Also shown is the total heat pipe ΔT versus heat pipe diameter.

From Figure 10 it is seen that a 1.66" diameter heat pipe with four 100 mesh arteries (0.25" diameter) and two layers of 100 mesh

FIGURE 8: HEAT PIPE DIAMETER FOR VARIOUS EVAPORATOR RADIAL HEAT FLUXES

SECTION A-A

FIGURE 9

KEY	REVISION NOTE	INITIAL	DATE	TITLE 13 KW Sodium Heat Pipe			
				NAME		STD. TOLERANCES	
				DRAWN <i>ED</i>	DATE <i>11-21-79</i>	DECIMAL .xx" ± .015	
				APPROVED <i>USMC</i>	DATE <i>11-26-79</i>	FRACTION ± 1/64	
				SCALE	MATL	ANGULAR ± .003 PER INCH OF LENGTH	
				ASSY or BILL		DWG. NO.	
				THERMACORE, INC.		B-82-2

FIGURE 10: REQUIRED HEAT PIPE DIAMETER

Table 5

HEAT PIPE

SPECIFICATIONS

Evaporator

Length: 2 m (6.6 ft)

Diameter: 4.2 cm (1½ IPS - 1.66 in)

Wall: 2.8 mm (schedule 10 - 0.109 in)

Heat Flux: Radial - 5 W/cm^2 (16,000 BTU/hr ft²)

Adiabatic

Length: 1.8 m (6 ft)

Diameter: 4.2 cm (1½ IPS - 1.66 in)

Wall: 2.8 mm (schedule 10 - 0.109 in)

Configuration: Flex joint to accommodate vibration.
Bends to accommodate location in bed and on engine.

Heat Flux: Axial: 1 kW/cm^2 (3,171,000 BTU/hr ft²)

Condenser

Length: Developed length: 112 cm (44 in)

Diameter: 2.7 cm (3/4 IPS - 1.05 in)

Wall: 1.7 mm (Schedule 5 - 0.065 in)

Heat Flux: Radial: 30 W/cm^2 (92,000 BTU/hr ft²)

circumfential wick is almost capable of transferring the required power of 13 kW with a 30°C ΔT . This heat pipe is limited by the pressure drop within the vapor flow when operating at its maximum capability of 13 kW. Thus, if better performance is desired, either the heat pipe diameter must be increased or the capillary wick pressure increased.

To increase the capillary wick pressure, a finer mesh must be used to fabricate the arteries. Figure 10 also shows the performance of a 200 mesh arterial heat pipe. However, 200 mesh is more expensive than 100 mesh, and the permeability of the mesh is lower. Thus four layers are required for the circumfential wick in the evaporator, and thus costs will be increased.

Other types of heat pipes should be considered, such as gravity fed heat pipes with circumfential grooves for liquid distribution, or a hybrid with grooves and arteries. Although full heat pipe performance calculations have yet to be made for a gravity assisted grooved heat pipe, or the hybrid heat pipe, they were evaluated when the cost estimates were carried out.

The four 100 mesh artery heat pipe, as seen in Figure 9 was used as the baseline heat pipe. Table 6 details the heat pipe wick and performance ΔT . Additional heat pipe performance calculations

Table 6

WICK SPECIFICATIONS AND PERFORMANCE

Wick

Type: 100 mesh (0.0045" wire)

Capillary Radius: 0.0127 cm (0.005")

Radial Design: 2 layers

Axial Design: 4 arteries (5/16" OD x 1/4" ID)

Performance

Heat Transfer Rate: 13 kW (44,000 BTU/hr)

Evaporator Wall ΔT : 6.5°C (12°F)

Evaporator Wick ΔT : 0.5°C (1°F)

Vapor ΔT : 6.0°C (11°F)

Condenser Film ΔT : 1.7°C (3°F)

Condenser Wall ΔT : 15.0°C (27°F)

Total ΔT : 29.7°C (54°F)

will be carried out following the results of the heat pipe cost analysis and appropriate feed back from the FBC and engine preliminary design.

Materials Consideration

The choice of heat pipe envelope materials 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 hydrogen permeation barrier for the heat pipe (to prevent heat pipe failure due to hydrogen intrusion).

Table 7 shows these several materials considerations along with two suggested candidate heat pipe materials, their nominal cost, composition and strength at elevated temperatures. Since Incoloy 800 is also a candidate material for the heater-head tubes, it is a good choice. However, to date no hydrogen permeation data nor sodium compatibility data has been found. 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. 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 and did not fail.

Table 7

HEAT PIPE

MATERIAL 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 ($< \frac{1}{2}\%$) 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)

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 a 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.

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 protective coating. Upon removal of this protective layer hydrogen is allowed to permeate the heat pipe wall.

Thus it was concluded 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 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 permeating 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 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. Table 8 presents the physical properties of these several coatings and the thermal expansion of the two suggested heat pipe materials from which a comparison can be made.

From Table 8, one sees that 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. Likewise, 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

Table 8

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)			

investigated further for potential use with Incoloy 800 and/or 18 SR heat pipes in a FBC.

Cost Estimates

To establish the estimated cost of the sodium heat pipe as shown (Figure 9), a yearly production rate of 40-2000 horsepower engines was used. This results in 12,000 heat pipes per year with 4% excess for spares. This rate is 50 per day, which is well within the capability of a small production organization with 10 workers, 2 supervisors (first and second shift) both of which would spend some portion of their time on the product, one engineer, a manager and secretary.

This type of 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 processer.

Thermacore personnel have in earlier job positions been involved in the development and use of rotary process equipment. Recently, Thermacore under DOE contract DE-AC04-77-CS04099, Cost Effective Solar Collector using Heat Pipes, did a heat pipe cost analysis and preliminary design of a rotary processing unit. A similar rotary

processer could be designed for the sodium heat pipes for the FBC-Stirling engine.

To arrive at a heat pipe cost, a detailed cost analysis was carried out for several heat pipe types and the two suggested materials. These results are seen in Table 9 .

The variation in heat pipe costs for the same base metal is approximately \$100 and is easily traced to the cost of the screen wick materials and the sodium to fill it. Thus one concludes that the gravity fed covered grooved heat pipe should be investigated to the fullest extent if cost is to be minimized.

Based on the several heat pipe designs which were cost estimated, the following equation was developed:

$$\begin{array}{rclcl} \text{Heat Pipe} & & \text{Material} & & \\ & = & & \times & 2.91 - 0.19 \frac{\text{Material Cost}}{\text{lb}} \\ \text{Unit Cost} & & \text{Cost} & & \end{array}$$

Using the equation at \$4.75/lb. yields a heat pipe cost twice that of the materials cost, which is a number often quoted for the sake of cost estimating similar operations.

Table 9
INDIVIDUAL HEAT PIPE COST
BASED ON 12,000/YEAR

<u>Heat Pipe Type</u>	<u>18 SR</u>	<u>Incoloy 800</u>
4 Arteries/Radial Screen 100 Mesh	\$304.78	\$405.45
Grooves/4 Arteries 100 Mesh	\$250.91	\$351.57
Grooves/Cover Gravity Aiding	\$201.15	\$305.63

THIS PAGE
WAS INTENTIONALLY
LEFT BLANK

4.0 ENGINE CONCEPTUAL DESIGN

This section covers the work performed at USS on a conceptual design study for a large stationary Stirling engine for power generation in the 500 to 3000 hp size range.

Initial Selection

Based on our 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. 1000 - 3000 hp units could be obtained by incorporating 2 - 6 basic engine modules.

Since it has not yet been possible to work in detail with both direct and indirect heat transport systems within the short time at our disposal, and since no decision has been reached as to what type of direct or indirect heat transport system is going to be used, we have concentrated our design work in this study to an engine suited for a direct fired liquid fuel combustion system.

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 11). The multi-module engine could then be a three-crank V-configuration with the center crank common to the basic engine modules.

FIGURE 11: U-4 AUTOMOTIVE ENGINE

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, should that be employed as the indirect heat transport system, 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 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. Mean operating pressure was set at 15 MPa.

Table 10 gives the results of scaling which were subsequently used in the analysis.

Comments on Scaling

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

Table 10

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	850 mm
Number of regenerator 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 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

cold space) penalizes 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) penalizes 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, calls for longer cold connecting ducts, and creates larger manifolds and void volumes between heater and regenerators. A large number of 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.

Analysis

Performance calculations have been performed for the engine module, and the results are shown in Figures 12 and 13. There are no auxiliary losses included and the efficiency is based on the heat supplied to the heater tubes.

ASSUMPTIONS:

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

FIGURE 12: SHAFT POWER

ASSUMPTIONS:

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

FIGURE 13: EFFICIENCY

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.)

If 900 rpm is chosen, the engine module power will be 615 hp (gross) with helium at full pressure (15 MPa) and after reduction for auxiliaries this will be enough for the required 500 hp. At 900 rpm the efficiency increases from 39% to 45% when going from helium to hydrogen. 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 has been 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 14 and 15.

In order to show the importance of equal heat input to all four cycles in a double acting engine, a calculation has been performed where the heat input to one cycle is decreased and the resulting heater temperature is calculated. (See Figure 16). From Figure 16 it can be

ASSUMPTIONS:

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

FIGURE 14: INFLUENCE OF HEATER TEMPERATURE ON POWER AND EFFICIENCY

ASSUMPTIONS:

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

FIGURE 15: INFLUENCE OF COOLER TEMPERATURE ON POWER AND EFFICIENCY

FIGURE 16: COUPLING BETWEEN HEAT SUPPLY TO ONE CYCLE AND HEATER TEMPERATURE AND POWER OUTPUT

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.

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 17). 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 surface extensions (See Figure 18). 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

SECT. THRU PREHEATER

HEATER, TOP VIEW

FIGURE 17:

CRANKCASE, APPROX. ENVELOPE

FIGURE 18: EXTERNAL HEAT SYSTEM

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 crank case.

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.

Gas Cooler

The gas cooler is similar to the P75 design, i.e. small diameter tubes joined to circular end plates.

Stress Analysis

Cylinder and regenerator housings for the P40 and P75 engines have earlier been analyzed using a finite element stress calculation program. Based on that result an estimate of required thicknesses for this concept was made. We first eliminated materials containing high amount of strategic elements like cobalt. The choice is then between high-strength nickel-base materials or less high-strength iron-base materials.

Based on a heater temperature of 760°C which gives a pressure vessel temperature of around 720°C , and a required life of 30,000 hours with an allowable stress of $2/3 \times$ creep rupture stress, the choice was between Inconel 713 C with an allowable stress of 265 MPa and Climax Molybdenum XF-815 with an allowable stress of 100 MPa. (The cobalt-base HS-31 has an allowable stress of 120 MPa). Considering the large pressure vessel diameters and the big difference between two materials, it was felt that Inconel 713 C was the better choice.

Later discussions revealed that another nickel-base material Inconel 718 C is much easier to braze and weld than Inconel 713 C. But Inconel 718 C should preferably not be used at temperatures too much above 700°C .

It was then decided to decrease the heater tube temperature to 720°C (same as for P75) which would give a pressure vessel temperature of 675°C . With a required life of 30,000 hours the allowable stress will then be 250 MPa for Inconel 718 C.

With Inconel 718 C 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.

Cold Engine System Conceptual Design

Cylinder Block

For ease of manufacture the cylinder block is split up in four equal quadrants (See Figure 19). Also with one single cylinder block and four individual cross-head liners between the crank-case and the cylinder block, stresses could be imposed on the cross head liner flanges due to liner length tolerances if the four liners were going to be bolted between the crank case and the cylinder block.

The cylinder extends down into the cylinder block with a wet liner guided by the cross head liner (See Figure 20). The cooling water

FIGURE 19: CYLINDER BLOCK

FIGURE 20: COLD ENGINE SYSTEM

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. The cooling water exits into an outlet connecting two adjacent quadrants. In this way the cooling water out is collected at four points around the periphery of the cylinder block.

With one common cylinder block, 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 in quadrants it would be desirable to be able to remove one quadrant without interfering with the other. But this is not easy. A few possible alternatives have been looked into.

Alt A (Fig. 21): This is the alternative that is shown on the detailed drawings. In this case the adjacent heater quadrant has to be removed before the cylinder block quadrant with its heater can be removed. The loads from two of the cold connecting ducts must be taken by some simple connection between adjacent cylinder block quadrants.

Alt B (Fig. 22): In this alternative it is possible to remove one cylinder block quadrant with its heater, but the cylinder block must be of an odd shape and the loads

FIGURE 21: ALTERNATE A

FIGURE 22: ALTERNATE B

from all four cold connecting ducts must be taken between adjacent cylinder block quadrants.

Alt C (Fig. 23): In this case the loads from the cold connecting ducts are taken within the cylinder block quadrant, but adjacent heater quadrant must be removed before removing the cylinder block quadrant, which also here has received an odd shape.

Alt D (Fig. 24): 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. 25): This alternative has not an involute heater as in alternative A thru D. The heater tubes extend radially from the cylinder manifold to the regenerator housings. By necessity the gap between the heater tubes 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 taken care of.

The main drawback is the much larger void volume on the cold side as compared to the other alternatives,

FIGURE 23: ALTERNATE C

FIGURE 24: ALTERNATE D

FIGURE 25: ALTERNATE E

but 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 crank case and is loaded by only the net piston force. Disassembling the engine can be done by removing the rod pin when the cross head is in its upper position, unbolting the cross head liner from the crank case, leaving the connecting rod attached to the crank shaft.

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.

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. 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 matters 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 preventing crank case oil from migrating into the cycle where it could cause fouling up of the regenerator with decomposition products and contamination of the working gas. The proposed piston rod seal, a so called pumping Leningrader seal, has proved to be a very effective seal both when it comes to keeping the volume above the seal dry from oil and in sealing the working gas. More testing is, however, needed since the rod diameter is so much larger. 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.

External Heat System

With a direct fired liquid fuel combustion system the engine must incorporate an air preheater (See Figure 18).

With 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 should not exceed the 1,200 mm diameter. This means that the preheater matrix must be located above rather than around the heater head, but this was considered acceptable since the engine height is not as critical.

The preheater was scaled from the P75 preheater, which also works at 720°C heater head temperature. With 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 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

Alternative Arrangements

From the beginning it appeared most difficult if not unachievable to find a heater head configuration or an engine configuration that could accept any type of heat transport system. We have concentrated our design work in this study to an engine with a direct fired liquid fuel combustion system.

Time has been too short to consider in detail other direct or indirect heat transport systems such as fluidized bed combustion, either direct fired, or forced convection helium loop, or sodium heat pipe, but some comments will be made here.

Heat Pipe

Since the heat pipe must be a hermetically sealed unit with the condensor section comprising the heater, it becomes necessary to put in some kind of sealed partition wall between the hot condensor space and the cylinders and regenerator housings. Since these are fixed to the cold cylinder block while the partition wall is hot, some flexible connection, e.g. bellows, has to be introduced. With the large number of pressure vessels involved and with the large heater this will be rather difficult.

Fig. 26 shows an alternative arrangement. The four 120 mm dia.

FIGURE 26: ALTERNATE ARRANGEMENT

regenerators for each cylinder has been exchanged for twelve (12) 70 mm dia. regenerators located on a circle around each cylinder. The 350 mm distance between cylinder has been increased to 500 mm but the overall cylinder block dimensions are still the same.

With a direct fired liquid fuel combustion system 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 in order to seal off the combustion gases.

The same basic configuration could be used with a heat pipe by rearranging the heater tubes. A plate could be attached to the upper part of the cylinder (see Fig. 27) through which one end of hair pin heater tubes could extend down to the regenerators. The heat pipe may be branched at some point so that the plate could be welded to a circular heat pipe branch. Since cylinder top, plate, and heat pipe tube are all at the same temperature, no thermal stresses are imposed. However, the ineffective tube part at the regenerator and the conical plate must accept a difference in thermal growth between cylinder and regenerators.

A heat pipe engine could very well be placed up-side down, if this simplified the heat pipe design.

FIGURE 27: HEAT PIPE ALTERNATIVE

Direct Fired Fluidized Bed Combustor

If a direct fired fluidized bed combustor requires horizontal cylinders and parallel and horizontal heater tube rows, it could be difficult to come up with a design without some sacrifice in performance. The necessity of keeping the void volumes down has been pointed out earlier; it is also desirable to have as much symmetry as possible for good flow distribution in the heater. A design similar to the one discussed under Heat Pipe could possibly be achievable with some rearrangement of regenerators.