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Status of Free Piston Stirling Engine Driven Heat Pumps— Development, Issues, and Options

Final Report

Report Prepared by

ARTHUR D. LITTLE, INC.
Cambridge, Massachusetts 02140

under
Subcontract 86X-00205C

for

OAK RIDGE NATIONAL LABORATORY
operated by
MARTIN MARIETTA ENERGY SYSTEMS, INC.
for the

U.S. DEPARTMENT OF ENERGY

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ABSTRACT

This report summarizes the results of a program to review the recent experience with free piston Stirling engine (FPSE) driven heat pump systems and to identify key technical and cost issues which should be the focus of the future program efforts. The effort emphasizes experience with the programs undertaken by General Electric Company (inertial compressor concept) and Mechanical Technology, Inc. (hydraulically coupled system) since they have received the major funding by the Department of Energy and others since the late 1970's.

Issues addressed by the study include the status and prospects of FPSEs as heat pump drives, design and operational characteristics which might limit reliability and life, unique factors such as gas spring losses, influencing the efficiency of FPSE, and the cost constraints on the system to be economically competitive with conventional equipment and other heat actuated heat pump options. Special attention was given to the impacts of unlubricated seal and bearing options on system life, reliability, and cost.

Based on the above, a comprehensive program was outlined consisting of supporting analyses, supporting R&D, and system development which, if fully funded, would lead to field testing of a commercial prototype FPSE/HP system in about 5 years.

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I EXECUTIVE SUMMARY

1.0 Program Description and Objectives

Over the past 6 years the Department of Energy (DOE) has provided support for the development of Free Piston Stirling Engine driven heat pump systems (FPSE/HP). The primary incentive behind these programs was to develop a highly reliable, efficient, gas driven heat pump unit for residential applications which would help conserve natural gas used in space heating, provide more even annual usage of natural gas by using it to provide air conditioning, and reduce electric utility peak demand problems in the summer. Despite the expenditure of significant resources by DOE and others (GRI, private companies, etc.) the FPSE/HP has not evolved toward a commercial product at the pace originally projected by its developers.

The FPSE/HP is being developed in parallel with other options for gas fired heat pump technologies including I.C. engine driven units and absorption units. Also the situation relative to energy costs and the efficiency of "conventional" technologies is in a state of flux and changing perceptions. For example, advanced gas fired furnaces have efficiencies of over 90% as compared to only 60-75% when many of the gas fired heat pump programs were initiated. The competitive environment for all gas fired heat pump options is, therefore, more demanding than heretofore.

Given the uncertain technical status for the FPSE/HP and changes in the external environment, the DOE determined that the situation should be reviewed and updated as part of its ongoing program planning activities.

This report summarizes the result of a 12 month program initiated during May 1984 which had as its primary objectives to assess the technical status of free piston Stirling engine driven heat pumps (FPSE/HP) technology and identify programmatic approaches for resolving remaining technical and economic issues which represent barriers to their commercial development.

The general approach taken was to reviews the literature base which exist in the FPSE field and to visit those organizations with relevant R&D programs. This information and experience was supplemented by that from related fields, such as high reliability spaceborne free piston refrigeration equipment, to make judgements on technical/economic issues and developing appropriate R&D program options.

2.0 Technology Status and Issues

In the long term, the original incentives to develop a cost effective gas fired heat pump technology remain valid, and the FPSE/HP concept still remains one of the more promising options for lower capacity residential or light commercial applications. Several of the key requirements for a successful system have been demonstrated as the result of past and ongoing programs including engine efficiencies approaching 30%. Recently*, significant progress has been made in demonstrating good life potential as the result of endurance testing in support of space power system applications which have similarly stringent life and reliability requirements.

* Late 1984-85

Notwithstanding, the above mentioned recent progress, integrated FPSE/HP systems have not demonstrated many of the essential features required for commercial viability including; high seasonal coefficient of performance, long life, and low maintenance.

Several of the issues associated with FPSE/HP which have not been fully addressed to date include:

- A reliable low cost, and efficient means of coupling a FPSE to a heat pump cycle.
- Analytical models which can provide a reasonable level of accuracy for projecting the performance of current configurations and new configurations which might be considered.
- Accurate assessments of loss mechanisms associated with bearings, gas springs, and flow through heat exchangers, and identification of design options for minimizing such losses.
- Development of subcomponents (seals, bearings, combustors, etc.) which meet the stringent combinations of technical/economic requirements of gas fired heat pump systems.

Even with the remaining technical/economic issues associated with FPSE/HP, this option remains one of the more promising for residential size applications. This is due primarily to the potential for mechanical simplicity which could lead to low maintenance operation which is particularly critical for small capacity applications. For example, FPSE/HP have no valves, mechanical cranks, or lubrication requirements which are primary maintenance problems with I.C. engines. However, as indicated in Section II the potential for mechanical simplicity, for a FPSE/HP has not yet been fully reflected in current designs.

It is, therefore, important that future program initiatives in this technology review overall design approaches and emphasize life, reliability, and maintenance issues which are critical to the commercial success of this option.

3.0 Program options

The FPSE/HP programs were initiated on the assumption that they were commercial system development programs starting from a sound base of technology, and leading in a predictable manner to early field tests. These programs, therefore, did not have the normal structure of a system R&D program which typically includes extensive supporting analyses, component development, and supporting R&D functions as well as system development. To some extent the lingering issues on rather basic questions of system configuration, life, costs, analytical tools, and component designs are the result of the initial focus on quickly initiating system fabrication and testing functions.

Notwithstanding the above observation, previous programs have formed a strong base for continuing with the development of a FPSE/HP system. As indicated in section IV, such a program would have three primary and mutually supporting components:

- Analytical Support
- Supporting R&D and Component Development
- System Development

The pace of the program would be determined in large part by the availability of financial resources.

The Baseline Program plan described in Section IV indicates that the time required to develop a system to the state of a field test prototype is estimated to be about 4 - 5 years if adequate financial resources are not overly constrained. Associated budgeting estimates indicate financial requirements during this period of about $20 million. This is comparable to the funding for FPSE developments programs by DOE and others (most importantly, GRI) since the late 1970's. The Baseline Program assumes that two systems are developed in parallel so that alternative design strategies can be evaluated (engine-heat pump cycle coupling, bearing & seals, etc.)

The program outlined includes an extensive configuration review process to identify which design options show the best promise of meeting the combination of performance and cost goals. This review will, also, help ensure that the substantial new system development effort proceeds only after systems analyses and preliminary designs indicate a reasonable chance for success. It should be noted that ongoing testing of FPSE system and components developed as part of current DOE, GRI, and NASA programs will be providing experimental data which will be important inputs into the review process.

II TECHNOLOGY STATUS

1.0 OVERVIEW

1.1 Approach

Figure 1.1 indicates the tasks associated with the FPSE/Heat Pump (FPSE/HP) Assessment Program, discussed in this report. This section summarizes the results of the Task 2 effort to review the status of FPSE/HP technology and to identify major technical and cost issues which must be successfully addressed to result in a system with commercial potential.

The approach pursued in undertaking Task 2 was to review the literature base which exists on FPSE systems and to meet with those organizations with significant activities in this field. Table 1.1 lists the names of organizations visited during the Task 2 effort. In addition, the Arthur D. Little Program Manager attended the Stirling engine sessions at the (August 20 to 24, 1984) IECEC held in San Francisco. These sessions included an informal evening session which focussed on FPSE developments and their applications as heat pump drives. Opinions expressed by a broad base of the Stirling engine community at this session are summarized in Appendix 1.0.

1.2 Application Requirements

The review of the status and prospects for FPSE/Heat Pumps must take into account technical performance and cost constraints imposed by the economics of the application. The primary constraint is that the FPSE/HP system be economically competitive with other options which can be realistically considered for the same application. These include:

- Gas furnace/electric air conditioning system.
- Electric heat pumps.
- Other engine driven heat pump options (I.C. engines, kinematic Stirling, etc.)
- Absorption units.

Analysis contained in several reports indicate that this broadly stated constraint results in the approximate technical and cost specifications of Table 1.2. Two key requirements, long life and high efficiency, of this application are discussed briefly below.

Life and Maintenance

The "system" must have a useful operating life of 10-15 years which implies, at least, 25,000 hours of operation and maybe as high as 50,000 hours. This statement is, however, not without its ambiguities since it does not exclude replacing parts which are subject to wear or deterioration. For example, burner assembly parts which are readily accessible for maintenance could, in principle, be periodically replaced.

By contrast, piston rings contained in a hermetically sealed enclosure would probably not be subject to field service and would have to last the life of the system.

Figure 1.1

PROGRAM TASKS

ASSESSMENT OF FREE-PISTON STIRLING ENGINE DRIVEN HEAT PUMP TECHNOLOGY

Task 1.0	MILESTONES AND COST SCHEDULE
Task 2.0	IDENTIFY AND CHARACTERIZE PRINCIPAL DEVELOPMENT PROBLEMS
Task 3.0	ASSESS DEVELOPMENT PROBLEMS AND FORMULATE APPROACHES TO THEIR SOLUTION
Task 4.0	DEFINE R&D PROGRAM OPTIONS
Task 5.0	FINAL REPORT PREPARATION

Table 1.1

ORGANIZATIONS CONTACTED

Mechanical Technology, Inc.
968 Albany-Shaker Road
Latham, NY 12110

General Electric Company
Valley Forge
P. O. Box 8661
Philadelphia, PA 19101

Gas Research Institute
8600 West Bryn Mawr Avenue
Suite 1100
Chicago, IL 60631

Argonne National Laboratories
9700 South Cass Avenue
Argonne, IL 60439

Martini Engineering
2303 Harris
Richland, WA 907352

SUNPOWER, Inc.
6 Byard Street
Athens, OH 45701

Energy Research and Generation, Inc. (ERG)
Lowell and 57th Streets
Oakland, CA 94608

NASA-Lewis Research Center
21000 Brookpark Road
Cleveland, OH 44135

Stirling Power Systems
P. O. Box 1187
Ann Arbor, MI 48106

Joint Center for Graduate Study
University of Washington
100 Sprout Road
Richland, WA 99352

Oak Ridge National Laboratory
Oak Ridge, TN 37831

The life requirements for this application could be the most difficult goal for any engine driven system to achieve and impacts significantly on the design flexibility for FPSE options.

A related issue to long life is high reliability leading to low maintenance. The maintenance issue is particularly critical for residential size heat pump applications due to the low absolute value of annual energy savings (albeit potentially high in percentage terms). Both economics and consumer preferences indicate that, at most, two maintenance calls per year might be acceptable and possibly only one. This implies that the FPSE/HP must operate 1,200 to 2,500 hours between scheduled maintenance visits.

By comparison, even heavy duty cycle internal combustion engines normally have scheduled maintenance intervals of 500 to 750 hours.

Engine Efficiency

A primary rationale for any gas driven heat pump unit is that annual energy costs for such a system will be significantly lower than for electric or gas driven alternatives. For a kinematic engine this usually implies that brake gross thermal efficiency (2_{bgt}) be in excess of 30 percent where efficiency is defined as:

$$2_{bgt} = \frac{\text{engine brake power output}}{\text{energy content of fuel at HHV + engine aid power at source efficiency}^{(1)}}$$

At this engine efficiency threshold the overall heating COP of an engine driven heat pump is 2.0 - 2.5 of design conditions (47^{o} F ambient) which results in significant cost savings as compared to a high efficiency furnace system. Significantly lower performance than this level will usually result in insufficient energy savings to make the system economically attractive.

As indicated in Section 4, for a FPSE this definition of efficiency is complicated by the lack of a shaft and the resultant close integration of the engine output with the compressor system.

1.3 Rationale for FPSE/Heat Pump

There are a number of engine drives which are being or have been considered for heat pump drives including kinematic Stirling, I.C. engines, and Brayton cycle engines. The FPSE is perceived as having potential advantages over the other options in several significant areas. The rationales put forth to justify these potential advantages are discussed briefly below.

1.3.1 Life and Reliability

All Stirling engines have the advantage of not requiring valves or subjecting the working space and lubrication systems[(2)] to hot combustion gases. These characteristics would tend to improve the life and reliability characteristics of this engine class. In addition, the FPSE eliminates the

---

(1) For most kinematic engines, engine aid power will be zero since accessories are engine driven.

(2) In Kinematic Stirling engines only.

Table 1.2

MAJOR PERFORMANCE REQUIREMENTS OF ENGINES FOR USE AS HEAT PUMP DRIVES

Efficiency*	30%+
Life	15,000 - 50,000 hours
Maintenance required	Once per year - minimal parts replacement
Emissions	Comparable to conventional HVAC equipment
Low Noise and Vibration	
Cost	$300-$400 per kilowatt (max)

*Brake gross thermal efficiency based on fuel input (at the higher heating value) and the equivalent of shaft power for a kinematic engine.

need for hermetic piston rods and a highly loaded crankcase mechanical drive with associated bearings and seals. In fact, in its simplest configuration the FPSE has only two moving parts, a displacer and a power piston. The superficial mechanical simplicity of the FPSE is used as the primary basis of the argument for very long life and good reliability. This argument can be further reinforced for those configurations using gas bearing and clearance seals in the engine and metal bellows or diaphragm seals on the outlet "shaft" thereby (ideally) resulting in no rubbing surfaces to contribute to wear. Such nonrubbing seals have, in fact, demonstrated good life potential when used in specialized refrigeration equipment being developed for space use and in small heart pumps.

The potential for life and reliability has been demonstrated in certain specialty applications but not yet in FPSE/Heat Pumps. As indicated in Section 3, the apparent reasons for this are complex and include a combustion of bearing and seal wear, high temperature, cyclic operation, and the means for extracting power from the FPSE concepts in the absence of a crank mechanism.

As a result of these factors, the ability of the FPSE now under development to meet the stringent life and reliability requirement of the heat pump application still has not been established.

1.3.2 High Efficiency

Ideally, Stirling engines have very high efficiency potential when working at temperature levels attainable with metallic alloys (≈500-800^{o}C). Kinematic Stirling engines have confirmed this efficiency potential with engine efficiencies as high as 35 percent measured on the automotive program. FPSE eliminate the losses associated with the tight shaft seals and other mechanical losses in the kinematic transmission. As a result FPSE proponents project that FPSE should have, at least, as high an efficiency as kinematic engines and possibly even higher.

To date, this efficiency potential has not been consistently demonstrated in an integrated FPSE/HP system. Again, reasons for this are complex and, include some combination of lack of control of dynamic motion, higher than expected losses in gas springs, seals, and bearings, and high losses in the means for coupling the FPSE to the heat pump cycles.

Further complicating the issue is the fact that existing analytical models have tended to overestimate FPSE performance - possibly due, in part, to a lack of quantitative understanding of fundamental loss mechanisms.

Notwithstanding the above issues, progress has been made in improving FPSE performance levels as exemplified by recent preliminary (early 1985) results from MTI which indicate efficiency levels approaching 30% in test engines. There is reason for optimism, therefore, that properly designed and operated FPSE can achieve necessary efficiency goals, at least, under steady state conditions. The efficiency requirements of heat pump applications are, however, particularly stringent and, therefore, further analytical and experimental R&D is needed to better clarify this critical issue.

1.3.3 Low Cost Potential

FPSE proponents emphasize that a FPSE/heat pump system can be fully mechanically integrated with no need for separate engine and compressor drive trains with associated connecting rods, crank rods, crankcases, bearings, and shaft seals. An example of the resultant simplicity is the Stirling duplex cycle which was only three moving parts.

On a qualitative basis the trends torward cost reductions due to the above are probably valid, although questions remain relative to the eventual cost structure of a practical FPSE/HP due to uncertainties in:

- o The cost of the power extraction system (as compared, for example, to a conventional, mass produced, heat pump unit driven by a kinematic engine).
- o The high tolerances and extreme care to cleanliness required by gas bearing and clearance seals, if these are used.
- o The need for very effective high temperature and long lived heat transfer components (air preheaters, regenerators, heater heads) to achieve efficiency goals.

It should be noted that the cost implications of highly effective and durable hot end components also pertain to KSE.

1.3.4 Low Noise and Vibration Levels

One potential advantage of all Stirling engines is low noise levels due to a lack of valves, and relatively smooth operation (as compared to I.C. engines). This advantage may be heightened for FPSE by eliminating the mechanical components associated with a kinematic drive shaft. Experience with selected FPSE tends to verify the low mechanical noise potential.

The primary source of noise from a FPSE/HP is likely to be associated with the high intensity combustion system. Little work has been done relative to assessing noise levels from this source and, therefore, its impact on system acceptability is still uncertain.

Most kinematic Stirling engines can be mechanically balanced resulting in low vibration levels. FPSE are, on the other hand, inherently unbalanced unless some form of opposed cylinder arrangement is utilized* (which further increases costs). In principle, this unbalance can be isolated by using an appropriately designed spring mounting system. Isolation will be facilitied by the fact that the engine operates at a fixed frequency. The implications of this on overall system design have, however, only been superficially explored.

As indicated above, the rationales put forth for the FPSE/HP system have only in part been verified by experimental and analytical work to date. The following sections focus on examining the issues and uncertainties associated with these rationales in order to identify key R&D activities which would most effectively result in their resolution.

---

*This has, in part, been done on the MTI system where a counterweight is used in the hydraulic transmission to balance out the motion of the compressor piston assembly and more recently on the RE-1000 test engine where a resonant damper has been successfully installed on the NASA-Lewis machine by Sunpower.

2.0 SYSTEM DESCRIPTIONS

There are at least 10 design options which have been considered for FPSE/HP systems. The following briefly describes four of the configurations which have received the most attention in government and industry sponsored programs. The diagrams are simplified to identify more clearly, the bearings, seals, and gas springs which are essential to system operation.

2.1 Inertial Compressor (G.E.)

Figure 2.1 is a simplified schematic of the G.E. concept utilizing an inertial compressor. Key features of this system include:

- The free piston double acting freon compressor is contained within the power piston and moves relative to the power piston during system operation. As a result, there are three degrees of freedom* (displacer piston, power piston, compressor piston and casing) which greatly complicates the system dynamics.
- Sliding bearings and seals are utilized on the power piston and displacer. As indicated on the schematic there are 6 sliding seals and bearings within the engine envelope. All seals and bearings operate unlubricated in a dry helium environment.
- The connection of the compressor housing (power piston) which is oscillating with an amplitude of up to 1.7 inches to the stationary heat pump housing is via flexible serpentine tubes which are flexed at engine frequency (30 Hz).
- The power piston assembly weighs over 17 lbs and represents a significant unbalanced load. This complicates achieving acceptable vibration and noise levels.

The system developed to date is designed to operate at 30 Hz with a helium working gas average pressure of 870 psi. In its present configuration, the motion is controlled only by internal dynamics of the 3 independently moving pistons. For reasons that are still not fully understood, the system natural frequency was lower (25 to 27 Hz) than the design value of 30 Hz with detrimental capacity and efficiency ramifications. An attempt was made to rectify this problem by adding a "second gas spring" for the power piston,thereby, increasing overall gas spring stiffness. The addition of this spring did not have the anticipated results and appeared to, if anything, reduce performance by virtue of increasing hysteresis losses. The disappointment with the results of adding the second gas spring suggests that this complex, dynamic system, combined with the fluid flow issues associated with interactions of the engine and compressor, were poorly understood, making analytical projections of performance particularly difficult.

Several methods of control for this system were considered during its development. In its most recent configuration, the system did not use any means of active control to determine the magnitude of piston motions, i.e., basically unconstrained motion. The system was operated at two firing levels as indicated by Table 2.1. The system output is nearly constant for

* The movement of the housing due to a basic dynamic imbalance in the system can add a fourth degree of freedom

Table 2.1

FIRING RATES FOR G.E. BURNER - PROTO 2

A. Firing Rates:

Ambient Air Temperature Over 85_F:
Q_B = 41,000 Btu/hr

Ambient Air Temperature Between 30_F and 85_F:
Q_B = 24,400 Btu/hr

Ambient Air Temperature Less Than 30_F:
Q_B = 41,000 Btu/hr

B. Other Characteristics:

Efficiency (LHV)	- 82%
Pressure Drop	- 2 in H_2O
Heater Head Heat Exchange Effectiveness	- 92%
Preheater Effectiveness	- 60%

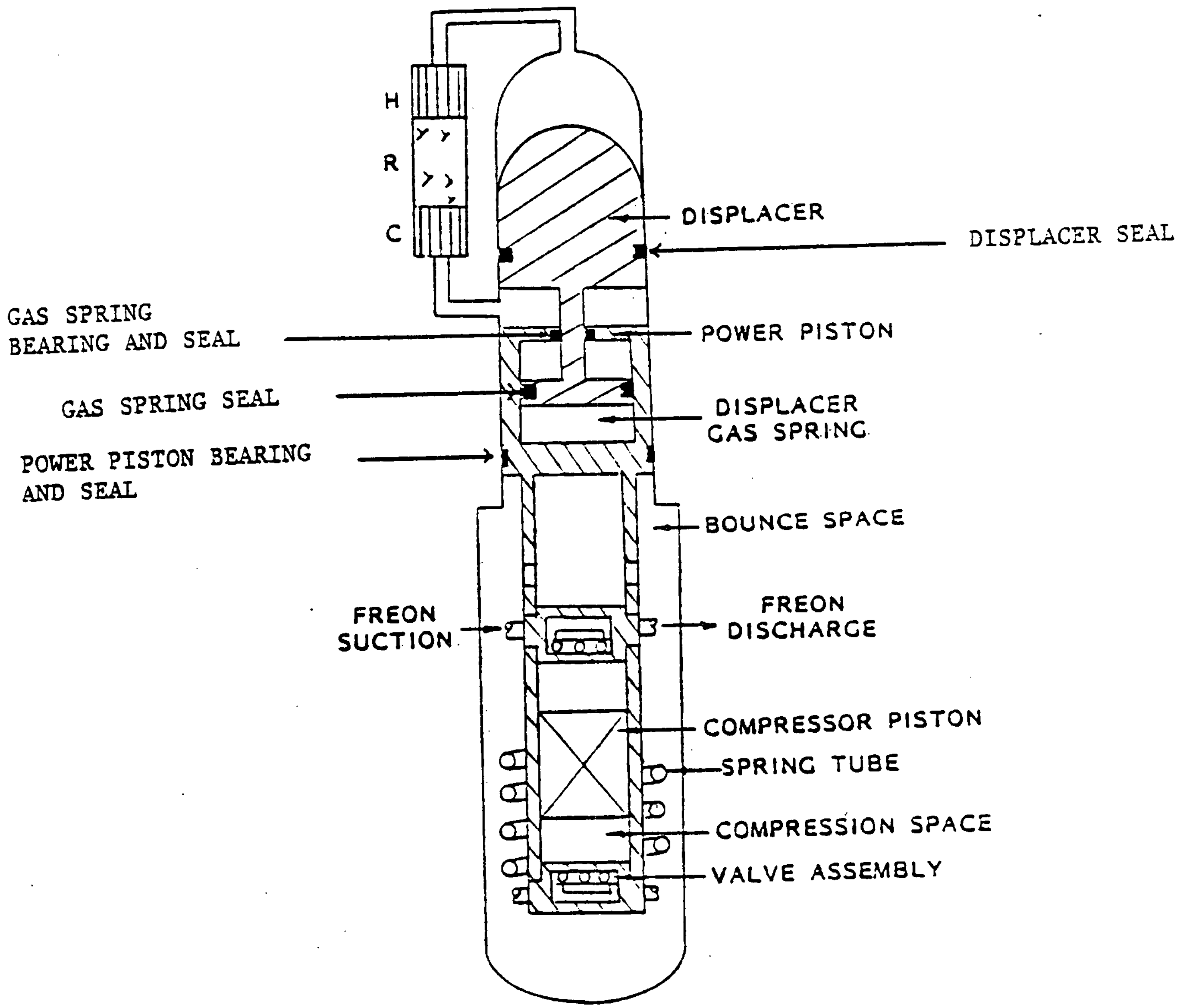

Figure 2.1

FREE PISTON STIRLING ENGINE WITH FREE PISTON LINEAR INERTIA COMPRESSOR a fixed firing level independent of compressor operating conditions, i.e., ambient air temperature. This is due to the fact that as ambient air temperature decreases (in heat pump mode) the compressor stroke will naturally increase since pressure levels in the compressor decreases. As a result, system output (and, therefore, burner heat demand) is nearly independent of ambient air conditions over a fairly wide range of operating conditions.

Unfortunately, at very low ambient air temperatures, the freon spring stiffness can be so low that the motion at the compressor piston becomes difficult to control and piston motion becomes excessive (possibly mechanically damaging). Also, the natural frequency of the compressor changes as freon gas pressure changes so that the system is not fully "tuned" under all operating conditions. In latter phases of the program, a system was developed to re-reference the compressor gas spring to a pressure level intermediate between suction and discharge freon pressure levels at low ambient air temperatures. Little testing was done on this system - however, if properly executed, it should be able to resolve excess stroke problems which otherwise occur at low suction pressures.

Notwithstanding the potential benefits of the variably referenced compressor gas springs, it is still unclear if control strategies can be implemented which result in high efficiency and stable operation over the full range of operating conditions.

2.2 Hydraulic Drive (MTI)

Figure 2.2 is a simplified schematic of the system under development at MTI using a diaphragm actuated hydraulic transmission system to transfer energy out of the Stirling engine.

In this system the "power piston" is incorporated into the mass of the hydraulic transmission. The restoring force is provided by the gas spring shown at the rear of the hydraulic transmission assembly. During operation an increase in pressure in the engine due to the displacer shuttling gases to the hot space results in downward movement of the diaphragm. Via a hydraulic mechanism, this movement is transformed into the movement of the refrigerant compressor assembly in a horizontal direction. A counterweight is provided in the hydraulic transmission which moves in the opposite direction of the compressor piston thereby providing dynamic balance for the power transfer assembly. This arrangement requires numerous seals both within the hydraulic system and between the high pressure (800 psi) hydraulic fluid and the low pressure freon contained in the compressor cylinder. The resultant friction and viscous fluid losses can be significant with present goals being about a 70 percent transmission efficiency at maximum load with lower values at other, more typical, load conditions. However, due to higher than anticipated losses, even these modest efficiency goals have not been achieved to date in operational hardware (including the counterweight).*

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* More recently (1985), an external counterweight has been added so that the internal counterweight can be removed. This should significantly reduce losses in the hydraulic transmission. As at the time of preparing this report this new arrangement had not been tested.

FPSE HEAT PUMP POWER MODULE

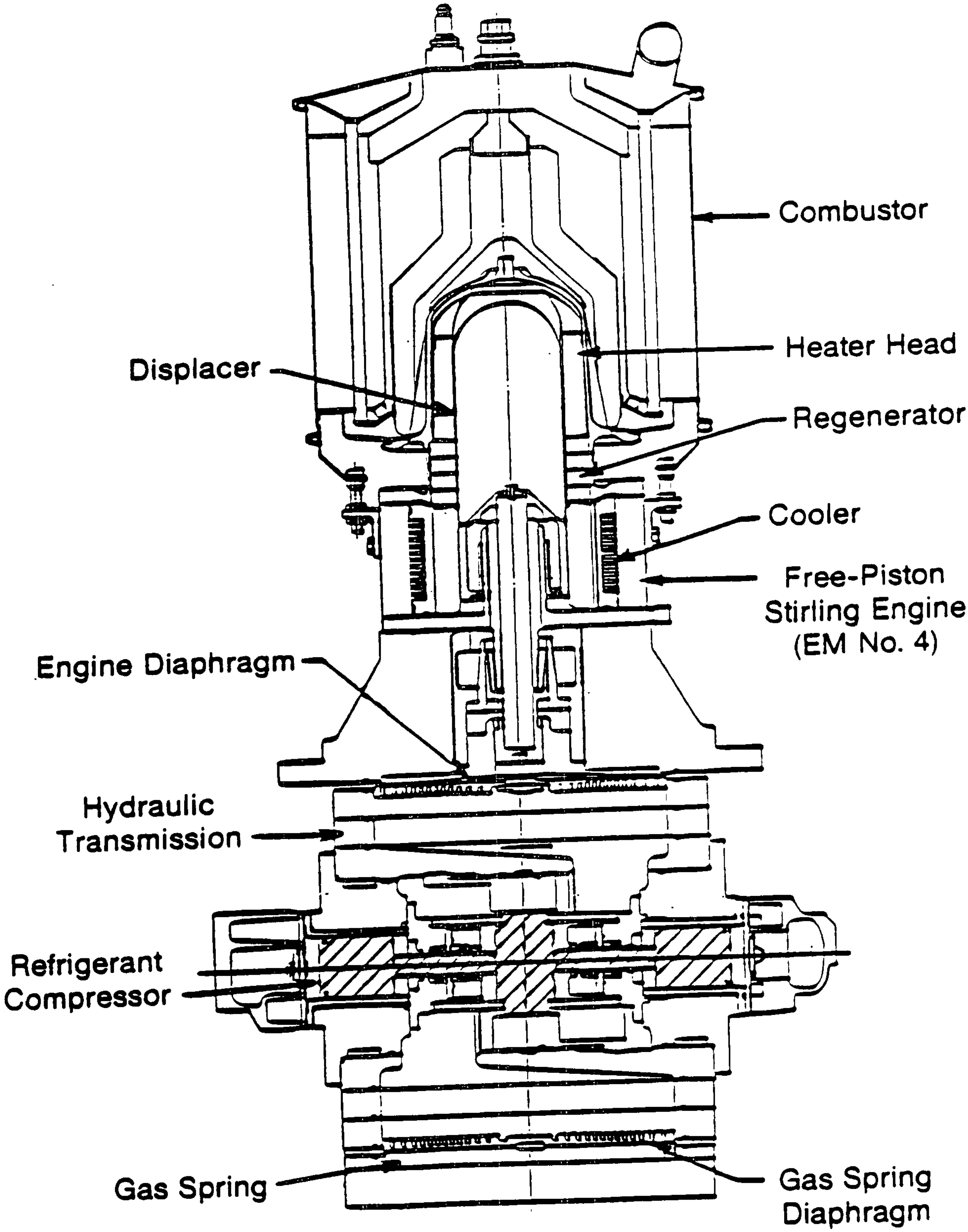

Figure 2.2

SCHEMATIC OF MTI HAHP

As indicated above, in this arrangement, the "power piston" is incorporated into the diaphragm coupled hydraulically actuated compressor assembly. MTI has elected to divide the engine/heat pump into two assemblies: the "engine," the boundary of which is at the diaphragm and a compressor assembly which includes the diaphragms, hydraulic transmission, and lower gas spring. This report continues with these definitions since they correspond to physically identifiable system subassemblies having measurable input/output characteristics. With these definitions, the "engine" has no power piston contained in the helium envelope - only a displacer piston with associated seals, bearings, and gas springs. In the MTI system, clearance seals and hydrostatic bearings are utilized on the displacer. If properly designed and fabricated, this configuration should eliminate wear. The hydrostatic bearings, in turn, require extracting high pressure helium from the engine at pressure levels above the mean. The means for doing this could include a pneumatic arrangement with check valves and buffer volumes. The mechanical and system efficiency implications of such arrangements have not been well defined at this time.

Capacity control is achieved by electromagnetically driving the displacer assembly at or near its natural frequency.* The stroke and phase of the displacer can, thereby, be controlled which, in turn, controls the output of the engine.

The operating mode now used results in variable displacer stroke depending on ambient air conditions. Maximum stroke is 0.75 inches at an ambient air temperature of 17° F and minimum stroke is 0.5 inches at an ambient air temperature of 47° F. The advantages of this control approach includes:

- The transmission losses under part stroke condition are much less than at full stroke.
- The capacity of the system does not vary rapidly with ambient air conditions since freon mass throughput is stablized by increasing stroke as suction pressures decline. This, in principle, allows the heat pump to supply all the heating needs even at low ambient air temperatures (unlike electric heat pumps which require back up resistance heaters).

Concurrently, with stroke variations, burner firing rates are modulated to achieve constant heater head temperature.

2.3 Duplex Stirling Cycle

Figure 2.3 shows the schematic of a duplex Stirling engine/heat pump similar to that investigated by SUNPOWER, Inc., as part of a program with GRI. In this arrangement the Stirling engine power piston is also the driving piston of the Stirling heat pump cycle. The difference in displacer piston diameters between the engine and heat pump sides of the cycle reflect the large differences in _T through which the two cycles operate (i.e., much more gas must be moved for a given energy flow on the heat pump side of the cycle.

This arrangement has the inherent advantage of using helium on both sides of the cycle, thereby facilitating the coupling of the engine with the heat

* A system similar to one proposed and patented by W. Martini.

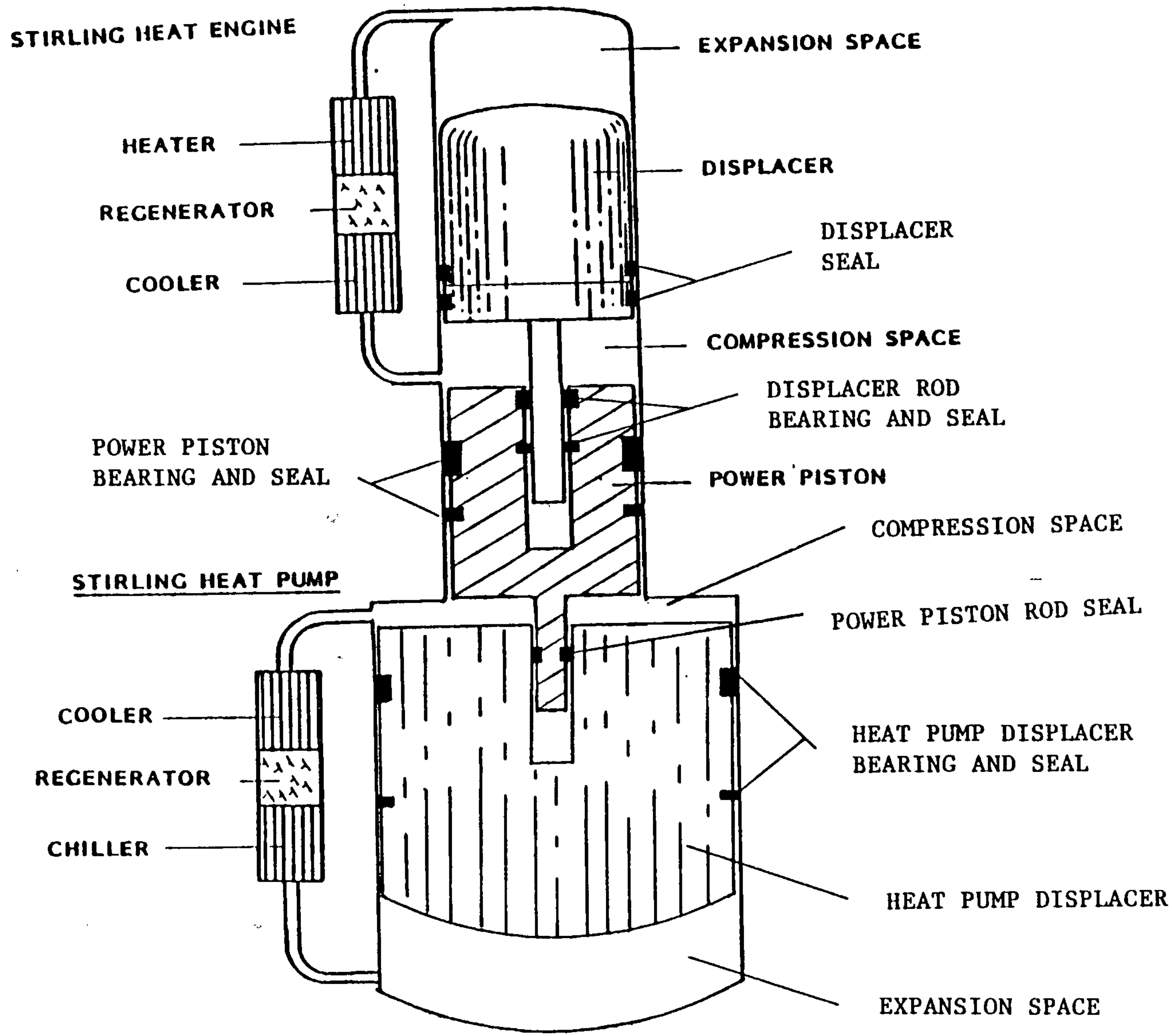

Figure 2.3 FREE PISTON DUPLEX STIRLING HEAT PUMP CONCEPT

pump. The resultant transmission efficiency of power is very high and could approach 90 percent with good power piston seals and low loss gas springs. One disadvantage of this approach is, however, that due to the low _T's the efficiency of the Stirling heat pump cycle could be unacceptably low unless large areas of heat exchangers are used to reduce gas to heat exchanger _T's to very low levels. However, SUNPOWER tests indicate that heat pump efficiencies approaching 50 percent of Carnot were achieved - a level which at least approaches that achieved by vapor compression equipment operating in that temperature range (ΔT's typically at 100° F).

As indicated in Figure 2.3, the duplex Stirling engine requires seals and bearings for the displacer pistons, power pistons, and gas spring rods. The impact of bearings and seals on system life and reliability are, therefore, not avoided by using this concept. It does avoid the need for continuous sealing, however (even when the system is not operating), since the only pressure differentials are those due to moving gas through heat exchangers and temperature differentials between working volumes. The use of a single working fluid (helium) also reduces the practical impacts of leakage that does occur (i.e., no equivalent of a shaft seal resulting in absolute loss of working gas).

The duplex Stirling cycle has the characteristic that the dynamics of the heat pump cycle are virtually unaffected by ambient air temperature levels (unlike vapor compression cycles), since helium working gas pressures are only weakly linked to heat exchange temperature levels. This should greatly simplify system control. For example, the output of the engine side of the cycle could be controlled by a driver (Martini) displacer such as the one used in the MTI unit.

2.4 Stirling Generator/Electric Heat Pump

Most concepts for using a FPSE to operate a heat pump require developing unique and, often, quite complex new freon compressor systems (or a new Stirling heat pump cycle equipment in the case of a duplex Stirling system). This can be avoided by using a FPSE driven linear alternator system (Figure 2.4) to drive a conventional heat pump unit. One disadvantage of this approach is the obvious losses and costs associated with the alternator/electric motor transmission. Also, the starting current of electrically driven compressors can be 2-4 times the steady state current, thereby complicating the design of the FPSE - linear alternator system. However, the recent development of high efficiency linear alternators (85-90 percent) may make this option of increasing interest.

In this arrangement, the motion of the power piston is used to operate a linear alternator. Most FPSE generators will require bearings, seals, and gas springs on both the displacer and power pistons as indicated in Figure 2.4. In principle these can be either sliding surfaces (per SUNPOWER) or gas bearing and clearance seals (MTI) or diaphragms/bellows.* The life and reliability issues associated with bearings and seals, therefore, are not

*NOTE: HoMach Systems Ltd. has recently introduced a small (80-150 watt) linear alternator system using a diaphragm seal based on technology developed by the group at Harwell, England.

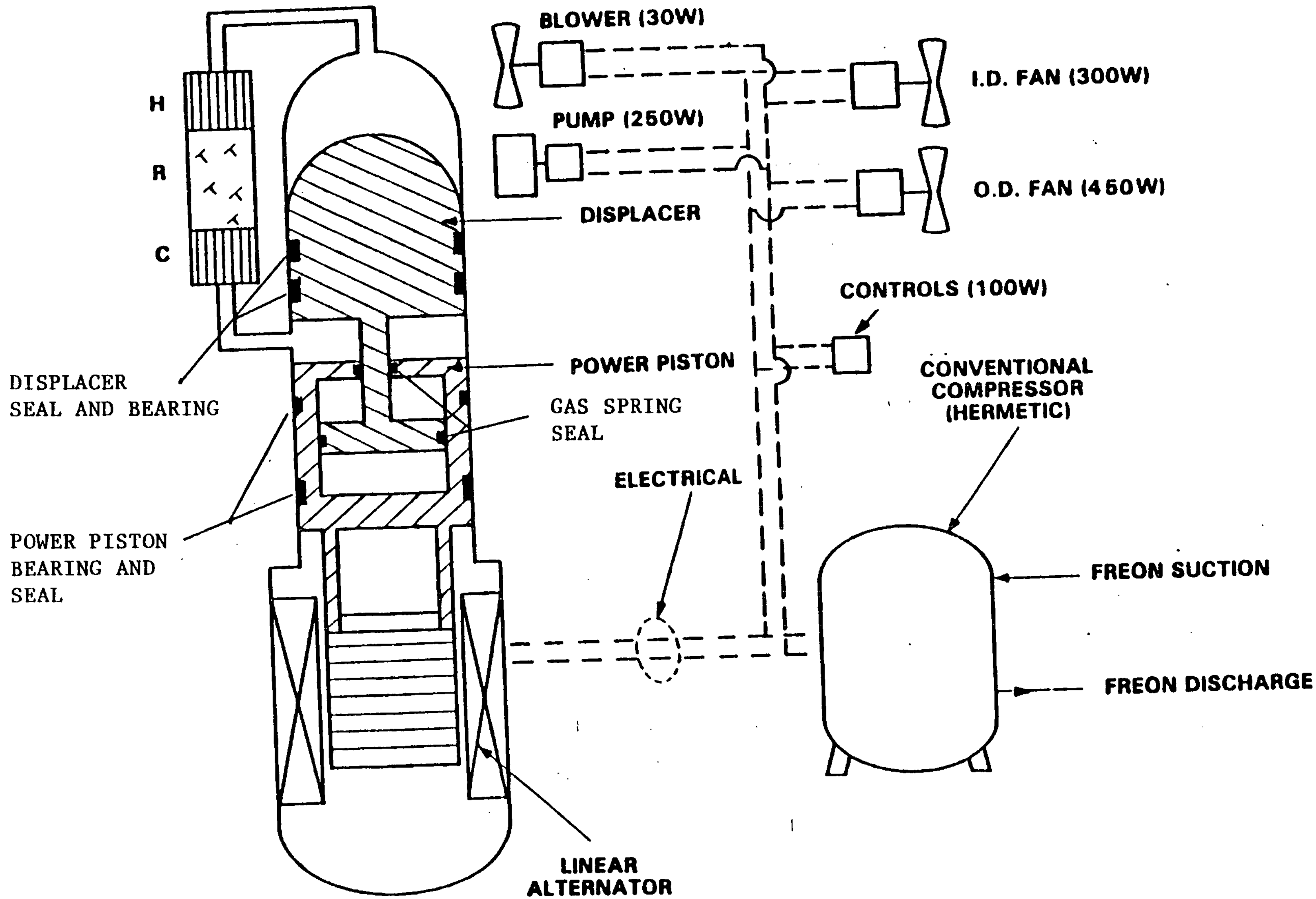

Figure 2.4 FREE PISTON STIRLING ENGINE WITH LINEAR ALTERNATOR

necessarily avoided by this option. The system can use a driven displacer to control displacer stroke and, therefore, power output which can be matched to that required by the compressor. The magnitude of the electric load directly impacts on the motion of the power piston (represented as a viscous load). There are only two moving pistons (and two degrees of freedom) in this system arrangement which could facilitate design and control.

3.0 LIFE AND RELIABILITY

3.1 Experience To Date

Most of the effort on FPSE driven heat pumps has been directed toward achieving acceptable performance levels rather than demonstrating life and reliability. As a practical matter, therefore, there is only limited experience with operating equipment. In spite of this limited operating time, several observations can be made relative to life and reliability based on data from those programs which have received the major funding over the last 5 years.

(A) G.E. (Linear Compressor Program)

No endurance testing was undertaken with completely integrated FPSE/heat pump systems as part of this program. However, this system used rubbing bearing and seals. Limited endurance testing of these subassemblies indicated that these would be major life limiting factors in system design given the operating conditions in the engine. Additional life and reliability issues are associated with system control and the hot end subassembly.

(B) MTI - Hydraulic Transmission

Relatively few operating hours (several hundred) have been accumulated on the engine/heat pump system. Recently similar engines driving a linear alternator have undergone life testing in order to demonstrate their applicability to space applications. Over 1,000 hours of testing* has been accumulated to date and life testing goals for this equipment have been raised to 10,000 hours.

It should be noted, however, that the heat pump system differs from the linear alternator systems since it uses a relatively complex hyudraulic transmission coupled to the Stirling engine via flexing diaphragms. With the counterweight the hydraulic transmission itself requires four rubbing bearing/seals and two shaft seals between the high pressure hydraulic oil and the freon working spaces of the compressor. The hydraulic subassembly could be a more life limiting factor than the power side of the engine itself. Little operating experience has been gained on a complete transmission/compressor subassembly.

As indicated above, the experience with the FPSE/HPprograms provide little guidance on the life and reliability potential of such systems. Experience from other programs indicates, however, that FPSE equipment may be able to achieve adequate performance in this regard. Examples include:

- The long term life testing of mini Stirling engine devices (5 watts) for use in artificial heart pumps. The units are couples to the load via bellows and have operated in excess of 10,000 hours. Larger units (15 kW) are being designed with stress levels in the bellows similar to that in the heart pumps.

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* As of early 1985.

- The extended life testing of diaphragm free piston Stirling engines at Harwell, England. Individual units with electric power outputs of up to 120 watts have been tested for as long as 80,000 hours. These units are coupled to a linear alternator via a lightly flexed diaphragm, thereby, eliminating power piston bearings and seals. The displacer is spring mounted and uses the surfaces between the piston and cylinder as a form of simple gap regenerator.

The applicability of this experience with very small Stirling like engines is not clear. For example, scaling up of the "Harwell Engines" would involve greatly increasing diaphragm diameters and strokes. Whether this can be done simultaneously consistent with reasonable engine dimensions is uncertain. Also, the use of highly effective regenerators to increase efficiency might necessitate displacer piston seals which would introduce this potentially life limiting factor.

As indicated in Appendix 2, some of the most germane experience relative to the life potential of free piston equipment may be related to the extensive programs over a period of 20 years to develop long lived thermal/mechanical equipment for space cryogenic applications. These programs have verified that properly designed seals and bearing have life potentials in excess of 10,000 - 20,000 hours. However, the practical application of this experience to FPSE for commercial applications has yet to be determined.

3.2 Limiting Factors

There are a number of technical issues which could limit the life and reliability of FPSE driven heat pumps. These include:

- Bearing and seals which must operate unlubricated in a dry helium environment.
- Hot end components which must operate over thousands of on-off cycles annually (at least 10,000*) at temperatures in excess of 1400° F.
- Controls which must match FPSE output with highly variable load (and, in some cases, dynamic) conditions.
- Coupling arrangements between the FPSE and heat pump compressors which often, but not always, involve some combination of numerous moving parts, diaphragms and bellows, or new design compressor assemblies (detracting from the superficial simplicity of the engine itself).

Of the above, the issues which could most influence the ultimate life and reliability and design of the FPSE system are the bearings and seals. As indicated in Appendix 2, there has been a significant effort to develop rubbing seals, gas bearing designs, etc. as part of long term programs to develop thermomechanical equipment for space applications. In many cases, such as with cryogenic refrigeration, the equipment must run unlubricated many thousands of hours in a dry helium environment - a similar requirement as for a FPSE/heat pump system.

This space related experience could provide some insights as to the measures required to attain long life and high reliability with such equipment.

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*Continuous modulation is being considered as a means for reducing cyclic operation. This is done, however, at the expense of additional annual operating hours.

3.3 Bearing and Seals

There are three approaches which are generally pursued for bearing and seals:

- o Gas bearing and clearance seals
- o Sliding bearings and seals
- o Barrier seals (diaphragms or bellows)

Due to the critical importance of bearings and seals in determining life, reliability, and performance, these options are discussed in more detail in Appendix 2. The following discussion only reviews their impact on system life and reliability as applied to configurations now under development.

3.3.1 Gas Bearing and Clearance Seals

By using gas bearing and clearance seals all rubbing surfaces and, therefore, wear can be eliminated from a FPSE. Such a system, for example, is now being considered for long lived operation of a FPSE generator in space where both the power piston and displacer piston are mounted on gas bearings. Experience with other projects* using this approach attests to the possible elimination of wear in free piston equipment in properly designed systems.

Presently, MTI is the only developer committed to the use of gas bearing and clearance seals in FPSE/heat pump applications. Their system uses hydrostatic gas bearings which will be pressurized by extracting high pressure gas from within the engine - probably the gas spring bounce space which is cyclically pressurized during each stroke. In principle this arrangement could result in highly reliable long lived operation of the displacer piston. The sources of unreliability in this concept include:

- o The gaps required in clearance seals and gas bearings are usually less than one mil if leakage losses are to be acceptable. With such tolerances, even small particles of foreign matter associated with system assembly trapped in the clearance gas can result in system failure.

- o During system start-up and shut down, there will be metal to metal contact between pistons and cylinder walls. Given the possible need for over 100,000 cycles over a useful life, this rubbing contact is an issue which must be resolved.

- o The hydrostatic gas bearings will require check valves and active controls which could introduce a new source of unreliability (Note: a design goal in space equipment is the complete elimination of all valves).

It should be noted, however, that similar issues to these noted above have been addressed in free piston equipment for space use. Proper design, meticulous care in cleaning and assembly, and proper choice of materials

*For example, the helium compressor in the R^3 project at Arthur D. Little, Inc., has demonstrated nearly 10,000 hours of operation when using hydrodynamic gas bearings and clearance seals on the pistons.

may successfully address these issues. As indicated in Section 5, however, the very close tolerances involved and care in assembly could significantly impact on manufacturing costs.

3.3.2 Sliding Seals and Bearings

Most Stirling engine equipment developed to date (including kinematic engines) uses sliding bearings and seals. The design and fabrication of such equipment is more consistent with commercial practice than are gas bearings and clearance seals. To date, in the FPSE programs, the life and reliability problems of the sliding surfaces have usually been overshadowed by the problems of achieving adequate dynamic stability and performance. Little life testing has been undertaken on which to assess the impact of sliding bearings and seals on FPSE life characteristics.

The limited life testing to date by such firms as G.E., however, suggests that wear rates on these sliding surfaces were in excess of those acceptable for heat pump applications. Table 3.1 provides a list of material combinations tested by G.E. in component level development activities. Major efforts have also been expended to develop long lived sliding seals and bearings as part of programs to develop thermomechanical equipment for spaceborne applications where lifetimes similar to the heat pump applications will be required. These programs have indicated promise of achieving extended sliding seal/bearing lifetimes assuming proper material selection and operating conditions.

There is also some favorable experience with contact piston seals in several of the kinematic Stirling engine development programs directed toward lower power density applications (i.e., nonautomotive). For example, recent results from Stirling Power Systems endurance testing indicate life potentials in excess of 10,000 hours.

The relatively favorable experience for sliding seals and bearings referred to above was, however, achieved only after extensive R&D effort which included endurance testing of sliding seal/surface combinations in special purpose test rigs which simulated the operating conditions in the applications under consideration. The pertinent operation conditions of the seal which were simulated were:

- temperature level;
- rubbing velocities;
- pressure level (related to pressure drop across the seal); and
- load reversals.

All of these are are of critical importance in determining seal and bearing life. Commonly used reference on seal wear criteria[1] provides initial guidelines for assessing conditions under which seal wear should be acceptably low. Encouraging experience in related technologies suggests that there may be FPSE designs and operating conditions which will allow for using sliding bearings and seals.

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[1] Such as Archord's equation referenced in Appendix 2.

Table 3.1

MATERIAL COMBINATIONS FOR SLIDING BEARINGS AND SEALS

Seal Material	Matrix	Filler	Producer	Coefficient of Expansion (1)
K30C	PTFE	Carbon Graphite	Koppers	50
Rulon J	"	Poly	Dixon	52
Rulon LD	Polymeric Fluoro-carbon	PTFE	Dixon	35
Envex 1228	Polyimide	"	Rogers	30
Envex 1115	"	MoS_2	Rogers	27
SP 21	"	Graphite	DuPont	27
SP 211	"	Graphite PTFE	DuPont	29

Note: (1) Parallel to molded direction (120^{o} - $400^{o}F$) in/in/^{o}F x 10^{-6}

	Piston Material	Ring Seal Material (1)	Liner (~12 RMS)	
			Material	Coating
1	2024-T4 Al	Vespel 21	440 C(2)	Cr Plate (2)
2	"	K30C	"	"
3	"	Rulon LD	"	"
4	"	K30C	"	None
5	"	Rulon LD	"	"
6	"	Vespel 21	"	"
7	"	Vespel 211	"	Xylan + MoS2
8	"	Envex 1115	"	CR + MoS2
9	"	Envex 1228	"	None
10	"	K30C	"	MoS_2
11	"	Envex 1228	"	MoS_2
12	"	K30C	"	Xylan
13	"	K30C	Nitronic-60	None
14	"	Rulon J	"	None

NOTES: (1) Approximate characteristics and composition of Ring-Seal materials are listed in Table 5-27.

(2) The current design specifies 4340 steel (Cr Plated) with a finish of 12 RMS.

This, in turn, suggests that operating conditions for rubbing seals and bearings could significantly impact on the overall engine design approach (contrary to experience to date where seal design "fell out of" the engine design).

3.3.3 "Barrier Seals"

Very small Stirling devices such as the heart pump (University of Washington, 5 watts) and thermomechanical generator (Harwell, England, 120 watts) have used diaphragms or bellows as a means of sealing the "power piston" from the working gas. Several of these units have operated for over 20,000-80,000 hours -- attesting to the reliability of such systems when maximum stress levels in flexing members are kept well below (20-40%) the fatigue limit. This approach is being applied to larger capacity equipment by MTI in their hydraulic transmission.

It should be noted, however, that in some configurations the use of a barrier seal in the power extraction system does not eliminate the need for seals and bearings (rubbing and/or gas) for the displacer piston (again, as exemplified by the MTI unit). However, the displacer seals do not have to be hermetic and, therefore, present a more tractable (although still difficult) problem.

To date, there is only limited experience on which to make judgements as to the applicability of barrier seals to larger, high efficiency, equipment such as required by heat pump applications. Observations on this issue include:

- There is no inherent reason why diaphragm seals cannot be made increasing larger to allow transfer at higher power levels. However, diaphragm size and deflection must be consistent with overall realistic engine designs. For example, the MTI diaphragms are 9.2 inches in diameter as compared to the engine displacer piston diameter of 3.2 inches. As a practical matter, the design constraints on diaphragm diameters may results in deflections and stress levels higher than often considered in the range required for consistently achieving long life. For example, the predicted stress levels in the MTI diaphragms are about 39,000 psi, while the University of Washington team, by contrast, tried to keep stress levels in their diaphragms and bellows under 20,000 psi.

- Bellows seals can be designed to operate well below the endurance stress limits if pressure balancing is used to isolate the bellows from the engine working fluid pressure (helium on the engine side and hydraulic fluid on the outside). The University of Washington is currently developing a 250 watt engine for the Army using a bellows seal/hydraulic transmission with the design based on their experience on heart pumps. Similarly this organization has undertaken design studies for 15 kW units using bellows seal transmissions. The need for pressure balancing and movement of significant quantities of hydraulic fluid results in a "non-simple" design with a multiplicity of bellows welded joints. Experience with the 250 watt Army unit should provide valuable experience on the scalability of this concept. Even though barrier seals are not a clear cut solution to the seal problem, they do have high reliability and long life if properly designed. This suggests that their use should be considered extensively and systematically as part of FPSE/HP development programs.

3.4 Hot End Components

The hot end subassemblies include the burner/nozzle assembly, engine heater head, and air preheater. For efficient engine operation these assemblies must operate at temperatures in the 1200-1400^{o} F range with some exposed burner liner material exposed to even higher temperature levels.

There is little experience on which to base the life projections of a system exposed cyclically to this severe operating condition. However, experience in industrial applications with high temperature air preheaters is instructive. With such heavy duty systems, cyclic operation is usually discouraged and a slow heating up of the system is often recommended to avoid excessive thermal stresses and mechanical distortions.

In light of industrial experience, present designs for FPSE emphasizing low cost and light weight may not hold up over long periods of time (limited testing suggests this is often true). Using proper materials and heavy duty designs (usually thicker and heavier) could probably result in systems of required durability. However, to date the cost and thermal mass implications* of acceptable designs do not appear to have received significant attention.

3.5 Power Extraction

Most discussions on the life and durability potential of FPSE/HP focus on the engine itself. It should be noted, however, that several concepts require complex coupling systems with a relatively large number of moving parts. All the power extraction concepts with major funding are more complex than a conventional kinematic drive freon compressor. As a practical matter the power extraction means could well be the life/reliability limiting factor in several of the system designs under active consideration.

(a) Diaphragm Couple Hydraulic Transmissions (MTI)

Figures 3.1 and 3.2 are schematics of the MTI diaphragm coupled hydraulic transmission. The power extraction system of MTI uses two large diameter diaphragms, 4 large area oscillating sliding seals in the hydraulic oil volume, an active hydraulic fluid inventory system, two high pressure hydraulic fluid to freon sliding seals, and an active oil leakage make-up pumping system. There has been little testing done (measured in 10's of hours) on a complete assembly and, therefore, there is only limited experience on the life and reliability of this system. However, the relative

*Heavier construction leads to higher thermal mass and higher cycling losses during part load operation.

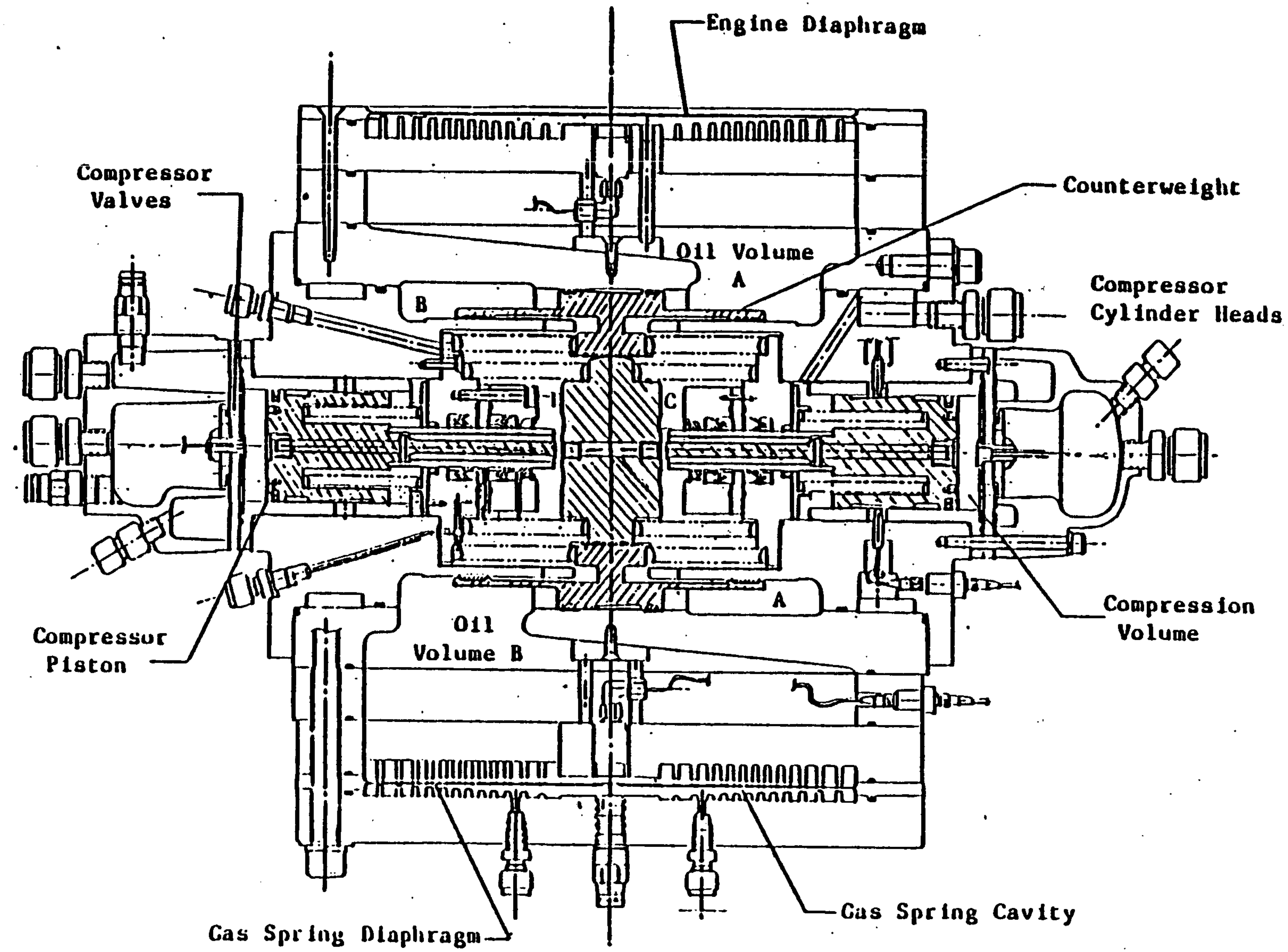


Figure 3.1 MTI HAHP COMPRESSOR AND HYDRAULIC TRANSMISSION LAYOUT

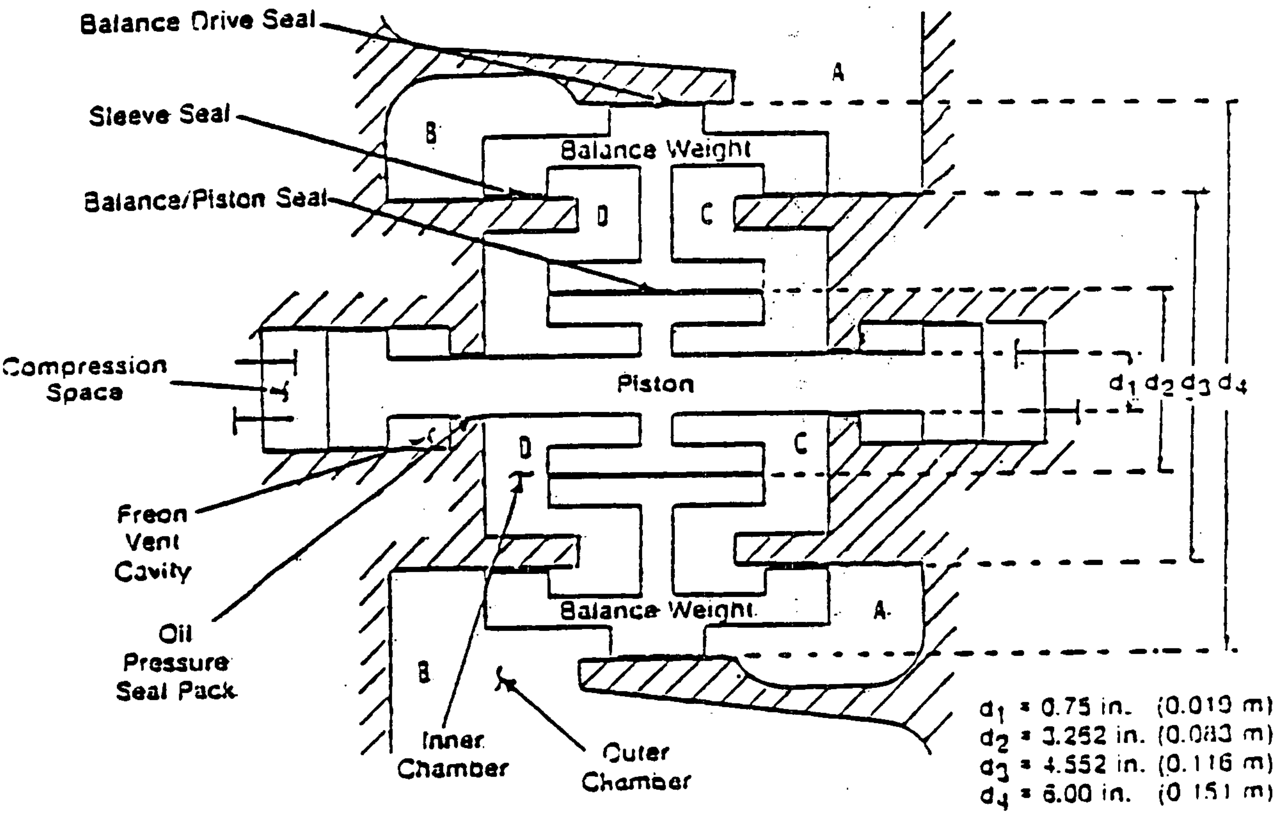


Figure 3.2

COMPRESSOR SCHEMATIC - MTI HAHP

complexity of the system is cause for concern as to its durability and ability to operate efficiently over long periods of time after seal wear has taken place.

Additional analytical and experimental efforts would be required to address this issue.

(b) Linear Compressors (G.E.)

Figure 3.3 is a diagram of the linear compressor unit developed most recently by G.E. In this arrangement the mechanism for transferring power to the compressor system is quite straightforward, i.e., movement of the power piston directly results in movement of the compressor piston relative to the power piston housing with no intermediate mechanisms. As a result,the life and reliability of the engine to freon compressor coupling is very high and, probably, not in itself an issue. This superficial simplicity of coupling is one reason this concept has been pursued for over 10 years.*

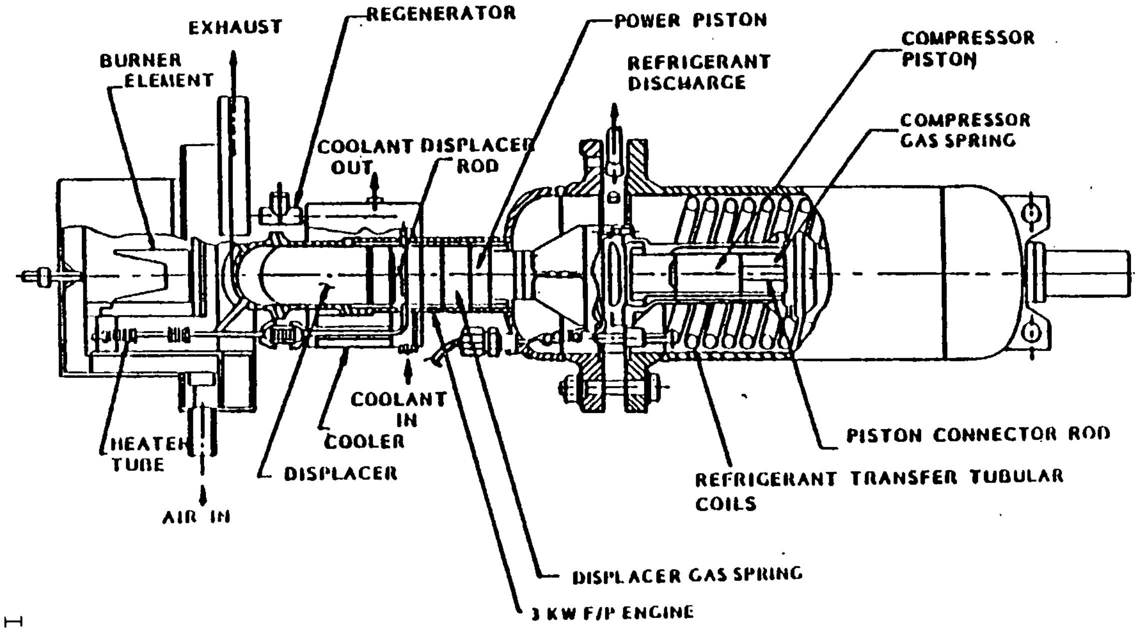
The dynamics of the linear compressor unit are directly tied to those of the Stirling engine (and vice versa) resulting in a complex 3 body** problem. As a result, there are unresolved issues as to the ability to reliably control the system over a wide range of ambient air temperatures (which directly affects the dynamics of the compressor assembly) and the impact on efficiency of off-design operation. As indicated by G.E. experience, this control issue is reflected in a mechanical reliability and life issue if the free piston compressor piston "bangs into" its stops as a result of low gas spring (almost disappearing) stiffness of low ambient air temperatures.

Additional sources of unreliability in this concept include:

- They cyclic stressing of the serpentine freon coils connecting the oscillating power piston from the stationary housing. As with diaphragms, the average stresses on this coil can be kept well within endurance limit values. Nonetheless, there is always a question regarding the life and reliability of critical flexing members - particularly when the realities of material and assembly imperfections are considered. Previously referred to experience with small Stirling devices as well as limited endurance testing at G.E. suggests, however, that proper design can circumvent these problems.
- The relatively high loadings (nonsinusoidal motion with high acceleration) and uncertain lubrication (no sump) in the compressor itself could reduce its life as compared to conventional compressor equipment.

The sources of unreliability indicated above have not yet been well quantified and so doing would require extensive component and system endurance testing. However, the inertial compressor system does have the advantages

*Starting initially with SUNPOWER and ERG.
**Actually, a four body problem, if engine casing vibration is considered.



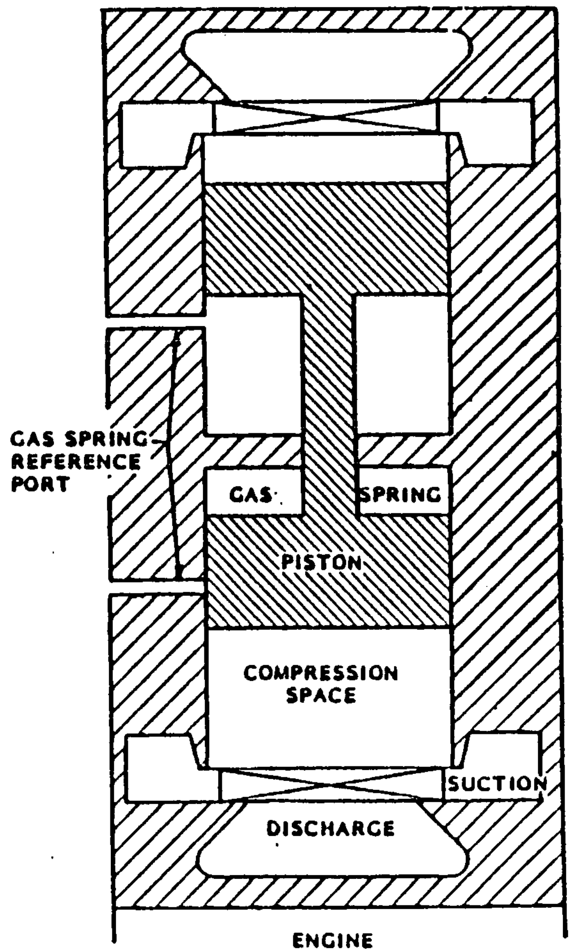


Figure 3.3 G. E. INERTIAL COMPRESSOR SCHEMATIC

of mechanical simplicity so that if the above issues can be resolved, this concept could be an attractive one.

(c) Linear Alternator

Conceptually, the use of a linear alternator to extract power from the FPSE system is very straightforward. Also, properly designed linear alternators could be as reliable as rotary alternators. As a practical matter, therefore, the power extraction means in this concept is probably not in itself a serious life and reliability barrier within the overall system.

4.0 EFFICIENCY

4.1 Experience to Date

As indicated in Reference 1, the definition and use of efficiency terms as applied to FPSE is complicated by the fact that no rotating shaft is available at which to measure useful power available. Thermal efficiencies for the engine section of the FPSE/HP can be defined based on the useful power that crosses the boundary between engine and load. Such thermal efficiencies which are commonly applied to the engine of a FPSE/HP can be somewhat misleading if the "load" or transmission assembly contains gas springs which are essential to engine operation but whose loss is accounted for as part of the load rather than the engine. Therefore to present a fair picture of FPSE performance it is necessary to deal with the efficiency of the engine-load combination as well as the engine alone.

Table 4.1 is a summary of efficiency based on information in the contractor reports as modified to use the definitions provided by Reference 1 to account consistently for engine aid power (combustion blower controller and control power attributable to the engine) and transmission losses. Table 4.1 also presents heat pump thermal COP's with and without penalty for overall parasitic power (i.e., engine aid power plus coolant pump and condenser and evaporator fans).

For the engine only, not considering combustion losses, the brake thermal efficiency ($_{bt}$) is defined as:

$$\eta_{bt} = \frac{\text{Engine Brake Power}}{\text{Energy Input to Engine Head + Engine Aid Power at Source Efficiency}}$$

The brake thermal efficiencies as measured to date range from 19 to 33 percent depending on contractor and system configuration. Superficially this efficiency range creates the impression of being comparable to that attained by kinematic engines (Stirling and I.C.).

However, when combustion system losses are taken into account "brake gross thermal efficiency" levels typically drop to 15 to 25 percent where:

$$\eta_{bgt} = \frac{\text{Engine Brake Power}}{\text{HHV of Fuel + Engine Aid Power at Source Efficiency*}}$$

This definition is similar to that of brake efficiency in a kinematic engine.

The transmission or load efficiencies indicated are defined as:

$$\eta_{T} = \frac{\text{Gross Load Power (i.e. Power to the Refrigerant Fluid)}}{\text{Engine Brake Power}}$$

This is roughly equivalent to the mechanical efficiency of an open shaft driven compressor.

*The numbers in Table 4.1 assume a "source efficiency of 30 percent" as recommended by Reference 1.

Table 4.1

FREE PISTON STIRLING ENGINE PERFORMANCE COMPARISONS

	Engine Only	Engine and Heat Source	Load or Transmission	Engine Heat Source and Load	Heat Pump Thermal COP(5)	
	Brake(1) Thermal Efficiency	Brake Gross(2) Thermal Efficiency	Load (3) Efficiency	Gross Thermal(4) Efficiency	95°F Cooling	47°F Heating
MTI (6)						
• Advanced Heat Pump Goal	.40	.33	.8	.26	1.2 (.83)	2.5 (1.5)
• Projection for Current (8) Heat Pump	.32	.26	.8	.21	1.0 (.69)	2.2 (1.2)
• Linear Alternator (Test Load)	.32	.26	.72	.19	---	---
G.E.	(C) (H) (7)	(C) (H)		(C) (H)		
• Proto #2 Projection	.30 .25	.23, .18	~1.0	.23, .18	.79 (.60)	1.6 (1.1)
• Proto #1 Accomp.	.19—.25	.14—.19	~1.0	.14—.19	.5–.7(.4–.6)	1.2–1.6(.9–1.1)
Sun Power						
• Linear Alternator		.18—.19	.87	.16—.17	---	---

Notes:

1. Brake Thermal Efficiency (ηbt) $\equiv \dfrac{\text{Engine Brake Power}}{\text{Energy Input to Engine Head + Engine Aid Power at Source Efficiency}}$
2. Brake Gross Thermal Efficiency (ηbgt) $\equiv \dfrac{\text{Engine Brake Power}}{\text{Fuel Flow *HHV + Engine Aid Power at Source Efficiency}}$
3. Load Efficiency ($\eta \ell$) $\equiv$ Power to Refrigerant/Engine Brake Power
4. Gross Thermal Efficiency (ηgt) $\equiv \dfrac{\text{Gross Load Power (i.e. Power to Refrigerant)}}{\text{Fuel Flow *HHV + Engine Aid Power at Source Efficiency}} = \eta bgt * \eta_\ell$
5. Thermal COP = Heating (or Cooling) Delivered/Fuel Flow *HHV
   Numbers in Parenthesis Include Parasitics (added to Fuel Flow *HHV at Source Efficiency of .3).
6. In the case of the MTI heat pump the boundary between engine and load is the diaphragm between engine and transmission. Therefore transmission viscous and gas spring loss are accounted for in the transmission or load efficiency rather than the engine efficiency.
7. (C) and (H) refer to Cooling and Heating respectively.
8. Projections for Current MTI Heat Pump Design Based on Component Performance from Test and/or Analysis.

Notes for Table 4.1.

1. Equivalent Thermal COP = $\dfrac{\text{Delivered Heat (or Cooling)}}{\text{Fuel Flow *HHV+}\dfrac{\text{Electric Power}}{.3}}$

   All systems involving freon vapor compression cycle use Westinghouse/ORNL fluid cycle COPs. All system goals, based on approximately 500 watts ancillary electric. As defined here, ancillary electric power is all externally supplied electric power except the compressor power for electric heat pumps.

2. Ultimate potential of FPSE is assumed similar to that of other engine designs - i.e. kinematic and I.C. engines.

3. Kinematic and I.C. Engines assumed to have brake gross thermal efficiency of 30 (HHV basis).

4. Based on discussions with Mr. R. DeVault of ORNL.

5. Condensing furnace based on ADL estimates. Air Conditioner COP taken as 10% greater than heat pump cooling COP.

6. Based on current and projected results of MTI and G.E. developments. Ancillary electric power ranges from 350 watts (MTI advanced heating design) to 1100 watts (G.E. designs).

7. ADL estimate based on data from several sources such as references 7 through 13.

The transmission efficiency indicated for the MTI system is 80% due to the viscous and leakage losses in the hydraulic transmission. Initial results with the counterweight indicated in Figure 2.2 were well below this figure (60 - 70%). However, recent (late 1985) results with an external counterweight were as high as the mid 80% range due to large reductions in viscous losses associated with counterweight motion. This indicates that acceptable coupling efficiency levels can be attained with hydraulic transmissions.

At design conditions, the efficiency at which a FPSE can drive the compressor of the heat pump refrigerant loop has been measured at 14 to over 20 percent after both engine and transmission losses have been accounted for. By comparison, a 30 percent kinematic engine driving a conventional mechanical compressor with an efficiency of 85 percent would have a comparable efficiency of 26 percent.

The last column is the resultant COP of the system when used to drive a conventional R-22 heat pump cycle. As indicated, based on a combination of measurements and analysis the design point heating COP's corrected for parasitic power range from 0.9 to 1.2 for current designs of both the MTI and G.E. systems. This level of COP is not sufficiently high to result in substantial energy savings relative to conventional systems (i.e., high efficiency furnaces and/or electric heat pumps).

However, improvements in the current engine designs indicate that heating COP's can be increased to 1.5. At these higher levels the FPSE/HP performance is appreciably better than conventional systems and is comparable to other engine driven heat pump alternatives, such as those using I.C. engine. As indicated by Table 4.1, good progress has been made in demonstrating that FPSE/HP systems can attain performance levels under steady state conditions of economic interest and competitive with other heat actuated options. The key issues will be whether these performance levels can be attained with equipment of required reliability and cost, and can be maintained during cyclic operation.

4.2 Observations on Efficiency

4.2.1 Loss Mechanisms

The common perception is that FPSE should be comparable in performance to KSE in the intermediate power range of interest (1-10 kw) which implies being able to approach 30 percent in the equivalent of brake efficiency. Most experience to date is almost 20 - 30 percent below this level. The reason for this are still not clear but include some combination of:

- o Poor phase angle and amplitude control between displacer and power piston possibly resulting in further (as compared to kinematic) engine deviations from ideal Stirling cycle performance.

- o Gas springs required to result in naturally oscillating piston position have hysteresis (thermal and leakage) losses which may offset loss reductions associated with eliminating mechanical drives.

- o Operation of electromagnetic displacer drive mechanisms (per the MTI system) which in their present configurations require external power thereby increasing parasitic power draws.
- o Gas bearings (hydrostatic) require extraction of high pressure gas generated by the cycle and, therefore, represent a net loss of useful output power.
- o Leakage losses of gas by clearance seals detracts directly from indicated power output.

As indicated above, there are a number of loss mechanisms associated with FPSE which are not present in a KSE at all.

Many of these losses are not well understood and probably are often under-estimated. For example, as indicated in Appendix 2, the leakage losses by a clearance seal which is poorly centered in the cylinder is 2.5 times that in the ideal situation (perfect centering).

Many, if not most, participants in the Stirling engine community feel that a well developed FPSE should be able to achieve efficiency levels comparable to those demonstrated in KSE. However, additional experimental and analytical work will be needed to assess the role of the various loss mechanisms and verify this efficiency potential.

4.2.2 Cyclic Losses/Seasonal Performance Characteristics

Almost all testing and analysis has been done for selected steady state conditions at which heat pumps are normally rated. The performance parameter of practical interest which define annual energy consumption is the seasonal performance factor (SPF). The SPF of electric heat pumps is known to be considerably lower than indicated by steady state performance due to thermal losses during transient operation (cooling down of heat exchangers, need to establish working fluid thermodynamic conditions on start-up, etc.). As a result of such cyclic losses the SPF of electric driven heat pumps will typically only be 75 to 85 percent of that indicated by steady state performance characteristics.

The operation of a Stirling engine (FPSE or KSE) requires heating up a relatively large mass of material contained in the heater head, air preheater, combustion chamber, and regenerator to relatively high temperature levels (800-1400°F). The alternate heating and cooling of the hot end components during cyclic operation results in a net heat loss to the system.

Due to the high temperatures involved, the cyclic losses in the Stirling engine (FPSE or KSE) are likely to be higher than those in the heat pump cycle itself. Preliminary estimates at Arthur D. Little (see Appendix 3) indicate such losses could easily be 10-20 percent taking into account only the thermal cycling of the hot end components. This does not include heating up of the regenerator which would further detract from performance.

These relatively high cyclic losses tend to favor a control strategy using continuous modulation in order to reduce the number of operating cycles.

This strategy, however, results in:

- o A high percentage of part load operation which for several of the systems under development could result in severe efficiency penalties.
- o In higher operating hours per year with associated ramifications on engine maintenance and life.

The issue of estimating SPF has not received much attention to date. The need to achieve good performance based on this parameter is, however, critical. System design and control strategy could both impact on SPF. As a result, near term attention is needed to quantify this issue better and determine how it might impact the system selection process.

4.3 Computer Projections

It is generally agreed that modeling the performance of FPSE is a much more difficult undertaking than for KSE. This is due to the direct interaction of engine/thermodynamic with system dynamics which occurs in a FPSE. In addition, the ability to model effectively the aforementioned loss mechanisms on engine thermodynamics and dynamics is highly questionable.

Most of the FPSE computer models are proprietary and, therefore, it is not clear what loss mechanisms are accounted for and how they are factored into the predictive process. For example, some of the programs account for the seal and bearing losses "after the fact", i.e., as add-on effects which do not directly affect gross thermodynamic and dynamic performance. The adequacy of this approach is questionable. However, errors so introduced are hard to quantify.

In general, there has been a tendency for hardware experience to be significantly lower than computer projections. The closure between experience and projections is improving but still requires calibration of the programs against specific data points in order to map performance characteristics of a single engine. As a result, these models are applicable only to a specific configuration and even then only when operating over a relatively narrow operating range. Reasons for this probably include the fact that the calibration processes do not necessarily result in a more accurate description of actual physical process and therefore do not necessarily result in a better model of physical reality.

The development of FPSE hardware is a very costly process. Widely available analytical models which are more accurate and applicable to a wider range of promising configurations than those now available would be a tremendous benefit to the industry and public sector user agencies alike by reducing costly and time consuming errors in hardware design. This is recognized by many in the community and is resulting in such programs as the oscillating flow field experimental work at Argonne National Laboratory and the development of FPSE computer models at NASA Lewis, and the harmonic flow analytical approaches at ORNL.

5.0 SYSTEM COST AND ECONOMICS

5.1 Cost Issues

There are three major factors which determine the economic potential for an engine driven heat pump:

- The initial installed cost of the system and, in particular, the premium which is paid relative to conventional options (gas furnace, electric air conditioning).
- The annual maintenance cost and useful system life.
- The system efficiency which determines, in large part, the annual energy savings which can be used to pay back the initial cost premium.

Several analyses[9,10] have been undertaken to quantify the above requirements and assess the potential for various system options to meet them.

These analyses indicate that in favorable geographical regions engine driven heat pumps show potential for resulting in net energy savings of $250-$450 per year as compared to conventional alternatives assuming performance and reliability goals are achieved (per Table 1.2). Experience indicates that short payback periods of 3-5 years (maximum) are required to result in a significant market penetration. This implies that the allowable incremental installed cost (i.e., for the consumer) of an engine driven heat pump system can be in the $750-$2000 range. In the HVAC industry, the fully-loaded manufactured costs of equipment (before manufacturer's selling costs, profit, etc.) are approximately 50 percent of the price of that equipment to the consumer. This differential is made up of a series of mark-ups to cover incremental selling, working capital financing and other costs throughout the distribution system:

Fully Loaded Factory Cost = 100
Manufacturers Mark-up to Selling Price (35%) = 135
Distributors Mark-Up (30%) = 176
Contractors Mark-up (20%) = 210

This implies that the incremental equipment cost at the factory can only be $400-$1000. It is this equipment cost goal which must be considered when assessing the potential for any engine driven heat pump system. When dealing with a KSE or an I.C. engine, the above equipment cost would refer to the cost of the engine system itself since the heat pump system (including compressor) would be of conventional design.

However, in a FPSE system the compressor (i.e., part of the heat pump system) is often an integral part of the engine assembly. For example, both the G.E. and MTI systems incorporate the refrigerant compressor into the FPSE assembly. The OEM cost of a conventional 3-ton compressor is only about $90-$120. As a first approximation, therefore, the equipment cost constraint on the FPSE/compressor assembly may be increased to the $500-$1100 range.

As indicated in the next section, the FPSE/heat pump systems under development show initial promise of meeting this goal (based on contractor reports and analysis) but it should be remembered that these analyses are presently based on significant extrapolations from present practice and contain many uncertainties, since designs that simultaneously meet efficiency, low maintenance, and life requirements have yet to be demonstrated.

5.2 Cost Estimates

5.2.1 General Electric

Table 5.1 is a cost estimate summary for the advanced version of the G.E. linear compressor system. The numbers shown are a direct manufacturing cost and do not include profit, mark ups, sales, and other expenses. The manufacturing cost indicated for the FPSE/compressor assembly alone is about $940 which falls in the upper end of the range of allowable equipment costs (before distribution/installation).

The above provides basic plausibility that this system might meet initial cost constraints if the engine/compressor/heat pump can be fabricated as assumed in the Proto 2 design. However, the design changes in bearings, hot end assembly, and controls required to meet other system performance constraints may not be realistically factored into the cost analysis at this time. Also, there may be a need for a backup on very cold days (similar to electric resistance heating with electric heat pumps) which has not been factored into cost estimates to date.

One potentially low cost approach to providing such backup is to oversize the engine combustion system and waste heat recovery system so that the system functions as a conventional boiler on very cold days. The details of designs and associated cost implications for such an approach (including diverting the combustion gases away from the heater head so it won't overheat) have not been prepared.

5.2.2 Mechanical Technologies Inc.

Tables 5.2 and 5.3 summarize direct production cost estimates for the MTI compressor assembly and engine respectively as presented in Reference 8. The tables shown are for advanced versions of this equipment assuming a number of cost reduction steps are taken as compared to present production prototype designs. At the 50,000 unit per year level, the compressor cost is $315 and that of the engine $429 for a total of $744.

This cost estimate at first blush appears to be within the cost range for the engine/compressor combination which is required for potential economic viability. However, the costs of Tables 5.2 and 5.3 are direct manufacturing costs only and do not appear to include factory overhead, sales, and other costs which must be accounted for to arrive at estimates of factory cost (with profit) and consumer costs.

These additional factors could significantly increase costs over those presented and result in cost structures similar to those of G.E. and subject to most of the same uncertainties as to the degree to which the system costed is consistent with the required life and efficiencies of this application.

Table 5.1

PROTO 2 PRODUCT UNIT MANUFACTURING COST ESTIMATE

SUBSYSTEM	DIRECT LABOR	O/H 212%	DIRECT MATERIAL	TOTAL	%
Engine	$ 56.77	$ 120.60	$ 309.81	$ 487.18	31
Compressor	20.39	43.33	167.85	231.57	15
Engine Compressor Assy.	4.84	10.29	13.50	28.63	2
Combustor	8.14	17.30	94.50	119.94	7
Pressure Vessel	3.02	6.42	34.09	43.53	3
Stand	1.31	2.78	24.44	28.53	2
Outdoor Unit	80.77	171.64	165.00	417.41	26
Indoor Unit	40.77	86.64	95.00	222.41	14
TOTAL SYSTEM	$ 216.00	$ 459.00	$ 904.00	$ 1579.20	100
ENGINE COMPRESSOR ONLY				$ 939.38	

* Subtracting out the "outdoor unit" and "indoor unit" heat exchangers which are part of the heat pump loop.

Source: Reference 4

Table 5.2

SECOND PASS (APP. B CHANGES) COST SUMMARY - HAHP COMPRESSOR

ANNUAL PRODUCTION VOLUME	WEIGHT LBS.	MATERIAL $	LABOR MIN.	LABOR $	BURDEN $	SCRAP $	TOTAL $	TOOLING -000- $
3,000	115.4	370.98	61.67	18.84	25.81	.48	416.11	404.0
20,000	113.97	296.85	51.29	15.65	21.28	.36	334.14	705.1
50,000	113.97	282.28	44.58	13.61	18.78	.32	314.99	800.0
200,000	113.97	238.52	52.35	16.05	31.95	.60	287.12	1087.7

SOURCE: Reference 8

Table 5.3

SUMMARY - MTI FREE PISTON STIRLING ENGINE COSTS (83 ECONOMICS)

DEVELOPED BY MTI

(In November 1981, Pioneer determined the cost of a Free Piston Stirling Engine for MTI. Based on that detailed cost MTI has made a number of design and material changes that have significantly reduced the cost. These costs are shown below. Without being able to specifically verify the cost effect of the changes, Pioneer would generally agree with the magnitude of the change.)

ANNUAL PRODUCTION VOLUME	TOTAL COST $ -000-
3,000	587
20,000	449
50,000	429
200,000	391

SOURCE: Reference 8

5.2.3 Observations on Cost Estimates

The cost estimates presented in the available contractor reports are not sufficiently detailed to define what cost factors leading to a selling price to the distributor are or are not included. It appears, for example, that the estimates include only direct manufacturing costs and direct factory overhead and do not include the mark up required (typically 35 percent) to account for selling, servicing, and profit. Also, it is not clear how the various contractors treat manufacturing equipment and facilities capital and depreciation costs. As a result, the cost estimates of the different contractors cannot be directly compared to one another with a very high degree of confidence. Also the relationship of the costs presented to a realistic equipment selling price cannot be determined based on the available information.

Future programs in heat actuated heat pump technologies should strive to have cost estimates done in a consistent and transparent manner so that the manufacturing costs, distributor selling prices and installed costs of different options can be compared to one another in a meaningful way.

5.3 Limiting Factors

As indicated previously the acceptable consumer cost structure of any engine drive for heat pumps is about $400-$1000 ($150/hp to $350/hp). This is considerably higher than for light duty applications such as lawn mowers or automobiles where engine costs of $20-30 per hp are common. From a cost point of view, therefore, the heat pump application is more lenient than automotive applications and could afford considerably more leeway in component design and use of high cost materials. Also, as indicated above, the contractor cost studies indicate potential for meeting the cost constraints using advanced system designs. It is not clear, however, that these designs can also meet system performance requirements. Consequently, there are still open questions as to the ability of a FPSE/compressor system to meet the cost constraint of this application. These include.

o Gas Bearings and Clearance Seals

Gas bearings and clearance seals offer good promise of resulting in long life, due to elimination of rubbing parts in the engine. However, in a high pressure helium engine (800+ psi) the radial clearances required to keep leakage losses acceptable are generally on the order of one-half mil (possibly even less). Manufacturing mating parts with clearances in the order of .0002 to .0005 inches can be performed at acceptable costs if the parts are relatively simple, e.g., the piston and cylinder of a hydraulic valve or a hydraulic valve lifter. However, when mating parts must have close clearances on several diameters, as is the case with some parts in some FPSE, very close tolerances (often measured in millionth of inches) must be held on concentricity and, in some cases on fits as well as on dimensions. This is not accomplished as easily as for simple piston/diameter fits, and results in substantial increases in manufacturing costs. Consequently, the various tradeoffs between clearance allowances, efficiency, design arrangements, operating conditions (for example, lower operating pressures tend to allow larger clearances) and costs require additional investigation.

o High Temperature Subassemblies

More detailed studies associated with automotive engines indicate that the high temperature subassembly can be one of the more costly elements of the engine. The high temperature subassembly consists of:

- The burner assembly with associated nozzles, blowers, controls, and burner box.
- The heater head of the engine with passages for flow of helium gas to result in good heat transfer.
- The air preheater which must have multiple narrow passages for the flow of combustion gases and combustion air to achieve the needed effectiveness (eff. = 0.92).

It should also be noted that the current combustion system designs may not meet the low emission requirements of this application. Approaches for rectifying this possible problem area considered by the contractors include EGR.

Little endurance testing has been done on the hot end components to verify their ability to maintain high levels of thermal performance and low emission levels over long periods of time - particularly under cyclic operation. It is therefore, difficult to make judgements as to the design modifications (if any) and associated cost ramifications which might be necessary to simultaneously meet all life and performance requirements.

Experience with other types of high temperature heat transfer equipment indicates that required design modifications could include some combination of using high temperature (low creep) alloys, heavier construction to reduce dimensional distortions, addition of EGR loop to reduce emissions, and possibly modulated burners. The cost implications of these and other modifications are still unclear.

o Power Transfer

The systems used to date for transferring power from the engine to the heat pump cycle vary widely in their cost implications. The basis of comparison for any of the power transfer mechanisms can be a conventional kinematic compressor drive including a, crankshaft, piston connecting rods, electric motor and hermetic housing. The OEM cost for this subassembly is on the order of $90-$120 for a 3-ton, 2-cylinder, unit. This represents a very low cost option with which to compete.

(i) Linear Compressors

One advantage of the linear compressor concept is that it's a mechanically straightforward means of coupling the power cycle to compressor piston motion. Superficially, this system is mechanically more simple than its kinematic counterpart with associated favorable cost implications. This is one reason for this concept has been pursued by several organizations.

The appearance of low cost is, however, clouded by several issues:

o The concept requires a large power piston and large bounce space volume which increases the size of the high pressure hermetic envelope comprising the engine/compressor assembly. For example, the estimated weight of the resultant power piston, compressor assembly, and bounce space envelop is over 70 lbs* or about a factor 1.5- 2 higher than the compressor assembly of a conventional unit (exclusive of electric motors).

o The compressor assembly (pistons, sliding bearing, seals, gas spring, inlet and outlet valves) is of comparable complexity as a conventional compressor and has a much larger total displacement** (roughly twice that of a conventional unit). There is, therefore, little potential cost advantage associated with compressor assembly as compared to conventional units.

It appears therefore, that the linear compressor, at best, has comparable inherent costs as a kinematic compressor system and probably considerably higher if a portion of power piston and engine envelope costs are assigned to the compressor system. On the other hand, the linear compressor does not appear to represent in itself a significant cost barrier to the development of this technical approach.

(ii) Diaphragm/Hydraulic System

The diaphragm coupled hydraulic system as presently designed is quite complex relative to a simple kinematic compressor of similar capacity. The transmission operates at engine working pressures and is therefore enclosed by a pressure vessel. The weight of the moving parts alone is about 19 lbs as compared to just a few lbs for a kinematic compressor unit. Weight and size alone do not determine the relative costs of equipment options but they are good indicators if the basic equipment is of similar complexity. This is certainly the case in this instance. The diaphragm/hydraulic coupled compressor is a sophisticated piece of fluid mechanical equipment with 4 sliding bearings operating in the hydraulic fluid environment and two sliding shaft seals which must seal against leakage of high pressure hydraulic fluid into the low pressure refrigerant volumes of the heat pump compressor.

This sealing task is not as difficult as in the case of KSE where the shaft seals must prevent leakage of helium or hydrogen. Nevertheless, experience at MTI indicate it is a formidable sealing problem which entails many of the same cost factors as in the shaft seals of KSE.

In summary, it appears that the diaphragm coupled hydraulic transmission imposes a significant cost penalty on the system in its present configuration.

5.4 Cost Comparison with KSE

Early in the program it was expected that FPSE/HP would have significant cost advantages as compared to KSE/heat pumps. The two primary reasons for this potential include:

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* Based on report drawings and dimensions. One reason for the large size and weight is that the flexible serpentine refrigerant coils are contained in the hermetic, high pressure, enclosure.

** It should be noted, however that the larger displacement results from the variable stroke operation and has the advantage of resulting in minimal loss of capacity as suction pressure is decreased.

- The probable high costs of providing effective shaft seals for KSE.
- The "direct coupling" of FPSE to reciprocating compressors thereby eliminating costs associated with engine and compressor crank cases, crank shafts, and connecting rods.

This perception should, however, be re-examined in light of experience with both FPSE and KSE over the last few years. Specific issues which merit consideration include:

- System Size and Weight

The total pressurized volume* of a 7.5 kW KSE in Heat Pump applictions is about 15 cubic inches. By contrast both the G.E. and MTI units have pressurized volumes (including bounce spaces, hydraulic transmissions, gas springs, etc.) in excess of 100 cubic inches. Since working gas pressure are over 1000 psi, the pressurized volumes can significantly impact on costs due to their need for hermetic fabrication and increased material usage which, in turn, also affects system weight. The higher pressurized volumes of current GE and MTI FPSE/HP designs will tend to offset some of the cost advantages associated with the FPSE by virtue of eliminating crank cases and associated kinematic linkages.

For mass produced equipment, size and weight are but one of the indicators which is used by industry to estimate costs and compare design options. The above preliminary comparisons do serve to indicate, however, that the current FPSE system designs have inherent cost ramifications which merit further examination.

- Shaft Seal Elimination

Recent experience with shaft seals for KSE at Stirling Power Systems indicates that this component may not be as severe a reliability and cost factor as perceived several years ago - particularly for single acting machines operating at relatively modest pressure levels of helium (as compared to double acting, high pressure, automotive engines). The role of shaft seals in differentiating the costs of FPSE and KSE may not, therefore, be significant and should be reexamined in light of recent experience.

In addition, some FPSE configurations require shaft seals between high pressure hydraulic fluid and the freon vapor in the compressor. Although this shaft seal problem is somewhat easier to deal with than sealing against pressurized helium it still has many of the cost and reliability ramifications of the KSE shaft seal.

- Direct Coupling

To date, the lack of a rotating shaft with which to drive a conventional vapor compression unit has proven to be more of a detriment than an advantage for FPSE since no low cost, reliable, and efficient direct

* Based on Stirling Power Systems V160 engine. Engine is a 15 kW unit derated to 7.5 kW for heat pump applications.

coupling mechanisms have been demonstrated. In fact, the power transmission means under most active investigation are probably considerably more costly than the kinematic linkages associated with a KSE/HP system.

- Conventional Component Availability

The FPSE/HP concepts under development all utilize compressor units of novel design. As such, they cannot directly take advantage of the economies of high production associated with conventional kinematic compressors and associated controls which are available to the KSE option.

As indicated above, the perception that a FPSE/HP should have cost advantages as compared to a system using KSE is not necessarily valid based on current FPSE/HP configurations and should be re-examined in light of recent experience with both FPSE and KSE development programs.

5.5 References

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7. "Free-Piston Stirling Engine Diaphragm - Coupled Heat-Actuated Heat Pump Technology Program," First-Half Final Report, September 1983, MTI Report No. 83FPSE13.

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6.0 COMPETITIVE ENVIRONMENT

6.1 Configurations

The purpose of this section is to define the performance and status of the most probable competitive systems. The specific systems to be considered are:

- o Free piston Stirling engine heat pump;
- o Kinematic Stirling engine heat pump;
- o IC engine driven heat pump;
- o Absorption heat pump;
- o Electric heat pump;
- o Condensing furnace and electric air conditioner.

The first four of these systems are either developmental or in very limited use for residential applications. The last two electric heat pump and condensing furnace and electric air conditioner constitute conventional system alternatives.

Other gas fired heat pumps which have been the subject of some developmental effort but are not specifically considered here are: Brayton cycles (difficult to scale to residential sizes); Rankine cycles (not considered competitive in cost in performance to other options); chemical heat pumps (in an early stage of development and currently targeted for larger applications); and ejectors (unlikely to have sufficiently high performance for space heating and cooling).

6.2 Comparison Criteria

Table 6.1 describes the various competing systems in terms of the following criteria:

- o Performance estimates expressed as an equivalent thermal COP for 95°F cooling and 47°F heating design points;

- o Estimates of installed system costs, life and annual maintenance costs;
  - o Commercial status;

- o Technology trends and development barriers including institutional considerations such as emissions and noise.

6.3 Performance Estimation Approach

Performance is described by an equivalent thermal COP defined as:

$$COP_{et} \text{ - Equivalent Thermal COP} = \frac{\text{Delivered Heat (or Cooling)}}{\text{Fuel Flow*HHV} + \left(\frac{\text{Electric Power}}{.3}\right)}$$

$$\frac{1}{COP_{et}} = \frac{1}{COP_t} + \frac{1}{.3\ COP_e}$$

Arthur D. Little, Inc.

Table 6.1 COMPETING SPACE CONDITIONING SYSTEMS

		Free Piston Stirling Heat Pump	Kinematic Stirling Heat Pump	I.C. Engine Driven Heat Pump	Absorption Heat Pump	Electric Heat Pump	Furnace & Electric A/C
Equivalent Thermal COP[1] 47°F Heat / 95°F Cool	Goal	2.0/.9[2]	2.0 /.9[3]	2.0/.9[3]	1.7/1.1[4]	.9/.9	.9/.9[5]
	Current	1.1 - 1.3/.6 - .8[6]	No current residential size units	Field test units approaching goal	1.04/.46	Current high efficiency units about 10-15% below goal	Current high efficiency units close to goal
Installed Cost		Expected to have some cost premium over electric heat pump. For this equipment to be economically viable the incremental cost can not exceed $750-$2,000.				$4,500 - 6,000	Comparable to Electric Heat Pump
Life		Not well defined but judged to have better potential than I.C. engine. For economic viability system life must be comparable to electric heat pump and annual maintenance cost only marginally higher -- i.e. goal of about $200/yr.			To achieve system life comparable to electric heat pump will require major overhauls or engine exchange. Resulting maintenance costs are expected to be relatively high - i.e. 200-400 $/yr.	10-20 yrs.[7]	
Maintenance Cost						$80-$150/yr.[7]	
Commercial Status		FPSE linear actuators available. FPSE heat pump under development.	Automative size engines subject of development and inlimited use. Little experience with small engines. Historically shaft seal perceived to be major problem.	Basic engine technology available. Both specially designs and automotive denvative being adapted to heat pumps.	Large single and double effect machines in use. Limited residential use of small single effect machines. Advanced designs in early stage of development.	Commercially available in residential sizes. Becoming mature technology.	Commercially available in residential size. Becoming mature technology.
Technology Development Trend and Barriers		Need improvement in • design tools • efficiency • coupling of engine to heat pump	Although, work on 15HP heat pump showing good seal performance, seals perceived to be problem.	Improvements in maintenance needed. Solutions to noise and emissions could be expensive.	Development work includes: o New cycles o New fluids	Cost/efficiency optimizations	Improved matching with distribution systems

Notes for Table 6.1.

1. Equivalent Thermal COP = $\dfrac{\text{Delivered Heat (or Cooling)}}{\text{Fuel Flow *HHV} + \dfrac{\text{Electric Power}}{.3}}$

All systems involving freon vapor compression cycle use Westinghouse/ORNL fluid COPs. All system goals, except FPSE, based on approximately 500 watts parasitic power. FPSE parasitics range from 350 watts (MTI advanced heating design) to 1100 watts (G.E. designs).

2. Low end of range indicative of projected performance of current design.

3. Kinematic and I.C. Engines assumed to have brake gross thermal efficiency of 30% (HHV basis).

4. Based on discussions with Mr. R. DeVault of ORNL.

5. Condensing furrnace based on ADL estimates. Air Conditioner COP taken as 10% greater than heat pump cooling COP.

6. ADL estimate based on data from several sources such as references 7 through 13.

7. Based on discussion with Mr. L. Wright at GRI.

$$\text{where } COP_t \text{ - thermal COP} = \frac{\text{Delivered Heat (or Cooling)}}{\text{Fuel Flow *HHV}}$$

$$COP_e \text{ - electric COP} = \frac{\text{Delivered Heat (or Cooling)}}{\text{Electric Input}}$$

For most heat actuated systems, the equivalent thermal COP is about 10 to 20 percent less than the actual thermal COP due to parasitic electric power. For an electric heat pump, the equivalent thermal COP = .3 x electric COP.

Common values of compressor COPs used on the MTI development(1)* and based on Westinghouse work for ORNL(2) are used for all system employing a freon vapor compression refrigeration cycle (i.e., all systems except the absorption and condensing furnace). The electric heat pump COPs are taken directly from this data and the electric air conditioning COPs are assumed to be about 10 percent higher.

The IC engine data and the kinematic Stirling results are based on an engine brake gross thermal efficiency, including combustion losses, of 30 percent.

The free piston Stirling engine results are developed from documents and discussions with MTI and GE(1,3,4,5,6).

The absorption system goals are based on analysis of advanced cycles involving absorption heat exchange cycles where ammonia is the refrigerant and a lithium bromite water mixture is the absorbent(7). Current performance is based on existing single effect heat pumps such as the Arkla ammonia water unit and the Allied roganic machine(8).

The condensing furnace is based on ADL estimates of the performance of the Lennox condensing furnace.

It should be emphasized that all the COP values indicated are for steady state performance under standard design conditions (47°F ambient). The seasonal COP's will be significantly lower for all the systems due to cycling losses and off-design operation of the systems. The impact of these factors could be more significant for all the engine drives systems than for conventional (furnaces, etc.) equipment. However, little effort has been made to date to quantify these impacts and determine how they affect the various engine driven heat pump options.

6.4 Comments on Cost and Life

Installed cost, life and annual maintenance costs are difficult to predict for equipment which is still in the development stage and has not accumulated any actual service experience. Contributing to this uncertainty are

* Reference numbers in parenthesis correspond to special Chapter 6 Reference Section.

the trade-offs betweens installed cost, life and maintenance costs, which have yet to approach optimization in the development equipment. Even where equipment has become a standard item at commerce, estimates from various sources may vary considerably - particularly in the case of annual maintenance costs.

Since the scatter band of cost and life estimates from various sources is large and comparable to the difference which might be expected to exist between the various systems, absolute numerical value have been assigned only to a baseline conventional system (i.e. electric heat pump). Other systems have been described in terms of their departures from these baseline values.

6.4.1 Installed Costs

Based on data from a number sources[9,10,11,12,13] the current total installed cost of an electric heat pump in new construction is estimated as $4,500-$6,000. Where ducting is already in place, the installed cost of an add-on unit would be about $1,500 less (i.e. $3,000-$4,500).

It is estimated that the installed cost of a gas furnace and electric air conditioner would be similar to the heat pump (with a high efficiency furnace the furnace-air conditioning combination may have a slightly higher cost than the electric heat pump).

The various fuel-fired heat pumps either vapor compression cycles powered by Stirling or I.C. Engines or absorption machines -- are almost certain to have a higher equipment cost than the electric heat pump. This differential cost exists because the compression subsystem of the fuel fired systems (engine compressor combination for the engine designs - or generator-absorber - solution pump for the absorption machines) is more expensive than the hermetic motor- compressor unit used in electric heat pumps.

As indicated in Section 5.0, if any of the engine driven systems are to be economically viable their incremental installed cost (as compared to the conventional options) must not exceed to $750 to $2000 range. This allowable incremental cost range is used Table 6.1. It should be noted, however, that it is still unclear whether any of the engine driven options can meet these cost constraints.

6.4.2 System Life

System life for electric heat pumps is estimated at 10-13 years[9,14]. Somewhat longer lifes (i.e. 10-20 yrs.) have been reported for gas furnaces and electric air conditioners.[14] Since the data base for electric heat pumps is probably biased by poor experience with early heat pumps in the 1950's and early 1960's, and since current equipment is much improved, it is reasonable to consider a life and 10 to 20 years for equipment being currently installed - as has been reported for central air conditioners. The gas furnace is also taken as 10-20 years although it may be possible that new high efficiency equipment will have a different set of problems than that of the lower efficiency equipment on which the data is based.

One of the major incentives to develop Stirling engines for heat pump drives is their potential to have longer useful lives and lower maintenance than conventional I.C. engines which represent a readily available alternative.

Limited useful life and high maintenance costs are serious concerns for I.C. engines in heat pump service particularly for residential size units. Although the life of the overall I.C. engine heat pump system may be extended to be comparable to that of electric heat pump systems by means of periodic engine overhauls and/or engine part replacements the resulting levelized annual maintenance costs will be high and, as indicated in Reference 20, will make it difficult for I.C. engines economically to address the residential market.

The life of absorption equipment is estimated by its proponents to be equal to, or possibly slightly greater than, that of electric heat pumps.

6.4.3 Maintenance Costs

Annual maintenance costs are not well defined for any of the heat driven systems and ultimately may be crucial determining economic feasibility.

Electric heat pumps maintenance costs from a variety of sources show a wide divergent due in part to the relatively poor reliability of early units and the improvements which have been made in more recent equipment. Also there is a major difference between maintenance costs for the first 5 years "in warranty" period and those beyond 5 years "out of warranty." Reference based on operating experience up to the mid 1970's[10,15,16] indicate "out of warranty" (or in one case average) annual maintenance costs of $200-250/yr. corrected to 1984 dollars. Two of the References[15,16] indicate "in warranty" costs about 80 and 130 $/yr respectively. Recent extensive work by Alabama Power Co.[17] indicates average maintenance costs of about $60/yr. for the heat pumps in their system. Alabama Power Co. offers service contracts to their customers which for a 3 ton unit would result in average annual costs of $65 during the first 5 years and $100 in years 6 thru 10[18]. A recent report by AGA[9] quotes an annual maintenance cost of $125. GRI estimated average costs of $150-175/yr.[19] As an overall average biased strongly by the Alabama Power work, a range of 80-150 $/yr was selected for Table 6.1.

The combination of a furnace and central air conditioning was considered comparable to an electric heat pump in terms of maintenance costs. However, as discussed in the preceding section high efficiency furnaces could introduce new maintenance elements.

Little data is available on the maintenance costs of fuel fired heat pump concepts. Some elements such as fans and duct filters are common to both fuel-fired and electric heat pumps. However the fuel-fired compression subsystem will have different maintenance characteristics than the hermetic electric motor-compressor.

In the case of absorption equipment the relative lack of moving parts other than the liquid pump could result in reduced maintenance costs.

Two maintenance costs are indicated for I.C. engine systems. The higher number of $400 per annum (Reference 20) is based on manufacturers maintenance recommendations for high quality engines designed with oversized sumps and filters to minimize maintenance requirements. At this level of maintenance requirements the I.C. engine driven heat pumps are not economically attractive for residential applications. The lower figure ($200 per annum) represents GRI goals[19] for I.C. engine driven heat pumps which have not yet been demonstrated.

The Stirling engine has a good potential for low maintenance. The lubricating oil is not in contact at any time with combustion products, therefore eliminating one of the major maintenance items. There are no valves or complex, precise injection, or ignition systems to need maintenance. Potential reliability problems unusual to the Stirling engine include unlubricated piston seals, rod seals, and the heater head. Although long term reliable operation of Stirling engines has yet to be demonstrated, it is projected, that Stirling engines might achieve more favorable maintenance characteristics than I.C.
engines. An annual cost goal of $200 is indicated for the Stirling engine systems which is consistent with one maintenance call per year with minimal need for parts replacement.

6.5 System Comparison

6.5.1 Free Piston Stirling Engine

The free piston Stirling engine heat pump is in the earliest stage of development of all options (except the advanced absorption cycles). Development emphasis is on understanding operation and in improving efficiency - as a prerequisite to field testing of semi-operational units needed to establish reliability and maintenance features.

The projected equivalent thermal COP which is based on the assumption that the FPSE design has the potential to achieve performance comparable to kinematic Stirling and I.C. engine designs shows more than a 100 percent gain over the electric heat pump in the heating mode - but little, if any, gain in cooling. It should be noted that the COP's achieved, and projected, for current FPSE developments are significantly less than these ultimate goals.

Like other fuel-fired designs, installed costs are estimated to be somewhat higher (approximately 20 percent) than the electric heat pump. Life is potentially comparable to that of the electric heat pump, and maintenance costs, while potentially lower than those of some other options such as the I.C. engine, will probably be higher than that at the electric heat pump.

6.5.2 Kinematic Stirling Engine Heat Pump

The kinematic Stirling engine heat pump based on an overall engine efficiency of 30 percent achieves equivalent thermal COPs slightly higher than those of the free piston Stirling engine heat pump. Experience in larger (automotive sizes) and the simpler dynamics associated with the kinematic drive has resulted in the kinematic engine being better understood and its efficiency (at least in larger sizes) better defined than for the free piston. Operating experience has not been extensive enough to define potential reliability. A major concern over the years relative to the kinematic drive is the shaft seal. One developer[21] claims that significant progress has been made in addressing this problem area when applied to a single acting 15 kW system and that operating lives of up to 10,000 hours have been demonstrated. Nevertheless, the role of the shaft seal in determining KSE O&M requirements still appear substantial. Also, it is not clear how serious the problem of scaling to residential size will be - particularly as it affects efficiency and reliability.

6.5.3 Absorption Heat Pumps

New absorption heat pumps represent substantial opportunities over prior designs, although the risks are substantial.

Older single effect equipment was capable of marginal performance advantages over conventional equipment in heating while it was inferior in cooling. New designs involving cycle innovations and new fluid combinations, as yet in very early stage in analysis and development, project performance equal to that of engine driven systems. Based on experience with the prior equipment, is possible that the absorption designs would be quite reliable with long life and relatively low maintenance costs.

6.5.4 I.C. Engine Driven Heat Pumps

I.C. engine driven heat pumps have a potential of good performance with residential field test units approaching the performance goals shown in Table 6.1. Work by Japanese and others suggests it may be possible to score improvements in what had been relatively (and prohibitively) high maintenance costs. On the debit side the I.C. engine may produce relatively high NO_x levels, and a more corrosive exhaust stream than the external combustion designs. Also, noise has traditionally been a concern with I.C. Engines. However, new designs in Europe and Japan have demonstrated successful noise control. The major uncertainty with I.C. Engine driven systems is whether noise reduction, emission control, and low maintenance cost can be achieved without prohibitive increases in equipment capital cost.

6.5.5 Electric Heat Pumps and Condensing Furnaces and Electric Air Conditioners

These two conventional systems represent mature technology where improvements in performance, cost or maintenance are unlikely to be dramatic in the next several years. In cooling, the conventional equipment is roughly comparable to the best potential gas-fired heat pump performance. In heating, gas-fired heat pumps could have significant advantages over the conventional equipment if goals are realized.

6.6 Observations

There currently is no thermally actuated heat pump system which can meet the needs of residential size applications. The I.C. engine driven systems probably are closest to demonstrating this potential. However, it is still uncertain if I.C. engines can achieve sufficiently long life and low maintenance to address residential applications.

- The thermal performance of FPSE heat pumps will, at best, be comparable to that of I.C. engine, KSE, and absorption heat pump units. Even approaching comparability will require, however, significant improvements over demonstrated performance levels.

- All the engine driven heat pump options must meet the same stringent initial cost requirements to be economically competitive with conventional options. These cost requirements have been made more rigorous in recent years due to the increased use of high efficiency condensing furnaces and improved performance of electric driven heat pumps and air conditioning units i.e., the potential for energy savings by using thermally actuated units has been reduced.

- The primary potential advantage for Stirling engine driven systems as compared to I.C. engine driven systems is long life and low maintenance. A demonstrated ability to achieve necessary goals in these areas is therefore critical to their successful development.

6.7 References

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III

DEVELOPMENT ISSUES/PROGRAM OPTIONS

1.0 BACKGROUND

As indicated in Section II, FPSE/HP are still in the developmental stage with few of the essential operating characteristics having been verified experimentally. As a practical matter, the FPSE/HP concept is still in the R&D stage and expectations of an early commercial development program when the programs were being initiated (in the late 1970's) were premature.

The experience with the FPSE/HP development is not unlike that associated with other novel closed cycle thermal dynamic equipment development programs. For example, several of the free piston cryogenic cooler programs have been underway for over 10 years with the efforts directed toward many of the same issues facing FPSE/HP programs such as performance, bearings, seals, dynamics, and controls. The relatively limited progress to date in approaching technical (never mind commercial) viability of the FPSE/HP, therefore, reflects the difficulty in successfully implementing this general class of technologies.

The development of the FPSE/HP concept should be done so as to address most effectively and directly the key barriers which could limit its practical utility. This must be done using the extensive experience gained over the last six years as a base but not be so tied to this experience that major changes in direction are not objectively analyzed.

Section II reviewed the technical status of FPSE/HP developments and quantitatively compared results to date with application requirements. In this section, the major technical issues are summarized in order to form a focus for developing R&D strategies in future program tasks. The issues are divided as follows:

- System Configuration Issues
- System Technical Performance and Analysis Issues
- System Cost and Economic Issues
- Subsystem Issues and Development
    - Bearings and Seals
    - Power Extraction
    - Hot End Components
    - Control

The format used is to make a brief observation on the issue and then to pose a series of questions which may be the subject of future R&D efforts. The program plan options described in section IV have as their objective to address these development issues and questions so that judgements can be made as to the potential commercial viability of FPSE/Heat Pump systems.

2.0 SYSTEM CONFIGURATION ISSUES/PROGRAM OPTIONS

a. Issues

Two basic configurations have received the major attention to date:

- o FPSE - Linear Compressor
- o Diaphragm/Hydraulic Transmission Coupled FPSE

These configurations were selected early in the program and, as a practical matter, there is little flexibility to significantly modify system configuration once the hardware phase is entered into. In comparison, other system arrangements such as:

- o FPSE - Linear Alternator
- o Duplex Stirling
- o Alternative Coupling Mechanisms
- o Kinematic Stirling

have received only modest backing and evaluation from the DOE sponsored FPSE/HP program, although work sponsored by others have addressed them to some extent. In addition, several promising alternatives such as those using bellows to provide for power transmission have hardly been considered at all despite their success in other (albeit smaller scale) applications.

Both G.E. and more recently, MTI have undertaken tasks to review configuration options. The conclusions drawn by these contractors were that the original configuration selected by each were reasonable and, therefore, they recommended no major change in direction to their our programs. However, given the large investments required to develop this technology, it appears prudent to review the whole issue of FPSE/HP configuration options in light of the experience over the past few years and to ensure that this review include inputs from a broad cross section of interested parties.

Questions

Major questions relative to means for coupling a FPSE to a heat pump cycle include:

- o Given the experience of the last few years, do the system configurations which have received the major focus of attention represent the best options for a FPSE/HP system?
- o Should there be a review of Stirling engine/heat pump options in light of such factors as:
    - recent advances in linear alternator technology;
    - improved understanding of Stirling heat pump design;
    - additional experience with bellows based transmissions;

- claims that the shaft seal problem for KSE may be close to resolution for Stirling engines operating at modest (< 1000 psi) pressure levels*;
- recent advances in analytical techniques allowing a better treatment of 3 or 4 body systems with realistic loss mechanisms.

o How important is it to decouple heat pump load dynamics from FPSE dynamics in developing a practical system for operation over a wide range of loads?

o How big a burden does the requirement of new and novel compressor designs place on a FPSE/HP option? or put another way; how big an advantage is it to favor those configurations which can use conventional refrigerant compessors on the heat pump loop?

o If the shaft seal issue asociated with KSE is successfully resolved, would there any longer be a rationale for pursuing the FPSE approach at all?

o Can KSE be scaled down to residential size yet still be cost-competitive?

3.0 SYSTEM ANALYSIS ISSUES/PROGRAM OPTIONS

The analytical tools used to estimate system performance and to provide design guidance have been improving, or at least evolving steadily over the last few years. Nevertheless, most of the analytical tools still require a high level of empirically derived adjustments to obtain good agreement between analyses and experiments. As such, there are serious questions as to the ability of analytical tools to provide the information on which judgements will be made on potentially costly hardware development program - especially for those cases where it is most necessary to rely on those tools, i.e., for configurations or operating conditions which differ significantly from present practice.

As indicated earlier, the questions regarding the adequacy of analyses are even more complex for FPSE as compared to KSE.

In principle, engines with good performance can be designed and built without understanding all the thermodynamic processes. This is evidenced by the conventional I.C. engine, some aspects of which are still not fully understood. However, the FPSE must achieve close to its perceived performance potential to be an attractive candidate for heat pump applications. Additionally, in order to facilitate engine production and increase life it will be desirable to explore a range of engine operational options such as: decreased pressure levels, reduced compression ratios, alternative seal designs, and different operating speeds. It is critical to be able to quantify, by analysis, the effect of these design tradeoffs on system performance levels because establishing the data base empirically would be a long and costly process.

Reasonably accurate analytical models will be necessary to indicate the effect of design changes and, in particular, to provide guidance on how to design FPSE/HP which can be reasonably expected to achieve required performance levels (or to indicate _if_ required performance levels can in fact, be achieved).

Currently available analytical methods are not sufficiently well developed to perform the above functions.

Questions

There is a wide range of questions relative to the performance of FPSE/HP which require more consistent analytical investigation including:

- Do existing analytical tools realistically take into account loss mechanisms so that efficiency level projections are credible? In what specific areas do analytical tools need experimental verification? In what areas do they need improvement.

- Can any of the systems under development be modified to approach acceptable levels of efficiency?

- How would system designs be modified to improve life and efficiency characteristics?

  - Different pressure levels?
  - Different operating speeds?
  - Different couplings to loads?
  - Use of different bearing and seal designs?

- What are the effects of cycling on system performance given the substantial thermal masses associated with the heater heads, air preheaters, combustion chamber walls, and regenerators?

- What is the effect of the above on seasonal performance characteristics?

- Contractor performance _goals_ based on current configurations appear to indicate seasonal performance factors approaching 1.7 in the heating mode. Is this level of performance sufficiently superior to a condensing furnace or other more conventional options to justify this system approach? What further improvements in engine performance would be required to achieve adequate system performance.

4.0 SYSTEM COSTS AND ECONOMIC ISSUES/PROGRAM OPTIONS

a. Issues

The economics of a FPSE/HP system depend on:

- The initial cost of the system;

- The level and cost of annual maintenance;

- Efficiency of operation (i.e., energy savings);
- The useful life of the system;
- Application characteristics; and
- Energy prices

If life and efficiency goals are achieved, it appears that an "engine compressor" manufacturing cost of at most \$300/kW (\$1000 for a 3-ton unit) may be acceptable.

As indicated previously, preliminary projections by contractors indicate a potential to meet cost targets. However, these projections are based on system designs which have not demonstrated a capability to meet technical requirements. Meeting such requirements could require additional costs associated with, for instance, precision bearings, controls, ancillary equipment, and hot end components.

b. <u>Questions</u>

A program to better define the costs of FPSE/HP would have to address a wide range of questionsincluding:

- In systems using gas bearings and clearance seals, how realistically do cost projections reflect the need for high tolerance machining and meticulous care in assembly?
- Is the need for custom and unique transmission/compressor designs adequately accounted for in the cost estimates to date (for example, high efficiency hydraulic systems are known to be costly)?
- What are the cost implications of hot end component designs which can withstand over 25,000 hours of operation at 1300°F+ with cyclic operation?
- What range of O&M strategies can be considered to reduce system costs? For example, can designs which assume periodic replacement of rubbing seals be a realistic possibility and, if so, would this allow for substantial reduction in first costs?
- What are the tradeoffs between initial capital costs, O&M costs, and system efficiency for the technologies under consideration?

5.0 COMPONENT DEVELOPMENT & SUPPORTING R&D

To date, the FPSE programs have focussed their resources on developing integrated systems. A relatively limited amount of separate component development and testing was done as a part of these efforts and then only to fabricate a unit for immediate inclusion in a specific system as soon as possible. As a result, virtually all performance and endurance testing of key subsystems and components has taken place within engines. For this testing mode, obtaining data at the subsystem and component level is often quite difficult and endurance testing is limited by overall engine operating hours.

There are a number of key subsystems and components within a FPSE which can be developed and tested external to the engines themselves. This testing and development approach provides several important advantages.

- The information generated and designs developed can be used to support the development of a multiplicity of FPSE design options (hot end subassembly design and testing might be applicable to a multiplicity of engine/heat pump coupling system options)

- Extensive performance and endurance testing of subsystems and components can be undertaken in a more focussed and cost effective manner than if these elements have to be integrated with a complete engine (which might have other problems reducing testing time and flexibility).

- Subsystem and component level development can occur in parallel with system development and not restrict or be constricted by the progress of this development.

The need for highly focussed subsystem and component level development and testing is usually recognized in system R&D programs such as the development of a FPSE/Heat Pump. The program options outlined below recognize the need for such developments in key areas:

- Bearings and seals;

- Power extraction;

- Hot end components.

5.1 Bearing and Seals

a. Issues:

Bearings and seals play a critical if not dominant role in determining the life, efficiency, and cost of all the FPSE/HP configurations under active development. This fact is not adequately recognized in most programs and bearing and seal issues are usually relegated to secondary status in the evaluation of system design. By contrast, in the design of most thermomechanical equipment, bearing and seal issues have a major effect on overall system design (operating speeds, pressure levels, material selections, etc.). This is, for example, the experience with spaceborne cryogenic coolers having similar technical issues. (Long life, dry helium environment, etc.)

Most FPSE configurations can, in principle, be implemented with either contact bearings and seals or gas bearings and clearance seals, i.e., bearings and seals do not necessarily restrict the selection of overall configuration. However, the approach taken to bearings and seals can significantly impact on manufacturing approaches, efficiency, cost, and the specifics of operating conditions. It is critical, therefore, that this issue be addressed at the subcomponent level to determine which (possibly both) approaches are most likely to be consistent with overall system requirements.

b. Questions

Specific questions which should be addressed by a subcomponent development program in this area include:

- Can any rubbing seal and bearing designs and engine operating conditions result in adequate life and reliability?
- What changes in engine design would tend to improve the prospects for rubbing bearings and seals (reduced pressure levels, lower speeds, etc.).
- Can system designs be developed which allow for periodically replacing rubbing seals similarly as in I.C. engines?
- What are the losses associated with gas bearings and clearance seals of designs reasonable to consider for commercial equipment? What is the impact of those losses on engine efficiency?
- What is the impact of start and stop operation on gas bearing and clearance seal life?
- Which type of bearings and seals are most likely to result in an acceptable combination of efficiency, life, and cost?

5.2 Power Extraction

Issues

The approach used to couple a FPSE to the heat pump load can have a large impact on system efficiency, dynamics, life, and cost. The inertial compressor concept results in a highly coupled system between the compressor and FPSE such that a three (or four) body dynamics with little damping is the result. The advantage of this system is a conceptually straightforward configuration with low viscous or mechanical losses in transmitting power from the engine pressure wave to the freon compressor. The ability to control this system at stable and efficient operating points over a wide range of ambient air conditions must still be demonstrated both analytically and experimentally.

The diaphragm coupled/hydraulic transmission system now under development also has drawbacks, including:

- High friction and viscous flow losses.

- A large number of sliding surfaces and seals with associated potential for life limiting wear.

- The need for active oil inventory control system to account for leakage both within adjacent chamber in the hydraulic system and into the freon working space.

As a result of the above, it is unclear whether this particular diaphram transmission system is consistent with the efficiency and life requirements of the heat pump application.

As a practical matter, therefore, it appears that the means of reliably coupling the FPSE to a heat pump cycle has not been resolved and that the coupling mechanism may be a barrier to developing a credible FPSE/HP concept.

b. Questions

The FPSE/HP should give concentrated attention to the very important questions associated with coupling:

- Is the linear compressor concept a good idea if problems associated with dynamic control can be resolved? For example, if a linear compressor was used with an FPSE of improved design (possibly using active displacer control), would the system become a serious contender?

- What is the efficiency and life potential of the current diaphram coupled hydraulic transmission design given the fundamental friction and fluid flow loss mechanisms associated with the seals and internal bearing surfaces?

- What alternative means of power extraction have the best potential for resulting in efficient, stable, and reliable operation?

    - linear alternator?
    - duplex Stirling
    - Vuilleumier?
    - bellows seal?
    - diaphram and mechanical linkage
    - other

- Should more attention be given to alternative diaphragm or bellows seal arrangements based on the experience of such groups as the University of Washington and Harwell?

5.3 Hot End Components

a. Observations

There are stringent requirements placed on the hot end of an efficient Stirling cycle engine including high temperatures (approx. 1200^{o} F) cyclic temperature fluctuations, high thermal efficiency requirements, (80%) and high heat fluxes. All these requirements must be satisfied with a system which has useful life in excess of 25,000 hours with minimal maintenance and parts replacement.

To date, the performance of the hot end components have not in themselves been a major bottleneck in the development process. Problems of performance and reliability of other parts of the system have been so severe that any problems in the combustor system were comparatively minor and usually did not prevent performance testing or affect engine performance.

However, experience with high temperature combustors and heat exchangers for other applications indicate that the efficiency, life, reliability, emission control, and cost of the hot end components will become important issues once the problems in the engine itself start to be resolved. For example, to date a combustor hot end subassembly which simultaneously meets the stringent emission and efficiency requirements has not yet been demonstrated. The fuel should be combusted with as low as possible air levels to minimize stack energy losses without causing excessive CO. This range of excess air which satisfies these requirements is approximately 25-40%. Carbon Monoxide emissions are regulated by the ANSI standard Z21 1978 for gas fired heating systems. The present limit is 400 ppm in the flue gas. However, since CO is extremely sensitive to manufacturing tolerances, values in the range of 100-200 ppm are generally set as targets to ensure that 90% of all units manufactured meet the ANSI limit. NO_x emissions are not presently limited, however. Targets set by the GRI and California Emissions Standards indicate that NO_x should be less than 100 ppm.

Appendix 3 of the Task 2 Topical Report indicated that the impact of the hot end assembly on annual performance levels (cyclic losses), system life, and costs has only been superficially addressed, and that little endurance and long-term performance testing of hot end assemblies has been undertaken.

b. Questions

A program to identify viable hot end assembly designs which are applicable to a wide range of FPSE/Heat Pump configurations should address such questions as:

- Are there hot end assembly designs which could be used for a wide range of FPSE/Heat Pump options so that this assembly can, in large part, be developed as part of a separate generic development program?
- Can present hot end component designs achieve the life, efficiency and emission requirements of heat pump applications?
- What changes in design are likely to be necessary to meet system performance goals and how will they affect costs (heavier walled construction, use of high chromium and cobalt content metals, implementation of EGR, etc.)?

- What forms of maintenance and parts replacement would be acceptable (for example, replacing nozzles or combustor liners)?

- Would changes in engine design and operating conditions benefit the performance of the hot end components (lower working pressures to reduce stress and heat fluxes, etc.)?

The first question, in particular, is an important one since it may be a poor use of resources to have a multiplicity of system contractors developing individualized hot end assemblies if this is not necessary.

IV

MULTI-YEAR FPSE/HEAT PUMP PROGRAM PLAN

1.0 BACKGROUND

The overall program outlined to address the technical/economic issues discussed in earlier sections has the structure indicated in Figure 1.1. It is divided into three primary areas, but it should be noted that there is a continuous loop of interaction between the first two of them.

Component 1: Supporting Analyses and Model verification

To undertake both technical and economic analyses of FPSE/HP systems via a combination of model development and verification, cost and financial analyses, and market assessments. The output from this task provides the information for the DOE/ORNL program managers to oversee contractor developments, select system approaches for development, and define overall system design and cost goals.

Component 2: Component Development

To analyze, design, and test key components of the FPSE/HP independent of specific system development efforts. The output of this task provides performance information for purposes of systems analyses, model development, and designs for use in system development efforts.

Component 3: System Testing and Development

To develop an integrated FPSE/HP system(s) which the analyses and supporting R&D of Component 2 indicate as showing promise of economic viability.

The close interrelationship between the three major task areas is indicated by Figure 1.1. The resultant program closely parallels that of industry based R & D programs having an objective to develop a commercial product with associated management controls and decision points in order to limit risks.

As indicated in latter sections, the program outlined makes extensive use of past and ongoing activities in the Stirling engine field so that this experience is effectively utilized. In this regard it calls for close coordination between DOE sponsored FPSE/HP efforts, other DOE technology programs, (ECUT), and programs being undertaken with NASA or DOD sponsorship. It also emphasises closely examining relevant experience in related technologies so that the limited resource available to FPSE/HP R & D can be leveraged as much as possible. This experience includes the extensive bearing and seal developments in support of spaceborne cryogenic refrigeration.

This report provides a task by task description of a program leading to the development of FPSE/Heat Pumps over a 5 year time frame. This "baseline development" plan is the recommended scenario consistent with efficient use of funds and time but not driven by either. Section 4.0 indicates the schedule and resources for two alternative scenarios:

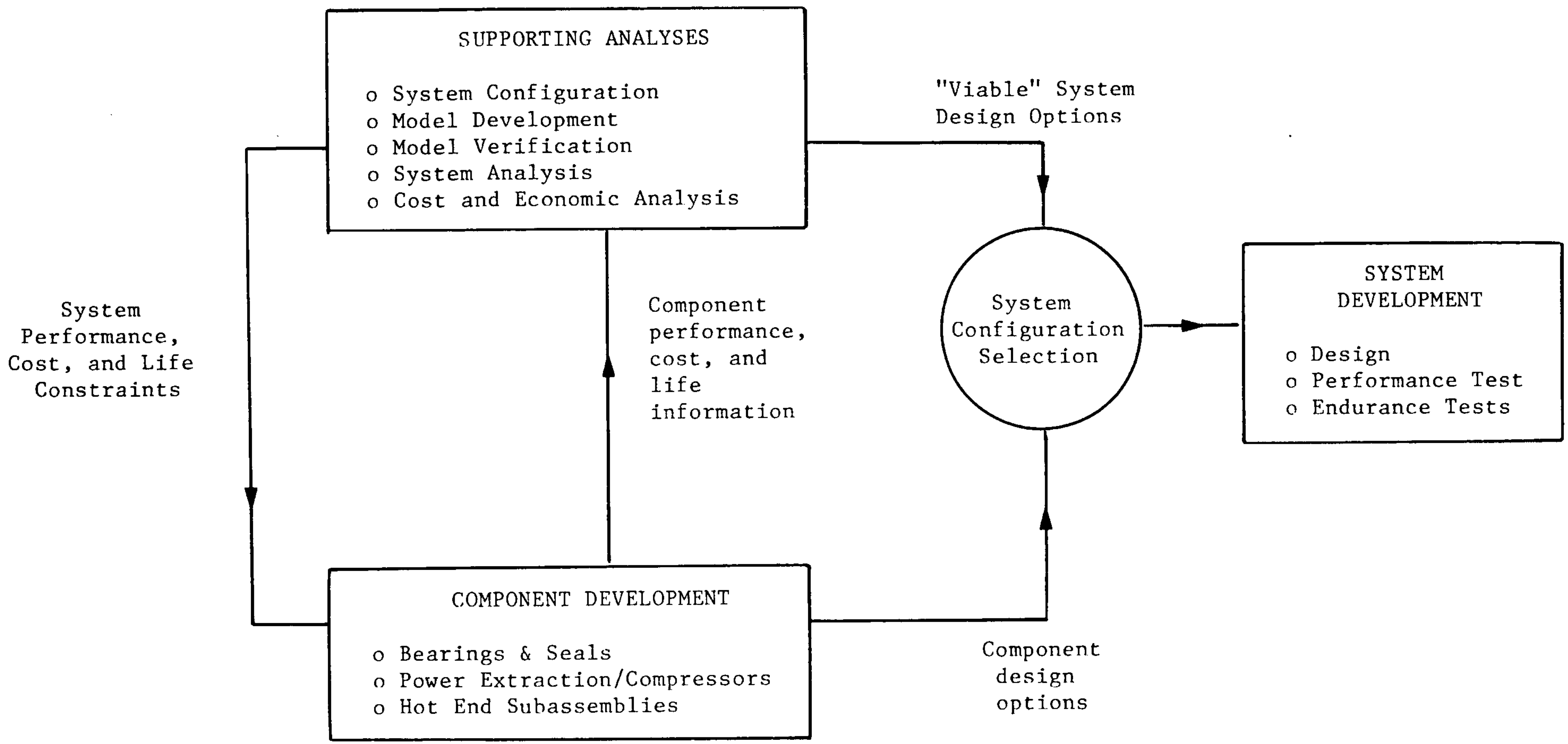


Figure 1.1 OVERVIEW OF FPSE/HP DEVELOPMENT PROGRAM

- Alternative A represents a significantly reduced rate of expenditure of government funds consistent with still pursuing overall program goals.

- Alternative B represents an accelerated program in order to minimize development time which is not restricted by funding rate.

Both alternative programs still assume efficient use of funds which, in turn, implies prudent program management with frequent milestones and reviews.

2.0 PROGRAM PLAN SUMMARY

2.1 Schedule:

Figure 2.1 is an overview of the schedule for the baseline FPSE/HP development program. As indicated, the schedule for the program outlined results in a close coordination of the system and component analyses and design effort which leads to system selection, design, and fabrication within three years, and endurance and performance testing within five years.

Features of the schedule include:

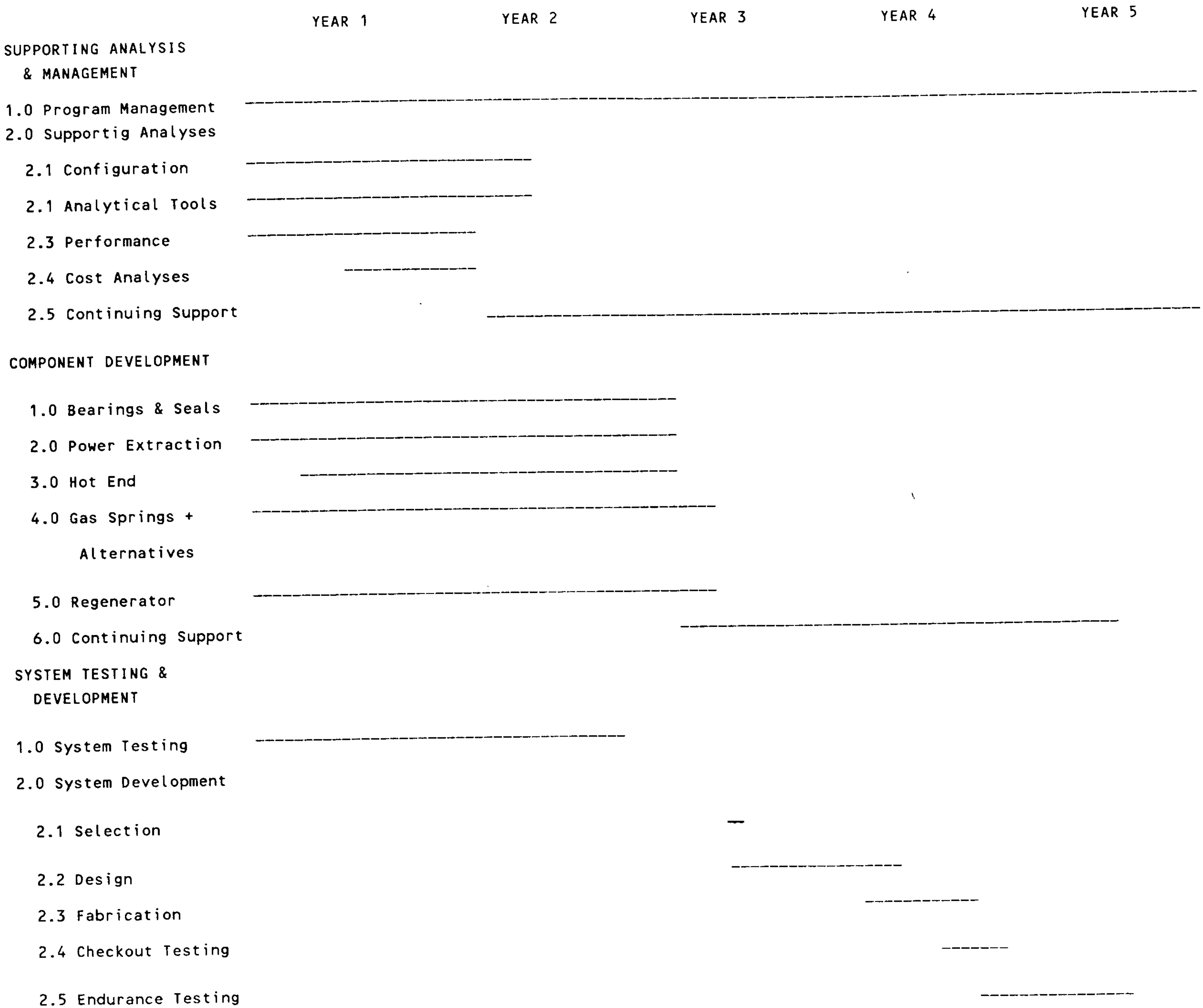
(a) The program is initiated with a review of the configuration options so that the option(s) selected for development can be made in a logical and defensible manner by the beginning of the second year. This review will build off those already undertaken but will have the benefit of more extensive experience in the design, analyses, and operation of a variety of free piston equipment options.

(b) Component development tasks are initiated early in the program to provide information to the system configuration selection effort (what kind of bearings, couplings configurations, etc.) The component developments started initially (bearings, seals, gas springs, etc.) are those where the information might impact on overall system design selection. Other component developments (hot end assembly, etc.) are started about 1 year after program initiation and will support the system design effort. The component development activities are, however, indicated as lasting the duration of the 5 year program. This will allow for endurance testing and design modifications independent of the system effort.

(c) System cost and economic studies are done in parallel with the configuration selection tasks. This is to assure that the selection of a system option is done with a clear, quantitative, understanding of what cost and performance goals are required to result in potential commercial success. This understanding is critical to making judgements on which configurations have favorable risk-reward ratios. For example, a high risk development with potentially attractive cost/performance characteristics might be selected over a low risk option with marginal potential for achieving needed cost/performance goals.

FIGURE 2.1

FPSE/HP DEVELOPMENT PROGRAM SCHEDULE (BASE CASE)



(d) Model evaluation and updating is being initiated early in the program and continues, throughout its duration. One reason for the early start on model updating is the need to provide the organization responsible for configuration selection with the tools to independently assess performance potential and identify development risks and uncertainties.

Model verification and refinement is supported by the system tests, oscillating flow tests, and the component development efforts.

(e) The renewed system development efforts start about two years into the program. At that time, information from the supporting analysis and component development tasks is available to make decisions on which systems to pursue. The baseline plan assumes that two system approaches are pursued in parallel through system hardware design, assembly and testing. These would probably involve significantly different approaches to FPSE/HP coupling and, possibly, to the approach taken to bearing and seals.

Because major decisions affecting configuration, means of coupling to the heat pump cycle, and component design will have been made, the system design and fabrication effort can be done in 18 months. An additional two years are indicated for performance testing, design modifications, and endurance testing. By the end of this period, sufficient information will be available to determine if commercial development is warranted.

2.2 Program Resources

Figure 2.2 summarizes the resources required by the baseline programs. For Component 1, both manpower and overall financial resources are indicated. Many of the tasks within this component would be done by DOE/ORNL staff and this division indicates the total level of staffing required. For the other components only the financial resource estimates are indicated. The tasks within these components will be done in large part by contractors and involve a complex mix of purchased equipment, professional staff, and specialized fabrication. The estimates indicated are based on experience with similar hardware design, fabrication, and testing programs.

The Analytical and Management Support function requires the equivalent of 6 professional staff throughout program duration. During the first year and a half the staff is heavily engaged in configuration evaluation, model and refinement, and system requirements definitions. Once the system development work is initiated, a longer portion of staff time is devoted to program management and evaluation functions.

The major financial resources of the program are seen to be associated with the Component Development and System Testing and Development components.

These two components require about $2 million per year for the first two years increasing to over $4 million per year once the new systems are being fabricated and tested. These cost figures assume that two system options are pursued in parallel as part of the baseline program. The total resource requirements are about $18 million over the course of the five year development program.

FIGURE 2.2
BASELINE PROGRAM - BUDGET SUMMARY
($ x 10^3)

	YEAR 1	YEAR 2	YEAR 3	YEAR 4	YEAR 5	SUMMARY
SUPPORTING ANALYSIS & MANAGEMENT						
Program Management	300	300	300	300	300	1,500
Supporting Analyses	1,600(1)	600	400	150	150	3,020
COMPONENT DEVELOPMENT	1,300	1,600	1,050	600	400	4,950
SYSTEM DEVELOPMENT & TEST						
Endurance Testing(2)	800	600				1,400
System Design, Fabrication and Test(3)		1,000	3,500	1,600	1,200	7,300
ANNUAL TOTALS	$4,000 K	$4,100 K	$5,300 K	$2,250 K	$2,050 K	18,170

1. Includes 500 K in test equipment for model verification
2. Assume utilization of currently available equipment
3. 2 Systems in parallel

2.3 Alternative Program Scenarios

2.3.1 Reduced Funding Levels

Major assumptions associated with this scenario are:

- All components within the program are stretched out on schedule
- Only one system is pursued in the system testing and development task.

The resultant schedule indicates a 6 year development program with decisions to pursue a renewed system development program deferred until the end of year 2.

The deferement allows more time for system analyses and evaluation as well as for more experimental results from the component and system testing efforts. Nevertheless, the potential for pursuing two divergent design philosophies into the system development stage is lost which places more pressure on the judgements involved in the system selection process.

The annual expenditure rate is reduced by over 50% from the baseline program with the total program cost being $9 million.

2.3.2 Accelerated Program

Major assumptions associated with this scenario are:

- The system selection process is accelerated so that it takes place at the end of 12 months.

- Four system options are initially pursued in parallel through the design phase. Only two systems are pursued into the fabrication and test phase. This allows for additional analyses and experimental data to be brought to bear before making decisions on the more costly system development activities.

The resultant schedule indicates a 3.5 year development program.

The annual expenditure rate is increased by over 50% from the baseline program. However, total program cost is increased by only 25% to $20 million due to its shorter duration.

SECTION 3.0 BASELINE PROGRAM PLAN

3.1 COMPONENT 1: SUPPORTING ANALYSIS AND MANAGEMENT

The management of the FPSE/HP program will rely on the Supporting Analysis Program Component to develop analytical tools for system evaluation, define system performance and cost goals with which to compare development programs, and to help identify which system options show the most promise of meeting program goals. This information is a necessary input to the DOE/ORNL program managers in order to be able to work constructively with private sector contractors, to be able to manage and evaluate their efforts, and to plan program initiatives which effectively utilize resources. This Component consists of the following major Task areas.

Task 1.0: PROGRAM MANAGEMENT
Task 2.0: SUPPORTING ANALYSES
  Task 2.1: CONFIGURATION ASSESSMENT
  Task 2.2: ANALYTICAL TOOLS REFINEMENT AND DEVELOPMENT
  Task 2.3: SYSTEM PERFORMANCE AND ECONOMIC STUDIES
  Task 2.4: SYSTEM COST ANALYSIS

Each of the tasks in the Program Analysis and Management Component is described in the following section.

A. WORK PLAN OUTLINE

TASK 1.0 PROGRAM MANAGEMENT

The FPSE/HP program will involve the participation of a multiplicity of both public and private sector organizations including:

- o DOE
- o ORNL
- o NASA
- o ANL
- o Universities
- and o Contractors

It is essential that this multifaceted program be effectively managed as per work output, schedules, and budgets if effective use is to be made of government resources. In this plan it is assumed that this management function is performed by the DOE designated support contractor.

The program management activity is discussed as a separate task to highlight its importance to program success and to indicate that specific personnel and financial resources must be directed toward this function.

This activity consists of the following subtasks:

Task 1.1: Program Planning and Review

The most important function of the program management team will be to prepare detailed annual (AOP) and multiyear program plans clearly delineating activities, resources, and schedules. The program plan will be periodically reviewed (usually on a quarterly basis) and modified to reflect ongoing technical experience, changes in policy, and financial resources.

Task 1.2: Technical Review and Evaluations

Technical/economic information will be generated throughout the program due to both in-house and contractor programs. The program management team will review the outputs of these technology tasks, provide direction for its continuance, and use the information to make program decisions.

Task 1.3: Contract, schedule, and Financial Management

The resources of the program will be utilized by a multiplicity of organizations. The program management subtask requires careful control of all these activities to assume adherence to contract terms, resolve issues pertaining to schedule and budgets, and identify problem areas requiring resolution.

As indicated above, the program management function requires a team with a first hand understanding of the full range of technical/economic issues associated with the FPSE/HP program in order to be able to effectively undertake the various planning, evaluation, and contractual management functions.

TASK 2.0 SUPPORTING ANALYSES

TASK 2.1 Configuration Assessment

The DOE support staff will undertake an analysis and review of alternative configurations with the assistance and support of private sector contractors and, possibly, government laboratory staff (NASA, etc.). This approach will include having the joint DOE Support Staff/contractor team;

- Define a set of promising FPSE/HP configurations based on past experience.
- Identify major technical and cost issues associated with each option.
- Discuss the configurations options with organizations (companies, etc.) with special background in the technologies involved.
- Assess development risks associated with the different system options.
- Rank the options as to their probability of meeting application requirements.

ORNL. the current DOE support contractor, has staff with a background in FPSE technology which can form the core of the evaluation team. This team will be supplemented by contractor staff and other government laboratory staff having some combination of experience to:

- Undertake performance analyses (possibly enhanced by outputs from the Systems Analyses Program).

- Address key issues affecting life and reliability (bearings, seals, hot end assemblies, etc.).

- Develop preliminary cost estimates (at least on a comparative basis) of different configurations.

- Estimate performance characteristics and assess development risks.

- Rank the options as to their probability of meeting application requirements.

Based on the above, recommendations will be made to the DOE program manager as to which system options (two in the baseline program) should be pursued into the hardware development phases. The review process leading to these recommendations will make use of analytical and experimental information as it is developed by the supporting analyses and component development tasks.

The configuration review function is an ongoing process such that the initial system design selections are periodically assessed as additional data is generated.

TASK 2.2 Analytical Tools Refinement and Development

The development of improved analytical models will be undertaken via three subtasks:

1. Review of Analytical Models/Develop Unified Approach

2. Model Improvement

3. Testing/Model Verification

In addition to the above, component testing and development will be taking place which provides important data for the model development. These include:

- Oscillating Flow Tests/Regenerator Development

- Gas Spring Assessment

- Bearing and Seal Analysis and Development.

Task 2.2.1 Review Analytical Models/Develop Unified Approach

There are a number of models of FPSE's which have been developed over the last few years. Several of the more detailed of these are proprietary to companies in this field, including those of MTI and Sunpower. In addition, government organizations such as NASA Lewis, Argonne National Laboratories, and ORNL have, or are in the process of developing, models of their own.

Since the government will be providing a major portion of the financial support for FPSE developments, it is important that it can undertake independent analyses of system options in its decision making processes and be able to verify projections made by development contractors.

To this end, this program will concentrate its resources in a single coordinating organization to review current analytical models and identify major gaps and uncertainties. Subject to constraints imposed by the proprietary nature of the industrial models, a review will be undertaken of currently used models (computer based and otherwise) with particular attention given to how they deal with known areas of potential uncertainty such as:

- o Transient heat transfer losses in cylinders and springs.
- o Gas bearing and seal losses in engines and gas springs.
- o Oscillating flow losses.
- o Oscillating flow heat transfer effects.
- o Gap losses.
- o Dynamic/Thermodynamic interactions.
- o Internal heat losses (conduction, radiation, etc.).
- o Regenerator effectiveness

Based on this review, a specific course of action will be developed to improve analyses capabilities.

Task 2.2.2 Model Improvement

Based on the recommendations of Task 2.2.1, public sector FPSE/HP analyses and computer models will be updated and improved to reflect the best state of the art knowledge available at the time. This will be an ongoing program that will make use of new information and data as it is generated by specific system and component testing.

Task 2.2.3 Engine Tests/Model Verification

There will be an ongoing program to update and improve analytical techniques via verification with engine testing. The engine system should be a "workhorse" system capable of being easily modified to test the effects of:

- o Regenerator Designs
- o Operating Speeds and Pressure
- o Operating Temperature
- o Bearing and Seal Designs
- o Power Extraction Options

Through testing, the range of applicability of analytical models can be better defined and specific areas for further work identified. This work could use the recent experience at NASA Lewis in testing the RE-1000 engine as a basis for a consistent, long-term, effort in model verification.

Also, the use of a Linear Alternator Dynamometer should be considered to provide flexibility in simulating a wide range of loads of potential interest.

TASK 2.3 System Performance/Economic Studies

Task 2.3.1 Seasonal Performance Studies

During this task, ORNL staff will analyze the seasonal performance of FPSE/HP based on a reasonable range of heating and cooling design point COP's. These analyses will take into account both off-design engine heat pump performance and the effect of thermal cycling. Possibly using some form of "bin-method" seasonal performance of FPSE/HP options will be estimated for several (5-10) regions of the country with sufficiently high heating loads to make heat pumps of practical interest. These studies will examine the role of engine design (thermal mass, etc.) and control strategy in influencing seasonal performance. Operational strategies which will be considered include:

- On-off operation through cycles similar as with electric driven heat pumps.

- Full or partial capacity modulation to reduce cycling losses.

- Longer cycle periods (to reduce short term cycling) by virtue of employing short term thermal storage.

The output of this task will be: (a) estimates of fuel savings for the different operating strategies; and (b) estimates of the impact of engine and system design options on seasonal performance factors.

Task 2.3.2 Economic Performance Estimates

Information from Task 2.3.1 will be used to estimate the economic performance of FPSE/HP options in different regions, and for a range of building types as measured by such parameters as annual energy cost savings. The "conventional" systems against which the FPSE/HP will be compared include:

- o High Efficiency Furnaces/Electric Air Conditioning

- o Electric Driven Heat Pumps

The output of this task will be estimates of energy cost saving relative to likely conventional systems using both current energy costs and future energy costs projections as provided by both government and industry sources. This information will be used to estimate the incremental cost a FPSE/HP can have as compared to conventional options using a range of commonly accepted payback period criteria.

Task 2.3.3 FPSE/HP Cost Constraints

Information from Task 2.3.2 will be used to define a range of acceptable installed (and manufacturing) cost for the FPSE/HP system. This will be done by:

- Determining the cost structure of the conventional options on both a system and subsystem level.

- Defining which subsystem the FPSE/HP has in common with the conventional options (heat pump, heat exchangers, air blowers, etc.).

- Defining which subsystems in the conventional system such as a condensing furnace, are replaced by the FPSE/HP (and constitute a credit).

- Defining the additional components associated with the FPSE/HP system (FPSE, compressor, exhaust gas heat recovery heat exchanger, etc.).
- Using the above information to determine what cost structure of the incremental equipment associated with the FPSE/HP can have and still be consistent with the output of Task 1.2.

This latter information will be generated based both on current energy costs and those which might exist at the time such equipment could reasonably be expected to come to market (1990's).

TASK 2.4 System Cost Analysis

Task 2.4.1: System Component Definition

In this task, the subsystems comprising a range of potential FPSE/HP system will be defined and these incremental components identified. Depending on system configuration, these could include:

- FPSE unit
- Compressor assembly
- Heat recovery units
- Supplemental burners
- Safety and control equipment
- Special mounting and insulation equipment
- Energy storage unit (for cycling control).

Two or three of the more promising options will be broken down as indicated above for cost analysis purposes.

Task 2.4.2: System Cost Analysis

Cost estimates will be made for the system defined in Task 2.4.1 at the component level and then at the system level. Costs will, in general, be those at the system assembly and packaging level since these are most readily obtainable for currently available equipment.

The highest levels of uncertainty will be associated with the FPSE and any unique compressor assemblies required by the system. These cost will be estimated based on appropriate combinations of system contractor projections and bottoms-up estimates based on subsystem configurations. The resultant system cost structures will then be analyzed and compared with allowable cost structures as indicated by Task 2.3.3.

Task 2.4.3: Cost Issues Identification

The results of previous tasks will be used to identify the major factors influencing system costs and to assess which system options have the best chance of achieving cost goals. Special attention will be given to defining areas of major cost uncertainty which need to be addressed by the R&D program. In the FPSE themselves, these could, for example, include gas bearings/clearance seals and hot end subassemblies. On a system basis, the cost of heat recovery, energy storage, and supplemental firing may require additional attention.

Task 2.5 Continuing Analytical Support

The specific analytical support activities discussed relative to Tasks 2.1 through 2.4 are focussed on making early management decisions on such critical issues as system configurations and performance requirements. However, the need for analytical support inputs to the program management function will be a continuing one as new information is generated and managerial decisions undergo periodic review. In general, this ongoing analysis will deal with the same issues as tasks 2.1 through 2.4, however the relative importance of the various issues is difficult to determine ahead of time. For this reason, this program plan assumes a separate task of continuing analytical support and accounts for the important activity both in the schedule and in program resource estimates.

B. SCHEDULE

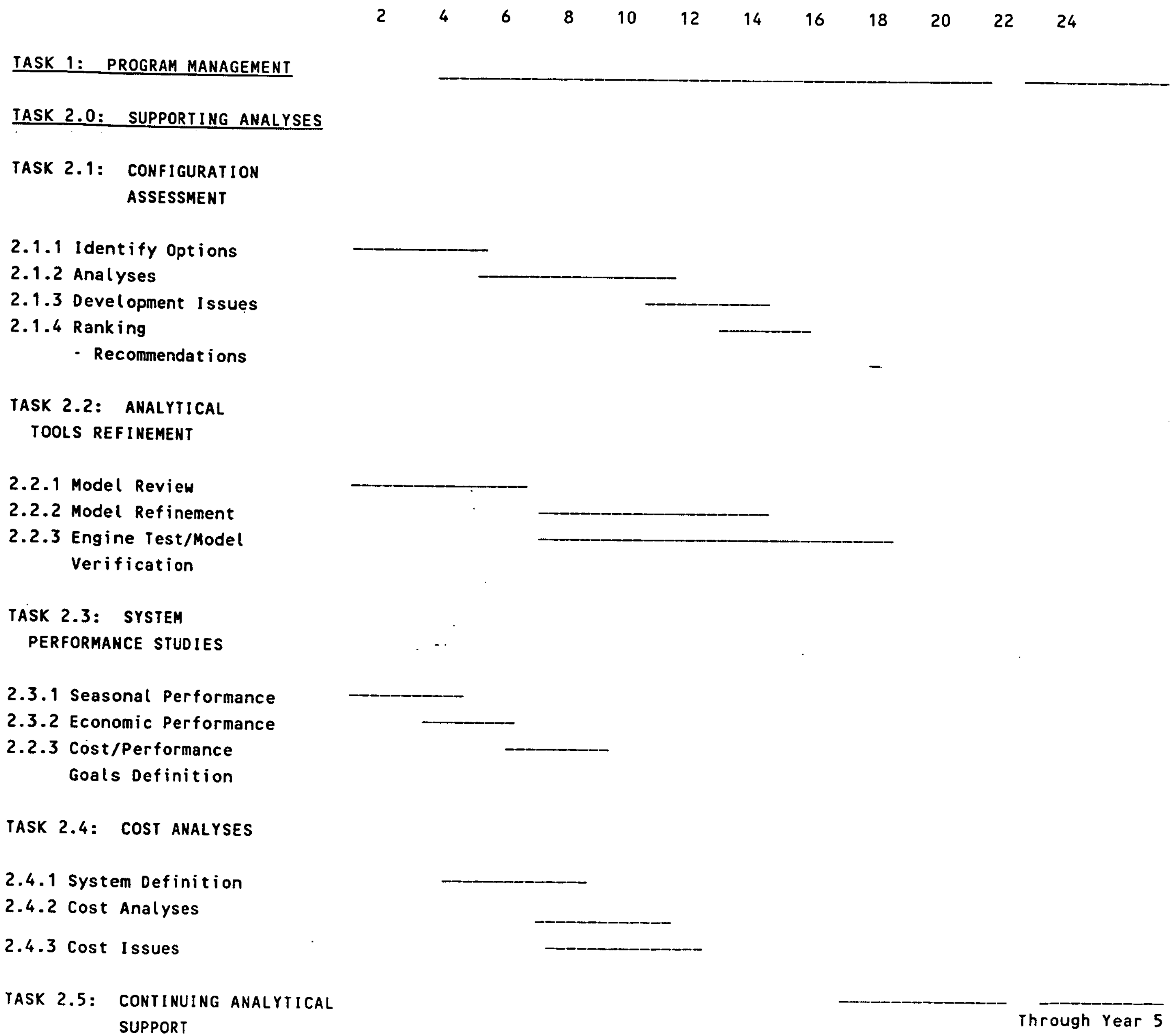
Figure 3.1.2 shows the overall schedule for the Supporting Analysis Component. The management tasks are ongoing efforts throughout the program. The solid lines for supporting analyses indicate a focussed effort within each task resulting in outputs needed for system evaluations and management R&D decisions over the first 18 months. The dashed lines indicate that the supporting analysis function is an ongoing activity throughout the program - albeit at a lower level of effort than during the first year and a half. Several features of the schedule are:

- The Configuration Assessment task results in a set of recommendations on how best to use resources in a system development effort within 18 months.

- The Analytical Tools Refinement task is indicated early in the program so it can provide some guidance in the Configuration Assessment task. This activity will be a continuing one and will provide improving capabilities to verify and update decisions made in Task 2.1, as the program progresses.

- The System Performance Studies task will provide an updated set of system performance and cost goals about 9 months into the program against which systems being assessed in Task 2.1 can be compared.

FIGURE 3.1.2
SCHEDULE: SUPPORTING ANALYSES AND MANAGEMENT



- The Cost Analysis Task will identify these subsystem and components with major cost uncertainties and impacts which merit particular attention in the Configuration Assessment as well as in future R&D programs.

C. LEVEL OF EFFORT/BUDGET ESTIMATES

The estimated level of effort for the program is indicated on Figure 3.1.3. This effort is divided between professional staff, support staff, and expenses. There is as yet no devision made between ORNL, government laboratory, and contractor staff made in undertaking this effort. It is assumed that on a man-year basis, the details of program implementation will not significantly impact the overall resources required.

Program management requires the equivalent of 2 staff per year. In the first year the program management function will focus on reviewing information leading to major hardware development decisions. In subsequent years, considerable effort will be required to evaluate and provide direction for contractor development and testing efforts.

The overall level of effort indicated for supporting analyses for the first year is 6.5 person-years with the major costs associated with model refinement and development functions. The resources required drop to 3.5 person-years by the second year, and to 3 person-years for subsequent years due to the aforementioned need to provide continuous analytical support to review and update the management decision process.

3.2 COMPONENT 2: COMPONENT DEVELOPMENT

To date, the FPSE programs have focussed their resources on developing integrated systems. A relatively limited amount of separate component development and testing was done as a part of these efforts and then primarily to fabricate a unit for immediate inclusion in a specific system as soon as possible. As a result, most performance and endurance testing of key subsystems and components has taken place within engines. For this testing mode, obtaining data at the subsystem and component level is often quite difficult and endurance testing is limited by overall engine operating hours.

There are a number of key subsystems and components within a FPSE which can be developed and tested external to the engines themselves. This testing and development approach provides several important advantages.

- The information generated and designs developed can be used to support the development of a multiplicity of FPSE design options (why should hot end subassembly design and testing be done separately for each engine system option?)

- Extensive performance and endurance testing of subsystems and components can be undertaken in a more focussed and cost effective manner than if these elements have to be integrated with a complete engine (which might have other problems reducing testing time and flexibility).

FIGURE 3.1.3

BUDGET ESTIMATES* - SUPPORTING ANALYSES AND MANAGEMENT

YEAR	1	2	3	4	5	TOTAL
1.0: PROGRAM MANAGEMENT						
o Personnel (person-years)	2	2	2	2	2	
o Resources ($)	300 K	300 K	300 K	300 K	300 K	$1,500 K
2.0: SUPPORTING ANALYSES						
2.1 Configuration Review						
o Personnel	1.5	0.5				
o Resources	200	70				
2.2 Analytical Tools						
o Personnel	3	2	2			
o Resources	1,100	400	400			
2.3 System Studies						
o Personnel	1					
o Resources	150					
2.4 Cost Analyses						
o Personnel	1					
o Resources	150					
2.5 Continuing Support						
o Personnel		1	1	1	1	
o Resources		150	150	150	150	
SUPPORT TOTAL						
o Personnel	6.5	3.5	3	1	1	
o Resources	1,600	670 K	450	150	150	$3,020 K
				TOTAL		$4,520 K

* Constant 1985 dollars

- o Subsystem and component level development can occur in parallel with system development and not restrict or be constricted by the progress of this development.

The need for highly focussed subsystem and component level development and testing is usually recognized in system R&D programs such as the development of a FPSE/Heat Pump. The Component Development activity consists of the following Task areas:

Task 1.0 Bearings and seals;
Task 2.0 Power extraction/compressor design;
Task 3.0 Hot end components;
Task 4.0 Gas Springs and Alternative Spring Designs;
Task 5.0 Regenerator Development/Oscillatory Flow Tests.
Task 6.0 Component Development - Continuing Support

Each of these task areas is described in the following sections:

A. WORK PLAN OUTLINE

Task 1.0 Bearing and Seals

Task 1.1: Contact Bearing and Seal Review

There is a great deal of experience with contact bearings and seals resulting from developing cryogenic coolers for space applications. Additionally, there are analytical tools for estimating wear between rubbing surfaces which, when combined with manufacturers data, can provide initial guidance on wear rates and the factors (pressure, temperature, velocities, etc.) which affect wear.

During this task the prior experience with rubbing seals in environments similar to those in Stirling engines would be reviewed in order to:

- o determine under what conditions (if any) contact bearings and seals have lifetimes of potential interest;
- o identify material combinations which are likely to have the best performance in Stirling engine applications;
- o determine what forms of seal testing appear to provide the most useful information;
- o determine how useful analytical tools are in estimating wear rates of seal material combinations.

One of the outputs of this task will be a recommendation to the program manager as to whether contact bearings and seals show reasonable promise to be able to satisfy FPSE/HP requirements based either on:

- A potential to achieve needed life without replacement assuming proper engine design and operating conditions.

- A potential to develop FPSE engine designs allowing for seal replacement at reasonably long intervals (8,000 hours+)

Assuming that contact bearing end seals show such promise the program will proceed to the following tasks.

Task 1.2: Contact Bearing & Seal Testing Program

The wear rate on contact bearings and seals is known to be highly application specific. Analyses and related experience can provide initial guidance on acceptable ranges of material combinations, designs, and operating conditions. However, only extended testing under conditions which closely approximate those expected in the actual system can provide the data needed for design purposes.

Consequently, this task would consist of:

1.2.1 Test System Design and Fabrication

During this subtask, information from Task 1 would be used to design seal and bearing testing rigs which can closely simulate conditions in FPSE. A multiplicity of these rigs would be built to allow endurance testing over a range of operating conditions of possible interest, and with a variety of material combinations showing initial promise of meeting application requirements.

1.2.2 Material & Operating Conditions Screen Testing

Initially a number of materials would be tested for intermediate durations (50-200 hours) to identify those material combinations and operating conditions where wear rates are sufficiently low to show promise of meeting life and reliability requirements.

1.2.3 Contact Bearing & Seal Endurance Testing

The material combinations identified in Task 2.2 under simulated operating conditions will be subjected to extended endurance testing (1000 hours +) to determine which, if any, might have necessary life characteristics.

Task 1.3 Contact Bearings and Seals System Impacts

The prior two tasks will indicate which range of seal operating conditions might be compatible with low wear and, hence, long life. These conditions include:

- temperature
- pressure
- velocities
- stress reversals
- material surfaces

This information will be used to identify possible FPSE design modifications which would enhance the prospects for using contact bearings and seals such as:

- o cooler wall temperatures in the vicinity of seal contact
- o lower gas pressure levels (with associated lower seal contact pressure)
- o reduced operating speed

The extent to which required engine sizing and operating parameters are still consistent with other system imperatives such as low cost and high efficiency will be assessed to determine if contact bearings and seals are realistic options for this application class.

Task 2: Power Extraction/Compressor Designs

Task 2.1: Hydraulic Transmission Studies

Several of the options for coupling FPSE to the heat pump cycle involve hydraulic transmissions e.g., the MTI system currently under development and several of the University of Washington concepts. Although different in detail, these systems involve similar issues regarding the behavior of hydraulic systems with oscillating flows, sliding bearings and sliding seals in the hydraulic fluid, and barrier seals separating the transmission from the engine working space. This task would include:

Task 2.1.1: Performance Analyses

The performance characteristics of several of the hydraulic coupling options would be analyzed in order to:

- Provide estimates of probable efficiency levels under a range of load conditions realistically taking into account.
  - -- compressibility effects
  - -- inertia effects
  - -- viscous and leakage losses in the bearings and seals
  - -- flow losses
- Examine the design options relative to bearings and seals and their effects on costs (see Task 1.2), efficiency, and life. This latter issue will be given special attention to determine if wear in the transmission is likely to be a significant life limiting factor and/or whether it will have significant efficiency impacts over the long-term (due to increasing leakage losses).

The results of these analyses will be generalized as much as possible to be applicable to this class of transmissions to provide initial guidance in the design and evaluation of alternative concepts involving hydraulic coupling between the FPSE and the compressor.

Task 2.1.2: Cost Analyses

Hydraulic coupling systems can impact costs in several ways including

- initial cost
- life (replacement costs)
- maintenance costs

To date the effect of the hydraulic transmission options on overall costs has not been addressed in sufficient detail, despite the large reservoir of experience available in the manufacturing costs, life, and maintenance of high precision hydraulic equipment in industrial applications.

Consequently, in this task the probable cost implications of various hydraulic coupling options will be examined using the industrial experience with such equipment as a basis. The output of this task will indicate:

- o The probable range of manufacturing costs given the precision requirements of high efficiency equipment.

- o The probable useful life of hydraulic equipment of the designs contemplated, and the associated impacts on overall economics.

- o The nature and costs of O&M associated with hydraulic transmissions.

Task 2.1.3 Experimental Verification

Existing hydraulic coupling systems will undergo both performance and endurance testing to verify the performance projections of Task 2.1.1. The equipment will be well instrumented to better identify and quantify specific loss mechanisms for comparison with analytical projections.

Endurance testing will be a particularly important aspect of the testing program in order to determine if efficiency remains at acceptable levels over extended periods of operation.

Task 2.2: Alternative Transmission/Compressor Designs

The previous task focuses on FPSE - heat pump cycle coupling via hydraulic means. This reflects the large effort directed over the last few years by several organizations toward this option and the resultant availability of systems for testing and analyses. However, other means of coupling FPSE to heat pump cycles will be explored early in the program since hydraulically coupled arrangements may not meet the combination of performance, life, and cost constraints imposed by heat pump applications. As noted in Section II, these alternatives could include:

- Inertial compressors
- Duplex Stirling
- Veullimeir cycles
- Linear Alternators

- Direct mechanical coupling via bellows

During this task, these alternative arrangements will be evaluated by undertaking the following tasks:

Task 2.2.1 Inertial Compressor Review

Despite previously described difficulties, the G.E. program indicated that the inertial compressor system showed promise of resulting in a relatively straightforward (mechanically) coupling between a FPSE and a vapor compression heat pump cycle. The experience with the G.E. program will be reviewed in more detail to assess whether there is a reasonable chance that this concept could be redesigned to have necessary performance and cost characteristics. Issues to be addressed by the review will include:

- Probable sources of poor performance resulting from inclusion of the second gas spring.
- Possible changes in overall design parameters, such as operating frequency, which could reduce size or improve performance.
- Options for achieving stable operation over a wide operating range.
- Potential benefits of using a driven displacer to facilitate control and reduce operational uncertainties.

The initial review will be based on experience plus limited calculations to make judgments on the potential for the inertial compressor concept to meet program goals if appropriate design changes were dmade. If these initial judgements indicate promise, the inertial compressor concept will be included as part of a more general effort to solicit new concepts for FPSE/HP arrangements from industry.

Task 2.2.2 General Review and Assessment - Alternative Coupling Systems.

This task will include a general solicitation from industry (probably via an RFP) of different options for coupling FPSE to heat pump cycles. This activity will proceed in three stages.

The most promising (1 or 2) of these options based on the results of the conceptual design studies will be selected for inclusion in the system design fabrication and test activity (Component 5).

<u>Task 3: Hot End Components</u>.

A structured program to address the issues of the hot end subassemblies could include the following tasks:

Task 3.1: Seasonal Performance Impacts

The thermal aspects of hot end subassembly design options should be analyzed to provide thermal information necessary to undertake seasonal performance analyses per Task 2.3 of Component 1. These analyses will include:

- Estimates of the thermal mass of components which take part in heat-up and cool-down cycles.
- Estimates of thermal losses from the combustion chamber and elsewhere during system operation. These estimates would include thermal losses occurring during the on and off cycles associated with part load operation.

This information, when combined with seasonal performance models, will allow assessment of the impact of the hot end subassembly on seasonal performance and identification of designs and operating modes which reduce cycling losses.

Task 3.2: Emission/Efficiency Analysis

The combustion system/hot end assembly must simultaneously meet requirements:

- o Result in an efficiency of over 80% for all important load conditions.
- o Result in acceptably low emission levels.

As indicated in the referred report, simultaneously meeting these conditions is made difficult by the high levels of air preheat required and the associated high flame temperatures. In this task, the design of and experience with combustion/hot end heat exchangers will be evaluated and measures identified for improving performance. Such measures could include exhaust gas recirculation (EGR) and combustion gas recirculation (CGR). The impacts of measures for reducing emission levels on efficiency will be closely examined so that design options which represent appropriate compromises between the conflicting performance requirements can be identified.

Task 3.3: Performance/Endurance Testing

A hot end assembly design which shows promise of meeting performance requirements will be assembled and subjected to both performance and endurance testing under conditions similar to those in an engine. This system will be properly instrumented in order to:

- o Measure emission levels during steady state and transient operation.
- o Measure efficiency levels during steady state and transient operation.
- o Measure thermal mass effects during cyclic operation such as would occur during system operation.
- o Measure changes in efficiency and emission levels after extended periods of operation.

- Identify components which influence life or reliability.

Task 3.4: Manufacturing Cost Analyses

Cost estimates by GE and MTI are probably optimistic given the components employed in the hot end assembly. These estimates assume technical developments and production rates inordinate with current technical status and potential market share for the Stirling heat pump. A manufacturing cost analysis will be completed to develop realistic cost estimates for the hot end component for a range of production rates consistent with the potential market share of the Stirling heat pump. The cost estimates will be based on detailed manufacturing cost analysis for key components such as:

- Special blower for hot exhaust as used with the MTI prototype;
- Two-stage air flow rate controller if two firing rates are used; Also being addressed in automotive program
- Air preheater and cost of its laborious assembly (including possible use of ceramics such as being developed for automotive systems); and
- Expensive high chromium cobalt content metals used in heater head.

Task 4.0 Gas Springs and Alternative Spring Designss

4.1 Gas Springs

4.1.1 Gas Spring Experience Review.

There has been experience with analysing and measuring gas spring issues for over 15 years as a result of Stirling engine and spaceborne cryocooler projects. This prior experience will be reviewed in order to identify major areas of uncertainties - particularly as applied to the conditions of Stirling engines. These uncertainties could be associated with:

- transient heat transfer coefficients
- impacts of frequency, pressure levels, and working gas
- impacts of seal design on leakage losses.

4.1.2 Gas Spring Testing

The information of Task 4.1.1 will be used to design a testing program for gas springs which will allow for measuring both thermal hysteresis and leakage losses. This program will address gas springs using both clearance and contact seals.

The testing program will consist of:

- Performance testing of gas springs over a range of operating parameters of practical interest in FPSE.
- Comparison of measured losses with those predicted analytically.
- Endurance testing of gas springs to determine the effect of operating time on losses and engine efficiency (possibly due to increased leakage losses - particularly if contact seals are used).

The information from this task will be used to quantify gas spring losses, identify favorable design options, and to verify analytical models.

4.2 Alternative Spring Designs

4.2.1 Review of Alternative Spring Designs.

Alternatives to the use of gas springs will be reviewed and assessed. The prior experience includes that of the Harwell group in England uses mechanical springs to establish the natural frequency of the power piston. These systems are quite small (smaller than 250 watts) and, consequently, this review would consider the range of mechanical spring designs which might be considered for use in FPSE of the capacity required by heat pump applications. These spring options include:

- Helical Springs
- Flat Springs (cantilever, flat coil, etc.)
- Leaf Springs

These options will be examined relative to their fatigue life, loss mechanisms, frequency capability, size, and impact on design options.

4.2.2 Mechanical Spring Evaluation and Testing

Based on the review of task 4.2.1, two of the favorable spring design options will be assessed in more detail. This will include:

- determining how they would be designed into system arrangements of practical interest
- quantifying their impact on FPSE performance (as compared to using gas springs)

If the above indicates that significant performance advantages could accrue from the use of mechanical springs, then this information will be used to design a testing program allowing for verification of mechanical spring/FPSE coupling arrangements. Up to two such configurations will be fabricated and subjected to both performance and endurance testing.

Task 5.0 Regenerator Development/Oscillatory Flow

One of the unique features of a Stirling engine is a regenerator which must periodically transfer heat to and from the working gas stream being shuttled between the engine hot and cold volumes. The performance of the regenerator as measured by pressure drops and heat transfer effectiveness is absolutely critical to overall engine performance. For example, more heat is transferred in the regenerator than in the heater head. As such, even relatively modest deviation from projected performance can significantly impact on overall engine operating characteristics. In this task, therefore, sample regenerator matrices will be tested under periodic flow conditions similar to those found in Stirling engines. The variable will include:

- working gas pressures
- temperature levels
- flow rates
- frequency
- regenerator matrix size and construction

The output of the task will be on quantitative information which will allow more accurate analytical determination of regenerator performance characteristics and the extent to which previous models have (or have not) been accurate. This output will, therefore, be a critical input into improving the overall accuracy of analytical tools used to model engine performance. It is assumed that most, if not all, this work will be done at the ANL oscillating flow test facilities established specifically to undertake this task.

Task 6.0: Component Development - Continuing Support

Tasks 1 through 5 will address early in the program the performance and design options of major components comprising the FPSE/HP. Most of the work described will be completed during the first 3 years of the program. There will, however, be a continuing need for component development and testing within the program - possibly addressing subassemblies which are not included in the previous tasks. There may also be a need to extend testing and redesign of components already identified by tasks 1 through 5 for more extended periods. The details of which components and the associated program requirements cannot be assessed at this time. However, this continuing support activity is an important one and is accounted for in the schedule and budget estimates of the baseline plan.

B SCHEDULE

In the baseline program, it is assumed the work starts on all the component development efforts during the first year of the program. Emphasis is given to generating information required for analytical model development and system configuration decisions which occur over the first 18 months of the program. As a result most of the initial analytical and performance testing efforts are done within the first two years. However, as indicated previously, component development and testing is an ongoing activity which proceeds throughout the program. (Figure 3.2.1)

C. BUDGET ESTIMATES

The budget for component estimates development and testing (Figure 3.2.2) indicate that between $1 and $1.6 million will be required annually over the first three years when critical information is being developed for system level designs, and analytical models. The level of effort is reduced in the latter years to about 500 K annually by which time the system level hardware will already be assembled and undergoing testing.

3.3 COMPONENT 3: SYSTEM TESTING AND DEVELOPMENT

The System Testing and Development Component of the program will include both testing of currently available systems and, after the configuration reviews of program Component 1, the development of those FPSE/HP configurations showing the most promise of meeting program goals. This approach allows for undertaking valuable endurance and performance testing using equipment developed via the programs of the past six years while still maintaining the flexibility to strike out in new directions if so indicated by an objective review process. Little, if any, time is lost in this process since the program now requires extensive testing of the systems developed to date (as compared to proceeding with the development of current technology) to identify performance and endurance issues requiring resolution.

This component consists of the following tasks:

Task 1: System Endurance/Performance Testing (using available systems).

1.1 Test System Selection (Test Plan formulation
1.2 Performance Testing

Task 2: System Design and Development

2.1 System Selections
2.2 System Design
2.3 Fabrication and Assembly
2.4 Checkout Testing and Modification
2.5 Performance and Endurance Testing

Each of the program tasks and associated resource requirements is described below:

FIGURE 3.2.1
SCHEDULE: COMPONENT DEVELOPMENT

YEAR 1 YEAR 2 YEAR 3 YEAR 4 YEAR 5

1.0 BEARINGS & SEALS (Contact)

1.1 Review
1.2 Test & Evaluation
- o Design
- o Screening Test
- o Evaluation Tests

1.3 System Impacts

2.0 POWER EXTRACTION

2.1 Hydraulic Systems

2.1.1 Performance

2.1.2 Cost Analyses
2.1.3 Testing

2.2 Alternative Designs
2.2.1 Review
2.2.2 Analyses
2.2.3 Testing

3.0 HOT END COMPONENTS

3.1 Seasonal Impacts
3.2 Emissions
3.3 Testing
3.4 Cost Analyses

4.0 GAS SPRINGS & ALTERNATIVES

4.1 Gas Springs
4.1.1 Review
4.1.2 Testing

FIGURE 3.2.1
SCHEDULE: COMPONENT DEVELOPMENT

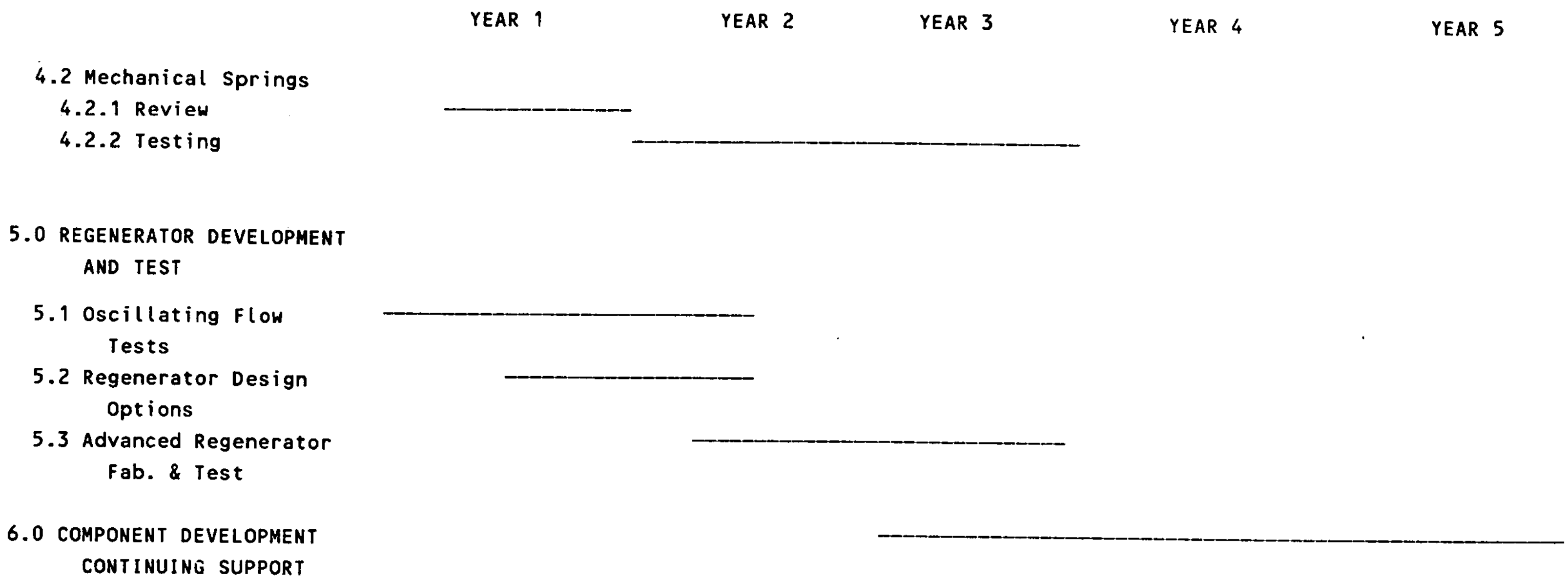


FIGURE 3.2.2
BUDGET ESTIMATES: COMPONENT DEVELOPMENT

	YEAR 1	YEAR 2	YEAR 3	YEAR 4	YEAR 5
1.0 BEARINGS & SEALS	250	300			
2.0 POWER EXTRACTION	400	400			
3.0 HOT END COMPONENTS	200	400	300		
4.0 GAS SPRINGS & ALTERNATIVES	200	300	150		
5.0 REGENERATOR DEVELOPMENT	250	200	100		
6.0 CONTINUING SUPPORT			500	600	400
TOTAL	$1,300 K	$1,600 K	$1,050 K	600	400

A. WORK STATEMENT

Task 1: System Endurance/Performance Testing

1.1 Test System Selection/Test Plan Formulation.

The status of currently available equipment will be reviewed as to their reliability, means of power extraction, and performance potential. Those 1 or 2 systems which would provide the most vital information for the FPSE/HP program will be selected for both performance and endurance testing.

For these systems, detailed test plans will be prepared which indicate:

- o What measurements will be made
- o The duty cycles which will be used
- o Operating parameters
- o The location and type of testing apparatus
- o The format for reporting
- o Resources requirements for testing, analyses, and repair.

1.2 Performance Testing

The test systems will undergo performance testing per the plan developed in Task 1.1. Issues which will be addressed in the performance testing program could include:

- o Engine efficiency
- o Coupling efficiency
- o Combustion System Efficiency
- o Variation of the above efficiencies with:
  - Load
  - Operating Parameters
- o Emissions
- o Effect of cycling
- o Start up time

Performance testing may (probably will) be performed on the same engines used for endurance testing. In such cases, limited performance measurements will be made periodically during the endurance program to determine if wear or other factors are affecting output characteristics.

1.3 Endurance Testing

The test systems will undergo endurance testing in order to establish the credibility of the FPSE in heat pump applications and to identify factors in current designs which limit life and reliability. Sufficient resources will be provided to allow for modifications and repair of key subsystems to allow endurance testing to continue so that a wide range of life limiting factors can be identified and factored into other program activities for resolution.

TASK 2: SYSTEM DEVELOPMENT

The development of a complete system will build on the knowledge gained from the previously described system analysis, model development, and component development tasks.

2.1 System Selection

The information gained from previous program tasks will be used to select one or two engine - heat pump coupling configurations for system level development. This task will include selection of approaches for:

- Engine system control.

- Coupling of the FPSE to heat pump cycles.

- Design of bearings and seals.

The criteria for selection of the engine/heat pump configurations will include:

- Potential for meeting performance and cost goals.

- Level of development risk.

- Degree to which experience in other (automotive, space, etc.) programs is applicable.

- Level of industry commitment.

2.2 System Design and Reevaluation

Detailed designs will be prepared for the systems selected in Task 2.1 for development. These designs will be at a level of detail to show all critical dimensions, materials of construction, tolerance levels, and methods of fabrication and assembly.

Based on these designs the ORNL program manager will reevaluate the system approach based on updated estimates of performance and cost, and on any reassessments of the development risks involved. In some cases this design review may make use of updated information resulting from the ongoing (see schedule) model development and component development tasks.

2.3 System Fabrication and Assembly

Assuming Task 2.2 indicates it is prudent to proceed, detailed subsystem and component designs will be prepared. The system parts will then be fabricated and assembled into a complete system. Where appropriate, subsystems or components will be separately tested to ensure compliance with design requirements. It is assumed, however, that most major subsystems are similar to those being developed and tested as part of the component development tasks.

2.4 Checkout Testing & Modification

Once assembled, the system will undergo performance testing under the full range of conditions required by heat pumps in regions of practical interest.

The system will be fully instrumented so that full performance mph at the subsystem level can be measured for use in system diagnostics and for purposes of analytical model verification.

During this testing phase, modest modifications in subsystem hardware will be made as deemed appropriate by the testing program and analysis results, to seek a performance improvement.

2.5 Performance and Endurance Testing

Once the checkout testing and modification tasks are completed, the system will undergo long term performance and endurance testing. This testing will be done using cycling schedules and capacity mixes consistent with that which might be expected in actual service.

Performance maps of the subsystem and system levels will be measured throughout the duration of the long term testing program to determine the extent of performance changes as the result of such factors as bearing/clearance wear, regenerator degradation, and combustor system design degradation.

The testing program will also include measurements of the effect of part load and cyclic operation on efficiency to allow for estimating annual performance levels.

B. SCHEDULE

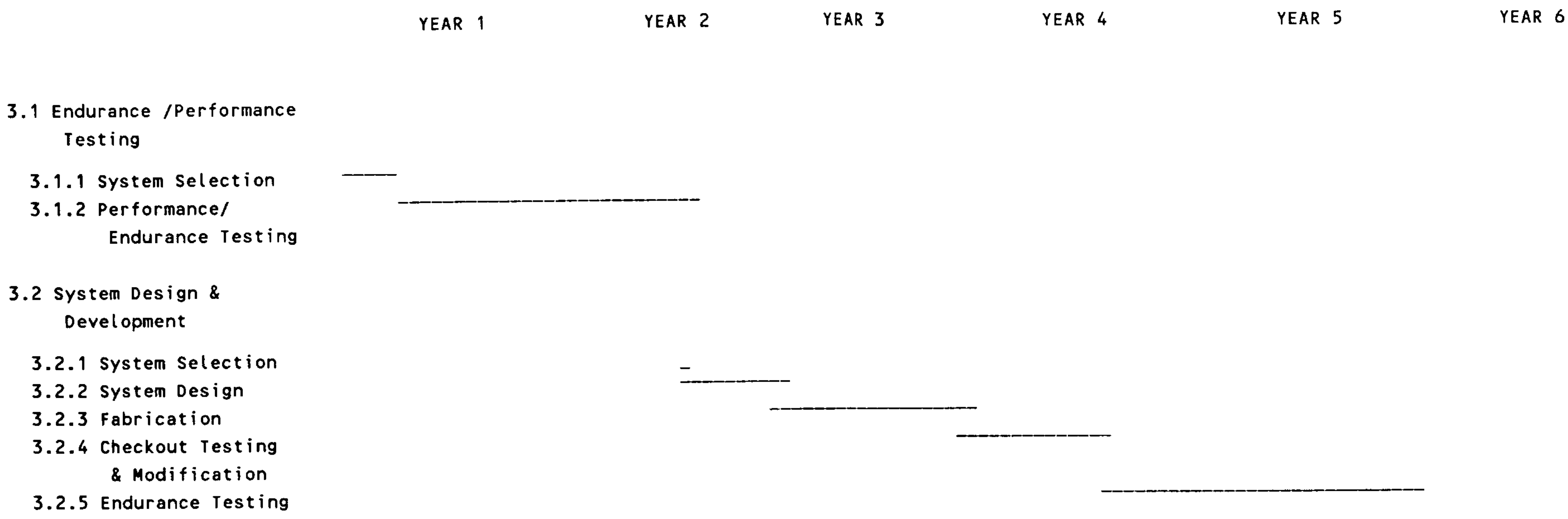
The schedule for this program component is shown on figure 3.3.1. As indicated, the testing of available equipment is initiated early in the program so as to provide early inputs in the planning and evaluation efforts. This testing program is shown as having a duration of 2 years which is sufficiently long to identify most of the life limiting factors associated with current designs. The duration of the testing program will, however, be periodically reviewed to determine if further testing is useful and can be done within reasonable cost constraints. This latter issue clearly depends on the reliability of the equipment under test.

The selection of configurations for further development is made 18 months into the program based on information generated by the configuration evaluations.

Because major decisions affecting configuration, means of coupling to the heat pump cycle, and component design will have been made, the system design and fabrication effort can be done in 18 months. An additional two

FIGURE 3.3.1

SCHEDULE: COMPONENT DEVELOPMENT



years are indicated for performance testing, design modifications, and endurance testing. By the end of this period, sufficient information will be available to determine if commercial development is warranted.

C. LEVEL OF EFFORT/BUDGETING ESTIMATES

Figure 3.3.2 provides budgeting estimates for this program component on an annual basis. Most of this work is expected to be done by contractors. Only dollar value estimates are given and these apply only to the estimated contract values to undertake these efforts. The associated DOE/ORNL management and review functions are accounted for in Task 1 of the System Analyses and Management Component.

As indicated the testing program of Task 1 has a cost of about $400 K per year per system for a total of $0.8 million per year for the baseline program. This reflects the expense in technician and engineering analyses manpower to undertake system level endurance testing.

The system development task budget estimates indicate that about $1 million is expended for the detailed designs for the two systems. As indicated earlier, at the end of this expenditure the system development programs will undergo thorough review before proceeding to the more costly hardware development tasks, which will require about $6 million, over a three year period.

4.0 ALTERNATIVE DEVELOPMENT PROGRAMS

The baseline Program represents a conservative approach to the FPSE/HP development effort with funding which allows for parallel system developments, and with sufficient time to undertake make major decisions only after extensive supporting analytical and experimental work has been completed. In this section, the schedule and budgetary impacts of two other program scenarios are considered.

Program A: Reduced Funding Levels
Program B: Accelerated Program

In these alternative programs the tasks to be undertaken are essentially the same as those indicated for the baseline program. However, the schedule for task initiation and completion are modified to reflect the different priorities and resources. These program scenarios are discussed briefly below:

4.1 Reduced Funding Levels

As a practical matter there are many demands on government resources which usually restrict the flexibility to pursue programs according to the most desirable schedule and including all the desired development tasks.

FIGURE 3.3.2
BUDGET ESTIMATE

	YEAR 1	YEAR 2	YEAR 3	YEAR 4	YEAR 5
Endurance/Performance Testing					
System 1	400	300			
System 2	400	300			
System Design & Development*					
Design		1,000	300		
Fabrication			3,000		
Checkout Testing			200	400	
Endurance Testing				1,200	1,200
TOTAL	1,800	1,600	3,500	1,600	1,200

* Two Systems

To account for the above reality; this scenario was developed assuming:

- o All tasks within the program are stretched out on schedule and in some cases started at later dates to reduce early year funding

- o Only one system is pursued in the system testing and development task,

The resultant schedule, Figure 4.1, indicates a 6 year development program with decisions to pursue a renewed system development program deferred until the end of year 2.

The deferement allows more time for system analyses and evaluation as well as for more experimental results from the component and system testing efforts. Nevertheless, the potential for pursuing two divergent design philosophies into the system development stage is lost which places more pressure on the judgements involved in the system selection process.

Budget estimates for this program option are indicated on figure 4.2. On average the annual expenditure rate is reduced by over 50% from that of the baseline program. Primary contributors to the large reduction are:

- o The extended schedules associated with the supporting analyses and component development tasks

and o the pursuit of only one system into the hardware design and development phase. This alone saves over $4 million dollars (over 20%) of program costs.

The resultant total program cost is $12 million as compared to $18 million for the baseline program.

4.2 Accelerated Program

Depending on national priorities there can be an incentive to accelerate the development of systems which show promise of significantly impacting on energy use and providing a competitive edge to U.S. based technology. FPSE technology could well fall into this category given its widespread potential for use in such divergent applications as heat pump, cogeneration, and space power. For this reason the schedule and budgeting impacts of an accelerated program were considered. Major assumptions associated with this scenario are:

- o The system selection process is accelerated so that it takes place at the end of 12 months. This, in turn, requires accelerating the schedules for key Supporting Analyses and Component Development tasks.

- o Four system options are initially pursued in parallel through the design phase. Only two systems are pursued into the fabrication and test phase. This allows for additional analyses and

FIGURE 4.1

FPSE/HP DEVELOPMENT PROGRAM SCHEDULE (REDUCED FUNDING CASE)

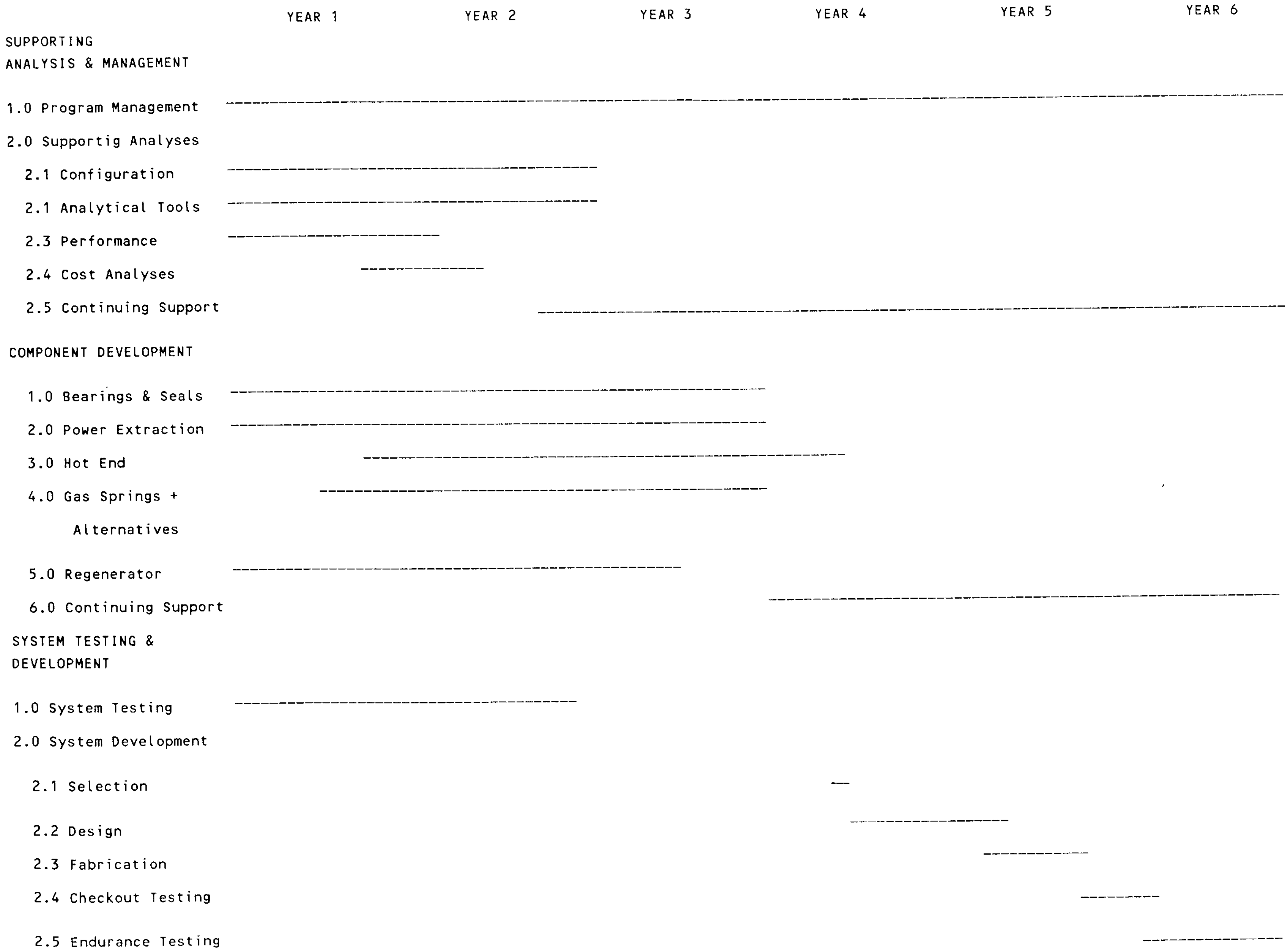


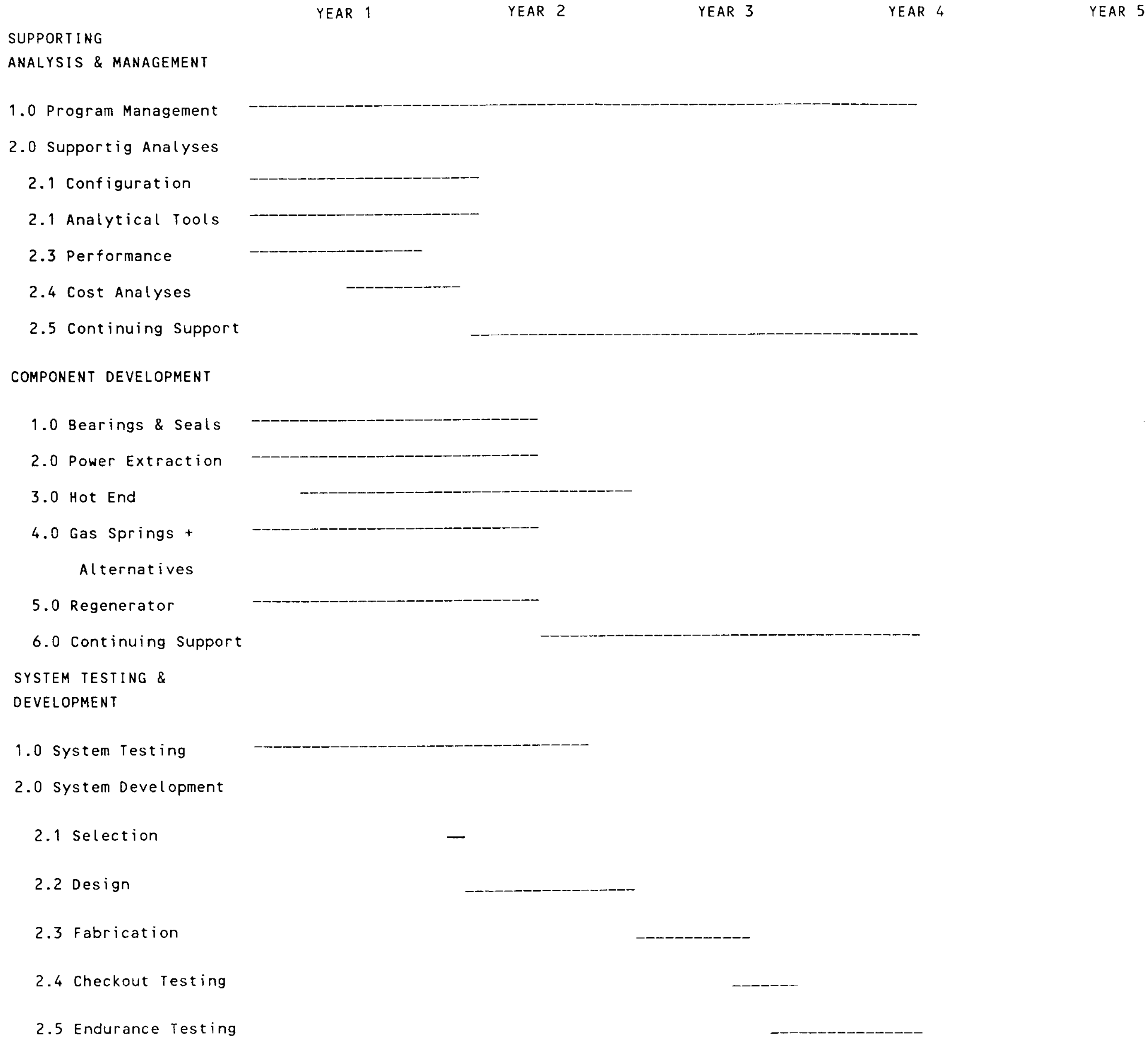
FIGURE 4.2
REDUCED FUNDING PROGRAM - BUDGET SUMMARY

	YEAR 1	YEAR 2	YEAR 3	YEAR 4	YEAR 5	YEAR 6	SUMMARY
SUPPORTING ANALYSIS & MANAGEMENT							
Program Management	250	250	250	250	250	250	
Supportig Analyses	1,300 (1)	500	300	150	150	150	
COMPONENT DEVELOPMENT	1,000	1,300	800	500	300	200	
SYSTEM DEVELOPMENT & TEST							
Endurance Testing (2)	400	300	-	-	-	-	
System Design * (3)	-	-	500	1,700	800	600	
TOTAL	$2,950 K	$2,350 k	$1,850 K	$2,650 K	$2,050 K	$1,200 K	#12,450 K

1. Includes 500 K in test equipment for model verification
2. Assume utilization of currently available equipment, one system only
3. 1 System.

FIGURE 4.3

FPSE/HP DEVELOPMENT PROGRAM SCHEDULE (ACCELERATED PROGRAM)



experimental data to be brought to bear before making decisions on the more costly system development activities.

The resultant schedule, Figure 4.4, indicates a 3.5 year development program. As indicated above, one way this shortened program is achieved is by "hedging bets" early on in the program by allowing for initial design of four systems in parallel. As with the baseline program, however, only the two which show the most promise are continued into the high cost fabrication and testing phases. The annual expenditure rate is increased by over 50% from the baseline program. Primary reasons for this are:

- More concentrated efforts associated with all program functions requiring a faster expenditure rate.

- Additional costs (about $1 million) to allow for pursuing multiple system concepts through the design phase.

However, total program cost is increased by only 25% to $21 million due to its shorter duration.

FIGURE 4.4

ACCELERATED PROGRAM - BUDGET SUMMARY

	YEAR 1	YEAR 2	YEAR 3	YEAR 4	YEAR 5	SUMMARY
SUPPORTING ANALYSIS & MANAGEMENT						
Program Management	400	400	400	300		
Supportig Analyses	2,000[1]	800	600	300		
COMPONENT DEVELOPMENT	1,600	2,000	1,300	1,000		
SYSTEM DEVELOPMENT & TEST						
Endurance Testing[2]	1,000	800	-	-		
System Design * [3]	-	2,000	4,300	2,000		
TOTAL	$5,000 K	$6,000 K	$6,600 K	$3,600 K		$21,200 k

1. Includes 500 K in test equipment for model verification
2. Assume utilization of currently available equipment
3. 2 Systems in parallel

Arthur D. Little, Inc.

APPENDIX I

FREE PISTON STIRLING ENGINE INFORMAL SESSION AT THE IECEC

BACKGROUND

As part of the IECEC meeting a session dealing with Free Piston Stirling Engines was held on Thursday evening, August 24th. The meeting was well attended with approximately 25 participants.

The meeting lasted approximately 2_ hours and was very useful in providing a broad spectrum of views on key issues affecting FPSE development.

In order to provide some direction to the meeting, I presented a brief talk at the beginning consisting of:

- o Technical/Cost Requirements for a FPSE/Heat Pump System
- o Observations on Technology Status
- o Technology Development "Questions"

A copy of the viewgraphs is provided in Attachment I.

The format of the meeting was intended to encourage disagreement with the "observations" and to solicit answers to the "questions" stated in the presentation.

Interpretations of the views stated at the meeting on major questions are summarized below.

Modeling Capability

These was something less than unanimity on the capability of the FPSE community to accurately model and predict engine behavior. Diverse comments included:

- o The major uncertainty is in modeling the dynamic behavior of unconstrained FPSE - particularly under changing load conditions.
- o Some felt that loss mechanisms were adequately modeled. Others, however, firmly disagreed and indicated, for example, up to 200 percent disagreement on losses associated with gas springs.
- o Somewhat contradictory to the first observation; one participant indicated that differing models often agreed on the gross behavior but would have significantly diverse predictions on individual entries (losses, etc.) leading to the overall predictions. This obviously signifies a significant lack of understanding which is obscured by "calibrating" models to agree with specific results.

Efficiency Potential of FPSE

There appears to be "weak" agreement on the notion that at design conditions a FPSE was about as efficient as a kinematic engine, i.e., losses introduced by gas springs, etc., roughly counteracted decreased friction losses resulting from eliminating the shaft seal and mechanical drive train.

However, several participants indicated strongly that off-design performance of the FPSE when driving a heat pump could be quite poor and difficult to predict. This is due to the direct dynamic coupling of the HP compressor to the Stirling engine dynamics. This, in turn, will change phase angle relationships in the engine with impacts on efficiency which are still not well understood. The adequacy of any control system to completely circumvent this problem was called in question.

Some felt this represented one of the major problems facing FPSE/Heat Pump where the engine is directly driving a vapor compression cycle with widely varying inlet suction pressures.

Rubbing Bearings and Seals

Many of the participants did not have an awareness of the life implications of rubbing bearing and seals (reflecting, in part, an analytical bent on part of many of the participants).

Two participants indicated that the marginal experience of G.E. and others with rubbing piston seals may have reflected lack of experience with such seals. They supported this contention with the claim that Philips is operating rubbing seals very successfully in cryogenic coolers for over 10,000 hours. Philip's approach to doing this in their equipment appears to be proprietary. It was pointed out, however, that part of the "art" includes using an initially "rough" cylinder surface which becomes coated with seal material after only a few hours of running time. After this running in period it is claimed there is no longer any net transfer of material from the seal to the cylinder wall.

Another participant indicated that IBM had, several years ago, published guidelines on rubbing seals as a function of temperature, velocities, pressure differentials, and other environmental conditions. He suspected that those guidelines would indicate that the conditions in a high power density Stirling engine were very severe for rubbing seals.

Gas Bearing and Clearance Seals

Those few participants with a knowledge of this subject felt that properly designed gas bearings and clearance seals should not represent a serious manufacturing cost problem despite the very close tolerances involved. Examples of similar close tolerances on parts used in hydraulic valves and I.C. engines were cited to support this view.

Hot End Subassembly Life

No life testing experience with hot end subassemblies was known to any of the participants.

Several were of the opinion that with proper choice of materials and design there should be little problem achieving the necessary life. However, none of the participants could cite experience life testing of hot end assemblies or even how extended life capabilities are factored into the design process.

FPSE/Heat Pump Coupling

The point was raised that the direct coupling of a FPSE to the heat pump cycle in itself raised serious control, efficiency, and stability issues. This is due to the fact that heat pump dynamics will be a function of suction pressure which is, in turn, a function of ambient air temperature. One participant stated, for example, that stable operation over a wide range of ambient air temperatures would be nearly impossible in an inertial compressor system. The participants were not quite so sure of the seriousness of this issue relative to the hydraulically coupled systems as exemplified by the MTI system. It was observed, however, that the dynamics of these systems were also influenced by freon pressure levels in the compressor.

Two approaches were discussed which would (maybe) simplify designs and decouple engine dynamics from the effects of wide variations in ambient air temperatures.

(a) Duplex Stirling

Directly coupling a FPSE to a Stirling heat pump cycle include:
- o Simplification of the coupling system so that the whole engine/heat pump system uses only 3 moving parts.
- o Relative insensitivity of capacity and dynamics of the heat pump cycle to ambient air temperature (which has negligible effect on pressure levels).

Concern was raised, however, about how to configure the Stirling heat pump cycle so it achieves acceptable efficiencies operating in the narrow temperature range of interest. For example, the possible role of "iso-thermalization" in improving the performance of the cooling cycle was discussed (inconclusively).

(b) FPSE Alternator ! Electric Vapor Compression

The performance of linear alternators has been improving which "re-opens" the possibility of a FPSE alternator driving a conventional electric heat pump. With a 90 percent alternator and 85 percent motor this option could have a transmission efficiency of about 75 percent which compares favorably with the MTI hydraulic option. Some participants, therefore, felt this option might be worth considering.

FPSE vs Kinematic Drive

The final issue discussed was the degree to which FPSE offered a more attractive option as a heat pump driven than a KSE.

One participant with KSE experience observed that the shaft seal problem of KSE was getting under better control and probably represented a lower risk development than the complex dynamics of FPSE. Resolution of the shaft seal problem would substantially reduce the incentive to pursue the FPSE option.

It was also pointed out by several participants that the performance of the KSE would probably be superior to the FPSE over the wide range of operating conditions. This is due to the decoupling of KSE dynamics from load conditions. These participants felt, in general, that the KSE provided a much better potential as a heat pump drive than the FPSE.

ATTACHMENT I

INTRODUCTORY VIEWGRAPHS

INFORMAL SESSION ON FREE PISTON STIRLING ENGINE DRIVEN HEAT PUMPS

IECEC MEETING

August 23, 1984

REQUIREMENTS (FOR HP)

LIFE:	25,000 - 50,000 HOURS
MAINTENANCE:	1 PER YEAR (2,500 - 4,000 HOURS)
EFFICIENCY:	_30% (BRAKE)
COST:	$200 - $400/HP

"CLAIMED" ADVANTAGES

- HIGH RELIABILITY/LONG LIFE
  - Few Moving Parts
  - Direct Coupling to HP CYCLE
  - No Shaft Seal
  - No Highly Loaded Bearings
- LOW COST
  - Simple Designs with Few Moving Parts
  - Direct Coupling to HP
- HIGH EFFICIENCY
  - Reduced Friction Losses
  - Reduced Engine ! HP Transmission Losses

OBSERVATIONS - LONG LIFE

- NOT DEMONSTRATED TO DATE AT REQUIRED POWER LEVELS
- STILL HAVE BEARINGS AND SEALS WHICH OPERATE IN DRY HELIUM
- GAS BEARINGS AND CLEARANCE SEALS INTRODUCE NEW SET OF RELIABILITY ISSUES
- ENGINE ! HP CYCLE COUPLING MECHANISMS OFTEN COMPLEX
- LIFE OF HOT END COMPONENTS OPERATING IN CYCLIC MODE IS VERY UNCERTAIN

OBSERVATIONS - HIGH EFFICIENCY

- NET EFFICIENCY OF MOST FPSE SYSTEMS HAS BEEN DISAPPOINTING (COMPARED TO INITIAL PROJECTIONS) - APPARENTLY DUE TO HIGHER THAN EXPECTED BEARING, GAS SPRING, AND SEAL LOSSES

- THE LOSS OF PHASE ANGLE CONTROL MAY MAKE ANY "UNCONSTRAINED" FPSE FOR A HEAT PUMP DRIVE INHERENTLY LOW IN EFFICIENCY AVERAGED OVER A WHOLE RANGE OF OPERATING CONDITIONS

- THE ANALYTICAL MODELS DO NOT APPEAR TO BE SUFFICIENTLY WELL DEVELOPED TO ESTIMATE THE EFFICIENCY POTENTIAL OF FPSE/HEAT PUMPS SUFFICIENTLY ACCURATELY FOR EVALUATION AND DESIGN PURPOSES

OBSERVATIONS - LOW COST

- MANY OF MOST COSTLY STIRLING ENGINE SUBASSEMBLIES RETAINED IN FP CONFIGURATION (HEATER HEADS, AIR PREHEATERS, ETC.)

- SOME SYSTEMS EMPLOY COMPLEX TRANSMISSION SYSTEMS

- HIGH PRECISION REQUIRED FOR GAS BEARINGS AND CLEARANCE SEALS

QUESTIONS

- DO MODELS ADEQUATELY ADDRESS LOSS MECHANISMS OF FPSE?
- WHAT ARE EFFICIENCY EXPECTATIONS FROM A FPSE?
- CAN RUBBING BEARINGS AND SEALS ACHIEVE LIFE REQUIREMENTS?
- CAN GAS BEARINGS AND CLEARANCE SEALS BE SUFFICIENTLY LOW COST?
- HOW LONG CAN HIGH TEMPERATURE COMPONENTS LAST DURING CYCLIC OPERATION?
- WHAT ARE THE BEST OPTIONS FOR COUPLING A FPSE TO A HEAT PUMP CYCLE?
- IS THE FPSE REALLY A BETTER BET THAN A KSE FOR HEAT PUMP APPLICATIONS?

APPENDIX II

SEALS AND BEARINGS

One of the key issues regarding the design life and performance of FPSE are the bearings and seals. Most of the literature on FPSE developments devotes relatively little attention to this technology area. One reason for this is that efforts to date have tended to focus on system thermodynamics and dynamics. However, experience in other programs (primarily for space cryocooler applications) which require the operation of moving mechanical elements in a dry helium environment indicates that ultimately life and performance of such equipment is determined, in large part, by the design of bearing and seals.

The following are charts used in a presentation at ORNL on September 19, 1984 which outline some of the relevant issues. Observations on important points made by these charts include:

- Exhibit A.2.1 indicates the types of bearings which could be used in an FPSE. Virtually all of these seal types have been used in Stirling engines.

- As indicated by Exhibit A.2.2, the leakage by a clearance seal is proportional to the <u>cube</u> of the gap thickness. To keep leakage losses to acceptable limits, gap thicknesses less than 1 mil and often less than 0.5 mils are required. This requires close manufacturing tolerances on the individual parts which comprise the piston/seal/cylinder assembly.

  Also note that the leakage losses increase significantly if the seal rod is not centered (as will always be the case in practice). In fact, leakage can increase by 250 percent (compared to the perfectly centered case) when the rod is positioned such as to rub the cylinder lining.

  Analytical models for estimating clearance seal losses generally do not account for off-center operation and, therefore, tend to underestimate this important loss mechanism.

- Exhibit A.2.3 indicates the effect that seal leakage has on system performance. All effects of seal leakage are deleterious, i.e., reduce performance, and the reduction in performance increases as the amount of leakage increases.

- Exhibit A.2.4 indicates several of the DOD programs to develop clearance and sliding seals for use in dry helium environments. Major programs are underway at Hughes Aircraft, Cryogenic Technology Inc., and North American Phillips. Many of the associated equipment development projects have been active for over 10 years which suggests the difficulty in achieving reliable unlubricated bearings and seals. Also, the experience gained from these programs may be directly relevant to FPSE developments.

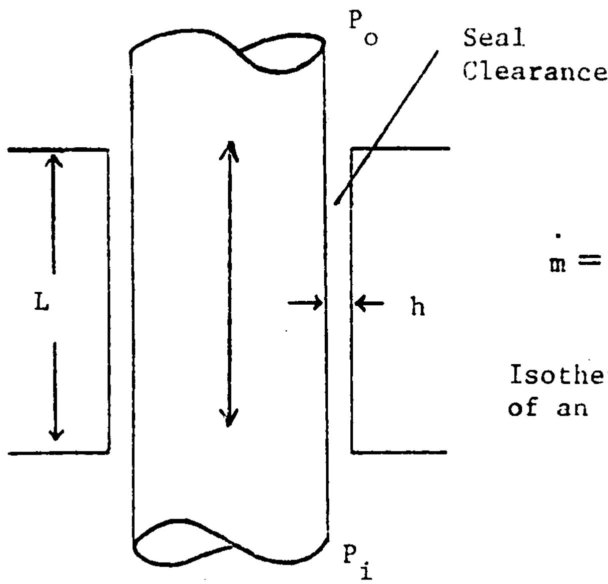
- As indicated by Exhibit A.2.5, wear rates for candidate mating surfaces can be estimates by Archard's equation. These wear rates are

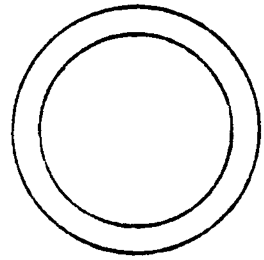
EXHIBIT A.2.1

TYPES OF DYNAMIC SEALS

- o CONTACTING
  - Lip Seals
  - Piston Rings
    - + Pressurized
    - + Balanced
- o NON CONTACTING
  - Clearance Seals
  - Labyrinth Seals
- o COMPLIANT BARRIER
  - Diaphragms
  - Bellows
  - Roll Socks

a) Concentric Seal

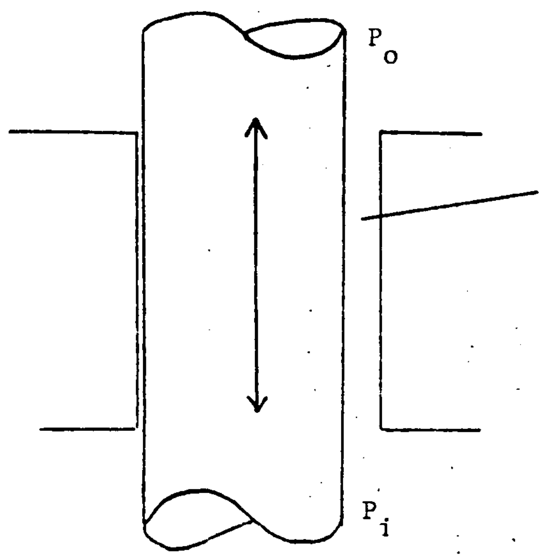


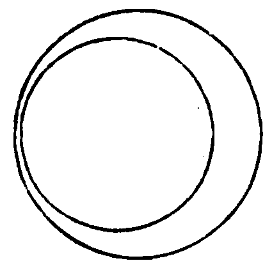


$$\dot{m} = \frac{12.5bh^3(P_o^{\,2} - P_i^{\,2})}{\mu LZRT}$$

Isothermal Compressible Flow of an ideal gas

b) Eccentric Seal





Seal Clearance

$$\dot{m} = \alpha \left[\frac{12.5bh^3(P_o^{\,2} - P_i^{\,2})}{\mu LZRT}\right]$$

$1 < \alpha < 2.5$ (for maximum clearance)

$\dot{m}$ = mass flow rate through seal
μ = fluid viscosity
Z = compressibility of gas
R = gas constant
T = temperature

EXHIBIT A.2.2

LEAKAGE THROUGH CLEARANCE SEAL

EXHIBIT A.2.3

EFFECT OF SEAL LEAKAGE

- SYSTEM DYNAMICS
  - LEAKAGE PAST GAS SPRING SEALS INCREASES "HYSTERISIS" LOSSES AND REDUCES STIFFNESS, AND THUS EFFECTS RESONANT FREQUENCY
  - LEAKAGE INCREASES DAMPING LOSSES, AND THUS EFFECTS AMPLITUDES OF MOTION AND PHASE RELATIONSHIPS BETWEEN MOTIONS

- THERMAL PERFORMANCE
  - LEAKAGE PAST DISPLACER CONSTITUTES A HEAT LOSS, SINCE LEAKAGE FLOW BYPASSES HEAT EXCHANGERS AND REGENERATOR
  - LEAKAGE INCREASES FLOW LOSSES AND THUS REDUCES EFFICIENCY

proportional to the normal force pressing the surfaces together, the distance over which the surfaces slide, and a "wear constant" which is empirically defined. In an engine, the sliding distance is, in turn, proportional to the operating speed (for a given displacement).

The problem with using this equation for accurate quantitative evaluation of wear is that, for a given material pair, the "wear constant" depends on many application specific factors such as temperature, load reversals, and gas pressure levels. This equation can be suggestive of wear rates but is generally regarded as being only accurate within a factor of 4-10 until application specific data is generated (Exhibit A.2.5). However, as shown on Exhibit A.2.6, the "wear constant" for unlubricated surfaces are about a factor of 50 higher than reasonably well lubricated surfaces and a factor of 500 higher than for excellently lubricated surfaces. This indicates the difficulty in developing long lived mechanical equipment requiring unlubricated bearings and seals.

- Exhibit A.2.7 summarizes the issues regarding seals. Suffice it to say that seals are among the most critical systems elements, since they are important determinants of both life and performance.

- As indicated by Exhibits A.2.8 and A.2.9 contact bearings have the same wear problems and uncertainties as do contact seals. Non-contact bearings are of two types, hydrostatic and hydrodynamic. These bearing configurations have the advantage of eliminating mechanical wear. However, both have associated losses which will detract from engine efficiency. Also, both require small gap thickness (_ 0.5 mils) to result in acceptable performance, which implies very high precision machining of piece parts and meticulous cleaning and assembly techniques.

EXHIBIT A.2.4

CURRENT DOD DEVELOPMENT PROGRAMS ON SEALS FOR REGENERATIVE TYPE CRYOCOOLERS

- HUGHES AIRCRAFT (KINEMATIC VM)
  - PRIMARILY EXPERIMENTAL PROGRAM TO DEVELOP SLIDING SEALS WHICH WILL LAST 50,000 HOURS IN A VM (DUPLEX STIRLING) CRYOCOOLER
  - SEAL DEVELOPMENT HAS RECEIVED EMPHASIS SINCE PROGRAM INCEPTION IN LATE 1960'S
  - CONDUCTING WEAR AND LIFE TESTS IN SIMULATORS AND IN 6 REFRIGERATORS
- CRYOGENIC TECHNOLOGY INCORPORATED (SPLIT STIRLING WITH FREE-PISTON DISPLACER AND KINEMATIC COMPRESSOR
  - ANALYTICAL AND EXPERIMENTAL PROGRAM TO INVESTIGATE SEAL LEAKAGE AND WEAR IN SPLIT STIRLING CYCLE CRYOCOOLERS
  - GOAL: TO OBTAIN 2,000 HOURS OPERATION LIFE
  - CONSIDER CLEARANCE SEALS AND RUBBING SEALS
- NORTH AMERICAN PHILLIPS (FREE-PISTON INTEGRAL STIRLING)
  - DEVELOPING CLEARANCE SEALS FOR MAGNETICALLY LEVITATED PISTON AND DISPLACER
  - EMPHASIS ON DETERMINING EFFECT OF LEAKAGE ON PERFORMANCE AND ON MANUFACTURING TECHNOLOGY

ADHESIVE WEAR OF SLIDING SURFACES

- ARCHARD'S EQUATION IS THE SIMPLEST AND MOST POPULAR FORMULATION FOR ESTIMATING WEAR RATES

$$V = \frac{KFX}{3H}$$

V = VOLUME OF WEAR

K = WEAR CONSTANT

F = NORMAL FORCE PRESSING THE SURFACES TOGETHER

X = DISTANCE OF SLIDING

H = PENETRATION HARDNESS OF MATERIAL

- THIS EQUATION IS GENERALLY ACCEPTED TO BE ACCURATE WITHIN THE RANGE $\pm$ A FACTOR OF 4 TO $\pm$ A FACTOR OF 10!!

WEAR CONSTANT IN ARCHARD'S EQUATION

- AN EMPIRICAL COEFFICIENT WHICH IS SPECIFIC TO THE MATERIALS AND THE ENVIRONMENT

- SOURCES OF DATA (IN DECREASING ORDER OF ACCURACY)
  - PRIOR EXPERIENCE IN SIMILAR APPLICTIONS
  - LABORATORY WEAR TESTS IN SIMULATED ENVIRONMENTS
    - STANDARD WEAR TESTERS
    - SPECIAL PURPOSE WEAR TESTERS WITH SIMULATED GEOMETRY AND ENVIRONMENT
  - TABULATED DATA IN THE LITERATURE

- RELATIVE VALUES OF WEAR CONSTANT FOR METALS

	IDENTICAL MATERIALS	INCOMPATIBLE MATERIALS
UNLUBRICATION (IN AIR)	50,000	500
POOR LUBRICATION	10,000	100
GOOD LUBRICATION	1,000	10
EXCELLENT LUBRICATION	30	1

EXHIBIT A.2.7

ISSUES REGARDING SEALS

- SEALS ARE CRITICAL COMPONENTS, EFFECTING BOTH PERFORMANCE AND LIFE
- SEAL DESIGN CANNOT BE SEPARATED FROM DESIGN OR THE REST OF THE SYSTEM
- SLIDING SEALS WITH "ADEQUATE" LIFE AND PERFORMANCE HAVE NOT BEEN DEMONSTRATED IN THE FPSE/HAHP PROGRAM OR IN OTHER PROGRAMS WITH SIMILAR REQUIREMENTS
- IN SPITE OF SEAL DEVELOPMENTS BEING CONDUCTED ELSEWHERE ON SEALS WITH SIMILAR REQUIREMENTS, SUBSTANTIAL DEVELOPMENT EFFORT WILL PROBABLY BE REQUIRED FOR THE SEALS IN EACH NEW DESIGN OR FPSE/HAHP
- CLEARANCE SEALS HAVE THE POTENTIAL TO ACHIEVE LIFE REQUIREMENTS BUT THEY WILL TEND TO REDUCE ENGINE PERFORMANCE AND BE MORE COSTLY TO MANUFACTURE THAN SLIDING SEALS

EXHIBIT A.2.8

TYPES OF BEARINGS

- CONTACTING
  - DIRECT CONTACT BETWEEN PISTON (DISPLACER) AND CYLINDER
  - RIDER RINGS TO SUPPORT PISTON
- NON CONTACTING
  - EXTERNALLY PRESSURIZED BEARINGS (HYDROSTATIC GAS BEARINGS)
  - SELF ACTING JOURNAL BEARING* (HYDRODYNAMIC GAS BEARINGS)
  - MAGNETIC BEARINGS

* ROTATION REQUIRED FOR OPERATION

EXHIBIT A.2.9

BEARING ISSUES

- CONTACTING:
  - SUBJECT TO SOME WEAR PROBLEMS AS SEALS
- NON CONTACTING
  - HYDROSTATIC BEARINGS REQUIRE A SOURCE OF PRESSURIZED HELIUM FOR THEIR OPERATION. ASSOCIATED LEAKAGE LOSSES AND IMPACTS ON PERFORMANCE MAY BE CONSIDERABLE. ALSO, THE PRESSURE EXTRACTION MEANS MAY REQUIRE VALVES AND BUFFER TANKS, ADDING TO SYSTEM COMPLEXITY.
  - HYDRODYNAMIC BEARINGS ARE A FORM OF JOURNAL BEARINGS AND REQUIRE THAT THE PISTONS BE ROTATING. INTERNAL LOSSES ARE NEGLIGIBLE BUT THE MEANS OF IMPARTING ROTATION MAY HAVE HIGH LOSSES.
- BOTH TYPES OF BEARINGS REQUIRE RADIAL CLEARANCE ON THE ORDER OF 0.5 MILS TO RESULT IN ACCEPTABLE PERFORMANCE

APPENDIX III

HOT END COMPONENTS ASSESSMENT OF STIRLING ENGINE TECHNOLOGY FOR RESIDENTIAL GAS-FIRED HEAT PUMP APPLICATION

by

Richard N. Caron

February 1985

Arthur D. Little, Inc.

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1.0 INTRODUCTION

1.1 Background

The hot end components in the HAHP are responsible for converting the fuel energy of natural gas to sensible energy which is transferred to the engine. Hot end components consist of the: 1) combustion chamber and associated controls; 2) heater head assembly; and 3) air preheater. The relative functions of each of these components are illustrated in Figure 1-1. The natural gas is combusted adiabatically in the combustion chamber at flame temperatures in the range of 2900-3600°F. The flue gas from the combustor transfers heat to the engine through the heater head. Efficient Stirling engine operation requires heater head operating temperatures of 1200-1400°F with high heat flux levels, typically 60,000 Btu/hr ft^2. Heat transfer in the heater head is desired to be isothermal so the combustion products leave the heater head at temperatures 100-200°F above the heater head temperature. Exhausting flue gas at these high temperatures requires a recuperator to recover this waste heat and transfer it to the incoming combustion air to achieve efficiency levels competitive with alternative heating/ cooling equipment.

The high operating temperatures and heat flux levels associated with the hot end assembly critically impacts the seasonal HAHP performance, emissions, and manufacturing costs. The magnitude of this impact is demonstrated by the following key observations:

1) The seasonal energy losses from the hot end assembly account for 25-40% of the fuel consumption of the HAHP.

2) The heater head temperature strongly effects stirling engine efficiency. In the vicinity of 1300°F heater head temperature, an increase of 12°F in heater head temperature increases power and efficiency of about 1%.

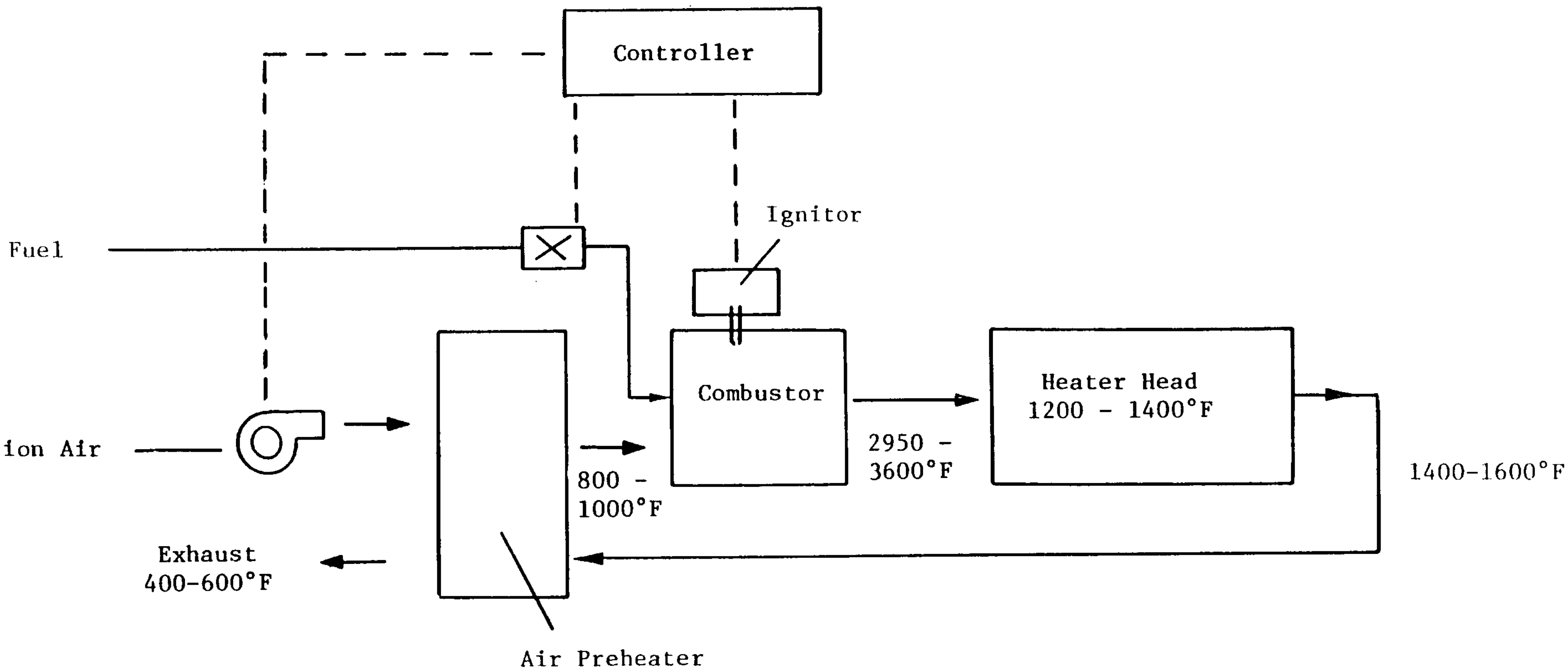

Figure 1-1. ILLUSTRATION OF HOT END COMPONENTS FUNCTION

3) High flame temperatures due to high levels of air preheat exacerbates NO_x emission levels. Measures such as EGR or CGR to reduce flame temperature must be employed. These measures usually have a deleterious impact on combustor efficiency.

4) The manufacturing cost estimates of the hot end components of both the MTI and GE HAHP prototypes indicate that the hot end will account for 20-30% of the total cost. This significant fraction is attributed to the relatively complex component designs (air preheater, heater head) and costly materials of construction (high chromium and cobalt content materials).

1.2 Summary of Target Systems Requirements

The range of target system requirements is presented in Table 1-1. Based on the overall system COP, a firing rate of 38,000-43,000 Btu/hr is required to satisfy the peak residential space heating and cooling loads in the majority of climatic zones of the U.S. The overall pressure drop of the hot end components (4-10" w.c.) must be low to minimize blower first cost and operating cost. The fuel should be combusted with as low as possible excess air levels to minimize stack energy losses without causing excessive CO. This range of excess air which satisfies these requirements is approximately 25-40%. Carbon Monoxide emissions are regulated by the ANSI standard Z21 1978 for gas fired heating systems. The present limit is 400 ppm in the flue gas. However, since CO is extremely sensitive to manufacturing tolerances, values in the range of 100-200 ppm are generally set as targets to ensure that 90% of all units manufactured meet the ANSI limit. NO_x emissions are not presently limited, however. Targets set by the GRI and California Emissions Standards indicate that NO_x should be less than 100 ppm.

Target combustor efficiency is 80%. This efficiency is defined as follows for steady state operation.

$$\eta_{SL} = (1 - \frac{\dot{Q}_{Stack\ Loss}}{\dot{Q}_{Input}})$$

Table 1-1

TARGET SYSTEM REQUIREMENTS FOR HOT END COMPONENTS

Firing Rate	38,000 - 43,000
Excess Air	25 - 40%
Combustion Air Flow Rate	8 - 10 SCFM
Heater Head Temperature	1200 - 1400°F
Heater Head Heat Flux	55,000 - 65,000 $Btu/hrft^2$
Heater Head Heat Volume	< 1 cu. ft
Exhaust Temperature	400 - 600°F
Combustor Efficiency	80%
CO_{AF}	100 - 200 ppm
NO_{xAF}	100 ppm
Pressure Drop	4 - 10" W.C.
Operating Life	25,000 - 50,000 hours
Operating Cycles	20 - 30 million
Manufacturing Cost	$120 - 150

The efficiency is a unique function of the exhaust temperature and excess air level. Note that this "stack loss efficiency" is not equal to the operating efficiency since there are on-cycle and off-cycle energy losses that are not considered. To achieve 80% stack loss efficiency at a 25-40% excess air level, the preheater must extract enough energy from the heater head exhaust so that the flue exhausted to the outdoors is in the range of 400-600°F. A more complete discussion of operating efficiency is presented in Section 3.1.

Heater head temperature must be in the range of 1200-1400°F to ensure that the overall system efficiency is superior to competitive options for residential heating and cooling. Load volume in the heater tubes must also be minimized since every one percent decrease in total volume causes a one percent increase in engine work output. The geometry of the heater head, combustor firing rate, and thermal efficiency must combine to give heat flux levels across the heater head on the order of 55,000-60,000 Btu/hr.

The life of the HAHP should be 15-20 years to be competitive with other heating/air conditioning systems which implies an operating life of 25,000-50,000 hours with 20-30 million cycles of burner activation. Manufacturing cost of the combustor must be in the range of $120-150 to meet overall system cost targets.

2.0 OVERVIEW OF HOT END COMPONENT DEVELOPMENTS

2.1 Free Piston Stirling Engine Heat Pumps

Stirling engine gas-fired heat pumps have been under development by GE since 1975 and MTI since 1976.(2,3,4) Both manufacturers have evolved several generations of combustors before settling on their present designs. Key characteristics of the hot end assembly for GE and MTI are summarized in Table 2-1. Although Sunpower has not developed a Stirling engine/heat pump, they have built many free piston Stirling engines for demonstration. Since Sunpower has fundamental differences in design philosophy from GE and MTI, their design approach will be discussed in this section.

2.1.1 General Electric Co.

GE developed two prototype units. During the course of Proto 1 development, General Electric embarked on a formal evaluation of the various means to burn natural gas. They considered bunsen type burners, diffusion burner, and transportation burners. Based on a methodical review of their operation, development status, and system requirements, GE selected the radiation transpiration burner. The diffusion and bunsen type burner designs did not provide acceptable temperature and flow uniformity. The radiant transportation burner was chosen over a water cooled transpiration because of its higher efficiency (burner is cooled by radiating directly to heater head instead of exhausting hot cooling water).

The main problems GE identified with Proto 1 are summarized in Table 2-2. GE solved these problems and modified the design of Proto 2 accordingly. In addition, Proto 2 incorporated a White Rogers dual firing rate two stage electric gas valve and a damper mechanism orifice hole to control combustion air flow. The gas valve and damper controls the fuel/air flow rate so that the burner fires at 24,000 Btu/hr when the ambient temperature is between 30-85°F, and 41,000 Btu/hr when the ambient temperature is outside this range.(18) GE claims that about 25% excess air level can be maintained for both firing rates.

Table 2-1

KEY CHARACTERISTICS OF STIRLING HOT END
FOR HEAT PUMP APPLICATIONS

	MTI	GE
Burner Type	Turbulent Burner	Radiant Transpiration
Firing Rate (Btu/hr)	41,400	42,300
Excess Air (%)	70	25
Heater Head Exit Temp (°F)	1555	1330
Heater Head Peak Pressure (psig)	1200-1500 psig	1000-1200 psig
Heater Head Efficiency (%)	90-92	92-94
Preheater Exhaust Temp (°F)	480	580
Air Preheat Temp (°F)	1350-1400	820
Air Preheater Temp (°F)	1467	1270
Air Preheater Efficiency (%)	94	63-68
Stack Loss Efficiency (%)	78-79	75-77
Radiation Loss (%)	2.4	2.7
Actual Steady State Efficiency (%)	75-76	72-74
CO (ppm)	80	200-300
NO_x (ppm at 25% EA)	1300	20-30
ΔP in w.c.	10-15	3-5
HC (ppm)	10-20	15-30
Heater Head Temperature (°F)	1290	1250
Heater Head Material	Inconel	Incoloy 800 Tubing Inconel 600 Fins

Table 2-2

PROBLEMS ENCOUNTERED WITH GE PROTO 1

Problem	Remedial Measure For Proto 2
Short life of material (400 hrs) due to excessive oxidation for stainless steel or Inconel	Burners made of Metex pressed-knitted structure of Kanthal Al or Hoskins 875 wires
Failure of stainless steel clamps which support burner element	Use threaded rod
Spark ignitor and flame rod made from Hoskins 667 (96 Ni-4 Mn) developed a layer of oxide which inhibited operation	Kanthal A-1 (80 Ni-15 Cr-5Al) could withstand burner wall temperatures without problems with oxide layer
Combustion/Engine Exhaust Seal Failure (Kawool rope-type seal exhibited excessive wear characteristics)	Reduce interface temperatures by mounting flange ring to combustor and use conventional type graphite impregnated PTFE seal
Accumulation of insulation debris on combustor interior, excessive conduction heat loss from combustor	Blanket instead of loose Kawool insulation on inside and increased insulation in exterior

A schematic of the Proto 2 combustor is shown in Figure 2-1. Combustion air enters the inlet duct and is distributed circumferentially by an inlet vane before it travels in an axial direction along the outer section of the corrugated air preheater to the mixing chamber. After leaving the preheater, the air is mixed with the gas which enters through a gap in the diffusion ring. The fuel/air mixture flows out of the ring and mixes with the preheated air. The air/gas mixture then flows through a restricted cross-sectional area (created by space gaps) and the swirler into a transpiration burner element of cylindrical construction. The element consists of chopped pressed and sintered Hoskins 5.6 mil diameter wire with 80% porosity. A laminar flame is established about 1 mm from the outer surface of the burner element. An electronic spark ignition initiates combustion: a flame detector circuit senses the presence of a flame.

Radiation shields are used to direct heat transfer to the heater head. The heater head consists of 12 1/4" ID tubes 16" long which have 8 fins per inch over a 4" section. Exhaust gases exit the combustion chamber along the inside diameter of the corrugated air preheater in an axial direction opposite to that of the inlet air. Two cylindrical sheet metal housings are closely fitted at the inside and outside of the corrugated sections. The hotter combustion products flow through the inner triangular ducts from one end while the colder fresh air is supplied from another end through the outer triangular ducts. Large circular torus chambers or manifolds are provided to maintain flow uniformity through all preheater flow passages. Radiation shields are also insulated around the inner surface of the preheater to reduce the heat loss from the combustion/heater head chamber.

The pressure drop across the combustor is 2 inches of water which allows the use of a low cost, low parasitic power commercially available air blower. However, the blower must be modified with an orifice and damper to provide dual flow rates. Materials used in the Proto 2 are presented in Table 2-3. In general, those components which are exposed to temperatures between 1200°F and 1800°F were fabricated from 310 stainless steel due to its good oxidation resistance in that temperature range. Components which were exposed to temperature less than 1200°F were fabricated from 304 stainless steel since it is less expensive than the 310 series.

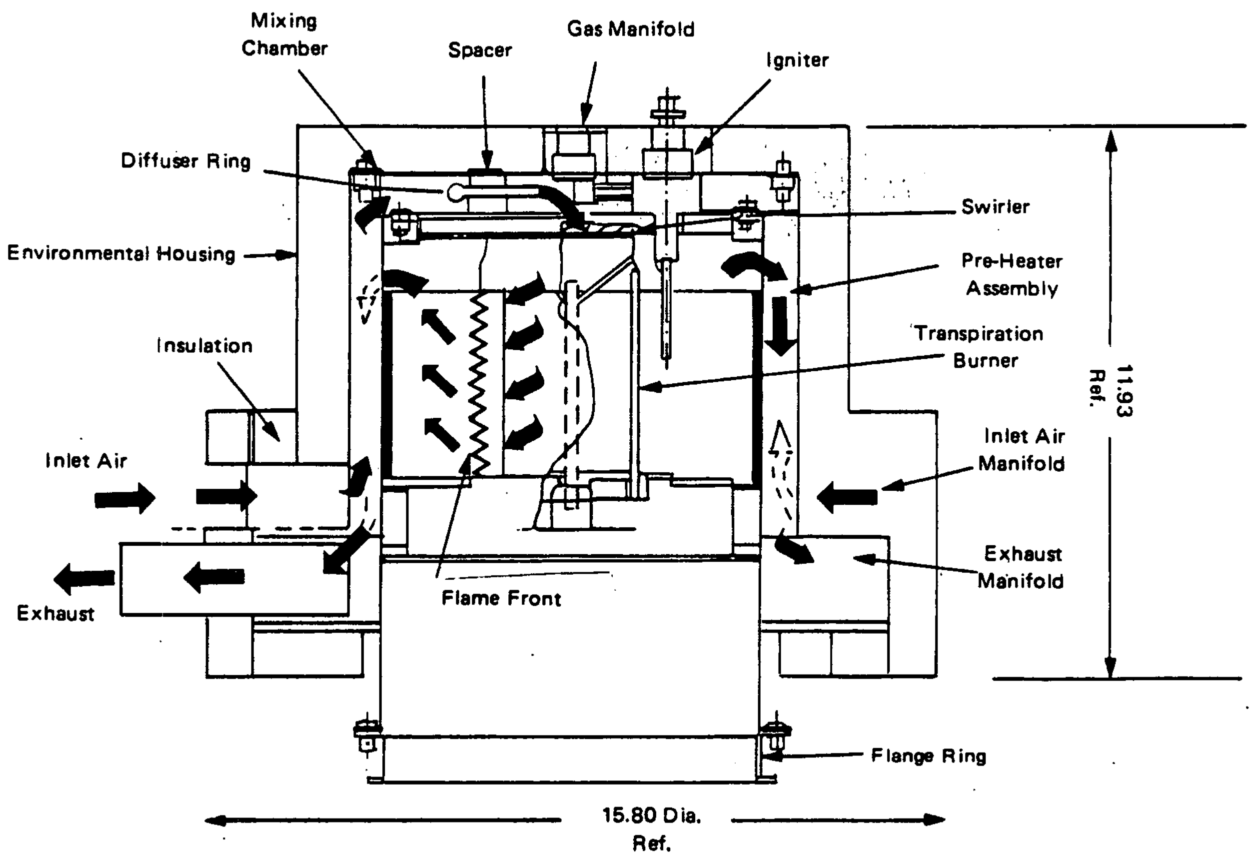

Figure 2-1. SCHEMATIC OF PROTO 2 COMBUSTOR

Table 2-3

PROTO 2 MATERIAL SELECTIONS
(General Electric)

Inlet Manifold	316L Stainless steel
Exhaust Manifold	316L Stainless steel
Housing	316L Stainless steel
Cover	316L Stainless steel
Heat Shield	347 Stainless steel
Radiation Shields	310/316L Stainless steel
Diffuser and Spacers	304L Stainless steel
Burner Support	347 Stainless steel
Burner Element	Hoskins 875, 80% porosity, 0.0056 inch dia. wire
Swirler	347 Stainless steel
Gaskets	Felted asbestos sheet packing
Insulation	Kaowool
Flame Detector	Kanthal
Igniter	Kanthal
Engine/Combustor Seal	Graphite impregnated PTFE

The key features of the Proto 2 design are summarized below.

- Transpiration Burner - Use of a transpiration burner enables NO_x emission targets (100 ppm) to be realized without exhaust gas recirculation (EGR) or combustion gas recirculation (CGR). The transpiration burner accomplishes low NO_x emissions by increasing heat transfer from the flame resulting in temperatures well below the adiabatic flame temperature. The flame temperature for the Proto 2 is estimated to be 2900°F at 15% excess air as compared to an adiabatic flame temperature of 3600°F. This produces a NO_x emissions in the range of 20-30 ppm. Unfortunately, quenching the flame also increases CO production. The CO level at 15% excess air is in the range of 200-300 ppm which is considered high. During integrated testing, the CO level was almost 5000 ppm at 15% excess air indicating that higher, excess air levels or flame temperatures will be required to safely meet the ANSI limit of 400 ppm (see Figure 2-2).

- Combustor Mounting - GE resolved the problem of engine vibration (1/8 inch amplitude, 30 Hz) on the structural integrity of the combustor and laminar flame stability by hard mounting the combustion assembly to the outdoor flame to isolate the combustor from the Stirling engine. This design also eliminated the need for flexible connections to the inlet gas line, and inlet and exhaust air duct.

- Radiation Shields - Improved heat transfer to the heater heat is accomplished by the use of radiation shields which limit heat transfer from the combustion chamber to the preheater sections and to direct the energy transfer via radiation back to the finned tube heat exchanger.

- Burner Temperatures - Burner inside surface temperatures are only 900-1000°F, which minimized flashback problems during combustor component testing. Outside burner temperature of 1700-1800°F require use of Kanthal AL or Hoskins 875 materials.

- Air/Fuel Supply - The Proto 2 burner has exceptional temperature and flow uniformity for a laminar flame. This is accomplished through the use of a circular torus fuel nozzle and swirl vanes.

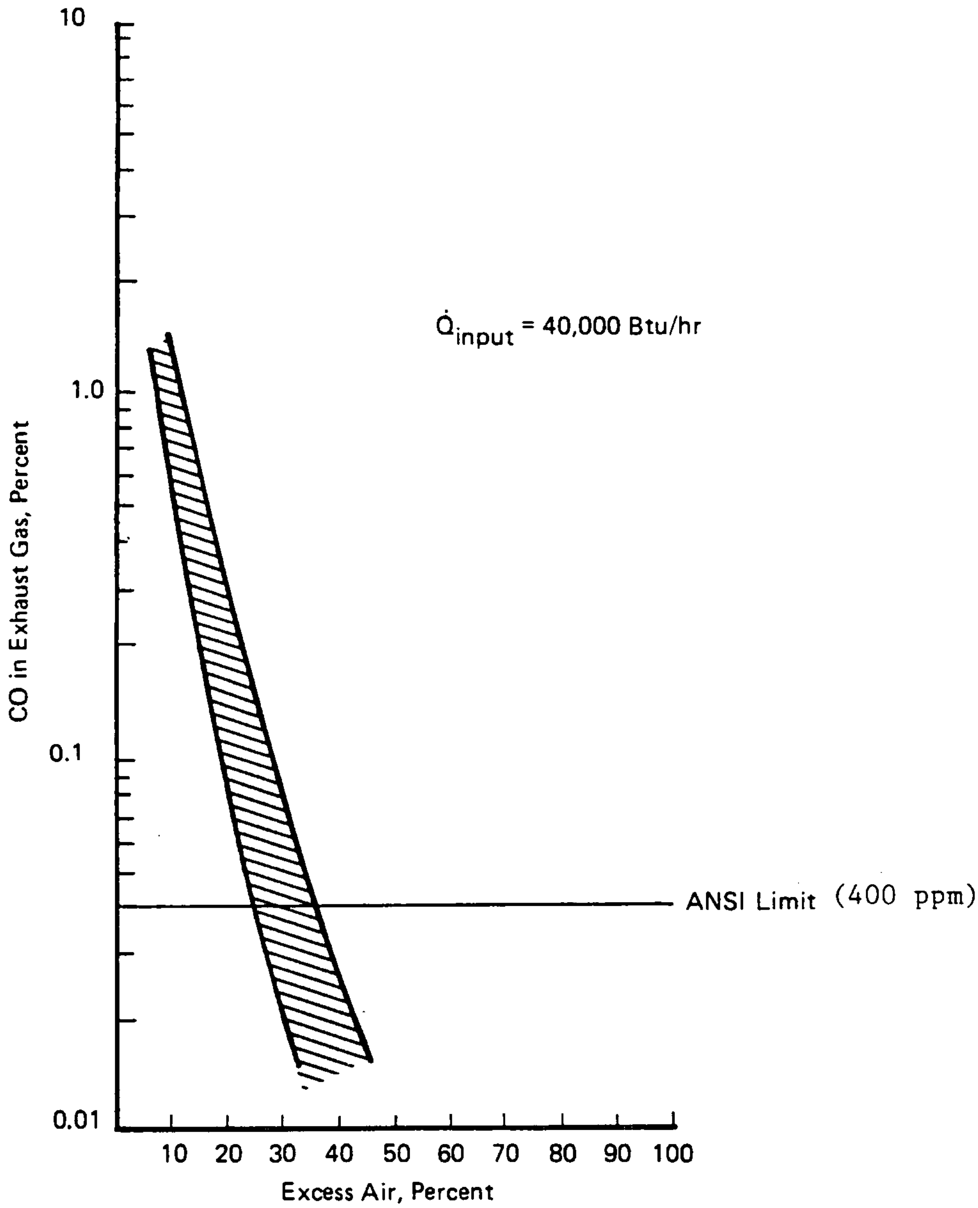

Figure 2-2. MEASURED CO EMISSIONS OF GE PROTO 2 DURING INTEGRATED TESTING

- **Burner Manufacturing Cost** - The Proto 2 burner has no seam as it is an integrated pressed axial knitted structure. This manufacturing cost is estimated at $10 per burner. Eliminating the seam saved about $5 per burner in manufacturing cost.

Key problems of the Proto 2 design are:

- CO emissions are high and may require operation a higher excess air levels.

- Life of burner and heater head materials has not been proven to meet 20,000-50,000 hr. GE had designed the combustor for 5500 hr with yearly maintenance.* Further development work will be required to meet 20,000-50,000 hour lifetimes.

- Flashback and preignition problems were not solved during integrated testing. These problems would cause the unit to fail ANSI tests and also reduce burner lifetime well under one year. GE did begin work with protective coatings for the burner but did not test their performance.[15]

- Two stage firing at constant excess air has not been demonstrated for long term operation. Teledyne Laars has incorporated the same 2 stage gas valve into their modulating boiler which resulted in substantial efficiency losses due to improper fuel/air control.

2.1.2 Mechanical Technology Inc.

MTI has been working on hot end components for free piston Stirling engine residential heat pumps since 1976.[3] The initial combustor relied heavily on designs employed in MTI's automotive Stirling Development Program.[13,14]

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*It is unclear whether GE intended to change combustor components during annual maintenance.

During the course of the combustor development for residential Stirling heat pump applications, many design considerations were identified through trial and error modification of the combustor prototype. The present design was obtained after numerous "cut and try" laboratory units were built and tested as a separate component.

The major accomplishments during the course of this development were:

- o increasing combustor stack loss efficiency from 70 to 80%;
- o providing uniform heater head temperature;
- o improving ignition reliability;
- o reducing pressure drop from 25" to 10" w.c.;
- o reducing noise; and
- o reducing combustor leakage from 15-20% to 6-8%.

MTI systematically evaluated 6 burner concepts for the HAHP: 1) pulsed combustion; 2) natural aspiration; 3) premixed; 4) laminar diffusion flame; 5) venturri and diffusion flame; and 6) turbulent diffusion flame. MTI rejected the first four concepts for the reasons listed in Table 2-4. Concept 6 was chosen primarily because it relied on the combustor design and required low development despite the fact that a hot exhaust fan is required. Concept 5 also uses one of MTI's existing combustor designs and was chosen as a backup.

Table 2-4

SUMMARY OF MTI CANDIDATE BURNER SELECTION

Burner Concepts Rejected by MTI	Reasons
Pulsed Combustion	Significant development required
Natural Aspiration	Insufficient flow of combustion products
Premixed	Preignition at desired preheat temperature
Laminar Flame	Larger size, moderate development

Description of MTI Hot End Assembly

The combustor consists of a natural gas turbulent burner and a preheater. Air enters the preheater at 70°F and is heated to about 1350°F by heat exchanging with the exhaust gases. The preheater is an annulus with a mean diameter of about 10 inches, radial thickness of about .4 inches and length of about 10 inches. The two streams are separated by a stainless steel corrugation of 18 fins/inch (245 convolutions). The preheater effectiveness is 94% with a pressure drop of 2 inches water. The preheated air is mixed with fuel in the burner and ignited, with flame temperatures of approximately 3550°F. The hot gas is then passed over the heater head, providing heat input to the engine. The gas exits the head at 1550°F, and is further cooled by transferring heat to the air intake stream in the preheater before exiting the system.

A drawing of the MTI external heat system consisting of the combustor and heater head is shown in Figure 2-3.

The heater head design is based on a monolithic heater head concept that has been under development at MTI for the past several years.[19] The monolithic heater head made of cast inconel consists of one piece head construction with fins on the inner and outer side. The one piece construction eliminates the fitting and brazing of the tubes required in a tubular design.

The key performance characteristics of MTI's hot end assembly are summarized in Table 2-1. Key features of the MTI hot end system are:

o use of monolithic heater head to improve reliability and reduce manufacturing cost;

o high preheat temperatures 1300-1400°F which precludes use of premixed burner and results in high flame temperature;

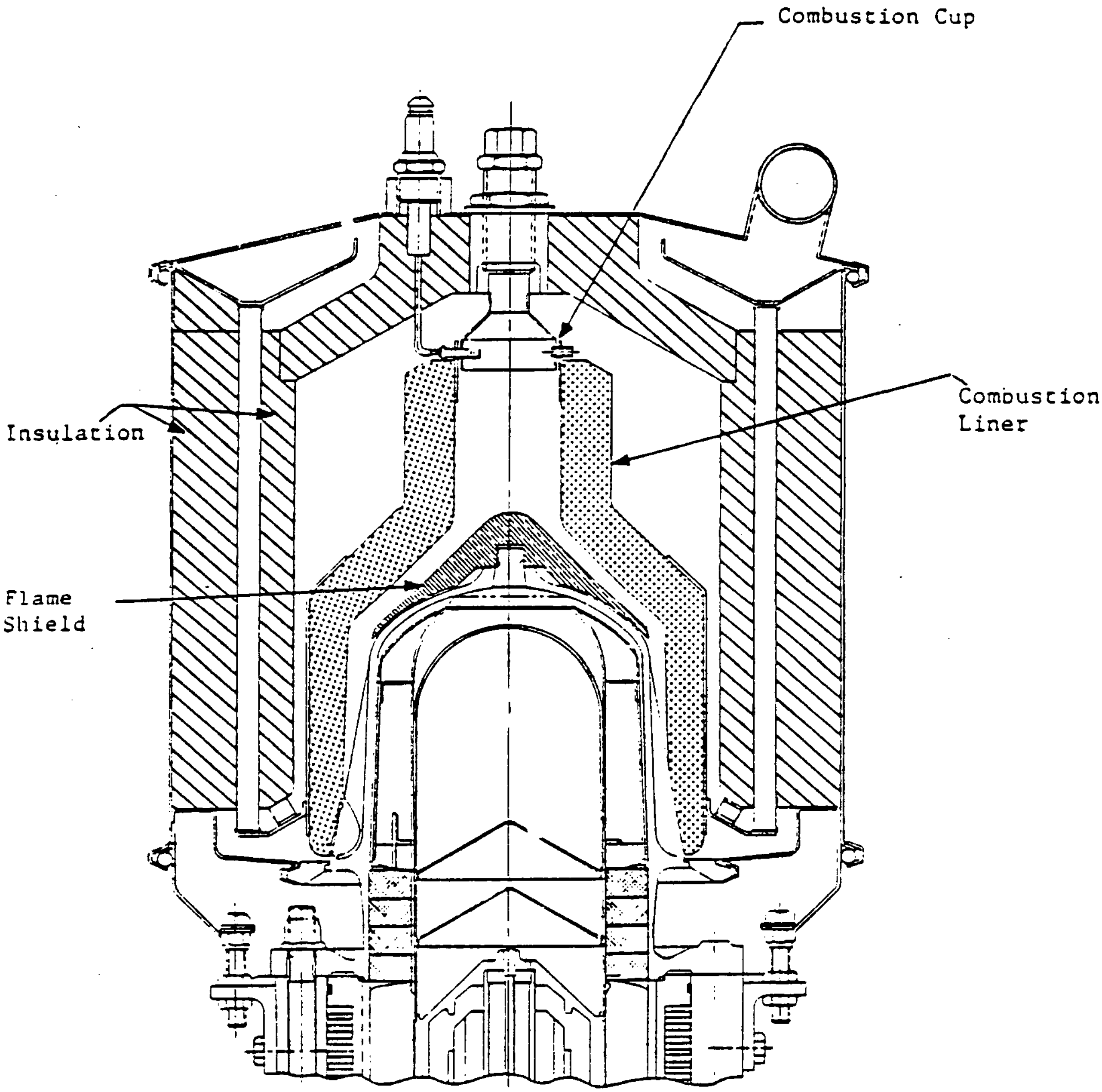

Figure 2-3. DRAWING OF MTI EXTERNAL HEAT SYSTEM

Critical problems with system which must be addressed in the future are:

1) Inconel 713LL is used for heater head material and is creep limited. Based on materials research completed for automotive Stirling engines, 700 series Inconel has not survived endurance tests by more than 750 hours for 1500°F heater head temperature and peak pressure of 150 atm.

2) Off cycle losses of hot end component are about 1500-2000 Btu/hr which can reduce seasonal combustor efficiencies by 10-20% (see section 3.2).

3) Due to the high flame temperatures, NO_x levels are high (700 ppm and 1300 ppm at 50% and 25% respectively), which will require either EGR and CGR to reduce to acceptable levels (100 ppm).

4) Combustor is not isolated from engine vibration which reduces structural integrity of liner. Only 480 test hours have been accumulated with present ceramic liner.

5) Burner has not been tested in cyclic operation. The thermal stresses during cyclic operation which result from a 10,000°F/in. temperature gradient across the burner wall may cause failure of the inner ceramic.*

2.1.3 Sunpower

Sunpower has not developed a Stirling engine heat pump but they have built many free piston Stirling engines for demonstration. Presently Sunpower uses 304SS or 316 SS for their heater head assemblies which are all of monolithic structure (i.e., no tubes). Sunpower feels these materials are adequate for 1110°F-1290°F operation in most applications. These materials are preferred over the Inconels due to their relative ease of machining and welding.

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*MTI does not report type of ceramic. Flexible fiber ceramics are more promising for cyclic conditions.

Sunpower indicated that with a monolithic heater head design, operating temperatures can be increased to achieve higher operating efficiencies that use existing technology. Garrett claims they could even now make a ceramic heater head for a low pressure (10 bar) engine and Sandian in Japan offered to supply ceramic heads to Sunpower.

2.2 Automotive Experience

The automotive application of the Stirling engine hot end components has been extensively tested. Pollutant emissions and life testing of the combustor has been conducted for the four test engines:

Ford Torino 4-215 swashplate
Ford Prototype 4-215
MTI Spirit P-40
MTI "Concorde"
Opel Rekord P-40 - United Stirling

With the exception of required lifetime, the automotive application is more severe than the HAHP. The required stress level of the heater head is higher and hydrogen is used instead of helium for the Stirling working fluid which may cause embrittlement of certain heater head materials. Lastly the combustion of liquid fuels as compared to natural gas has three deleterious effects: 1) emissions are more difficult to control because mixing of fuel-air is limited by atomization and evaporation of the liquid fuel droplets; 2) contaminants in liquid fuels, (sulphur, vanadium) add corrosion problems to burner element and heater head; and 3) soot deposition occurs in last half of the preheater.

Although the operating environment for the automotive application is more severe in the HAHP application, the design life is considerably shorter. The design life for the automotive study is typically 3500 hr as compared to 25,000-50,000 hr for the HAHP.

Design considerations for burners used in automotive applications have limited use for the HAHP due to the basic differences in rate controlling steps between burning liquid and gaseous fuels. Liquid fuels must be atomized and vaporized prior to and during the mixing process with air. Gaseous fuels need only be mixed with air before combustion.

The developments completed under the automotive Stirling R&D programs which has a direct bearing on the HAHP relate to emissions, preheater design and materials consideration. They are discussed briefly below.

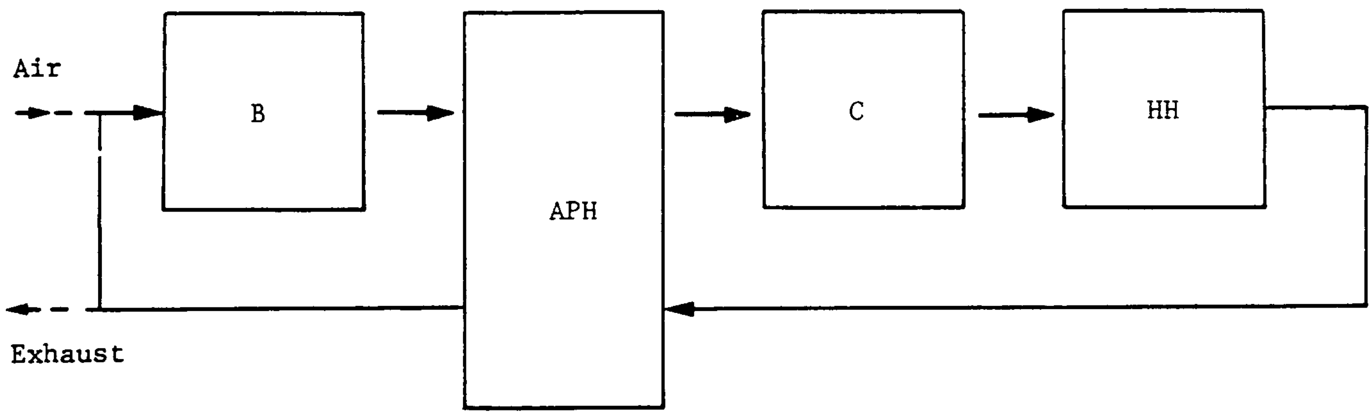
2.2.1 Emissions

The MTI Sport(5) and Ford Prototype 4-215(16) meets both the 1985 Federal (HC-0.25, CO-2.11, NO_x-0.62 g/km) and California (HC-0.25, CO-5.59, NO_x-0.25) emissions requirements. NO_x control was the primary difficulty in meeting these requirements. Automotive combustors operate in the 3300-3600°F range which makes the formation of excessive NO_x certain. To overcome this problem, Exhaust Gas Recirculation (see Figure 2-4a) has been implemented to lower flame temperature. The disadvantage of the EGR system is that the flow rates through the preheater is large in a considerable pressure drop. The blower capacity must be larger and increases parasitic losses. Combustion Gas Recirculation (CGR) is currently being developed by MTI(14) to overcome these problems (see Figure 2-4b). With the CGR system, ejectors are used to withdraw the combustion gas directly from the heater head exhaust back into the combustor inlet thereby bypassing the preheater.

Significant studies have been made in the automotive Stirling programs with respect to emissions control. The MTI Spirit P-40 has demonstrated in an actual vehicle to meet 1985 California emission standards (0.25g/km) while the Ford Prototype 4-215 also met this target in dynamometer tests. Based on this success, it is reasonable to expect that NO_x can be controlled to within 100 ppm for free piston Stirling heat pumps.

2.2.2 Preheater Development

In the automotive program, preheaters have been built and tested which exhibit preheater effectiveness of 85-90%, leakage in the range of 10-15% and proven lifetimes of 3500 hours. However, these preheaters have several problems:



B - Blower
C - Combustor

APH - Preheater
HH - Heater Head

Figure 2-4a. EXHAUST GAS RECIRCULATION SYSTEM

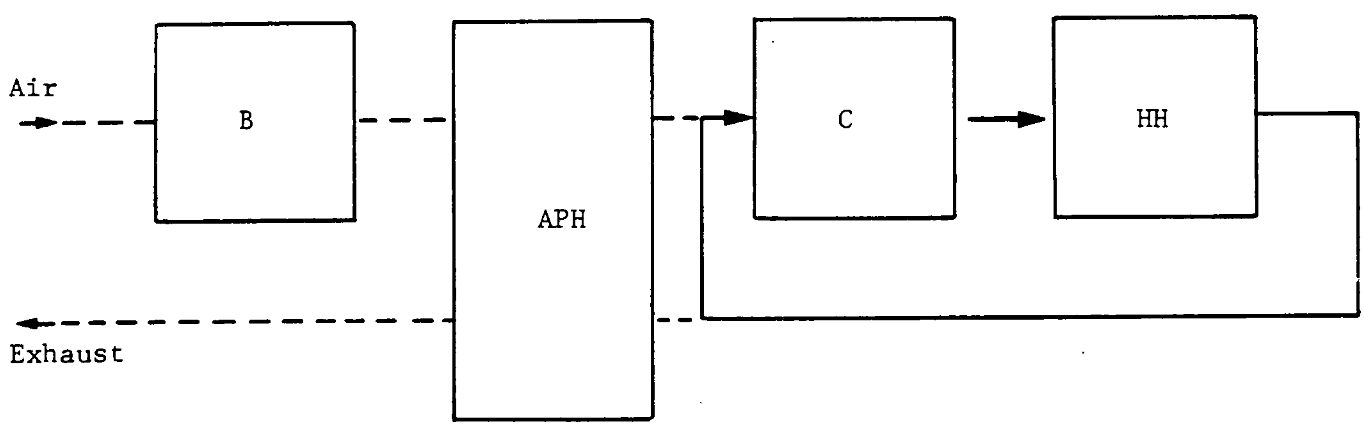


Figure 2-4b. COMBUSTION GAS RECIRCULATION SYSTEM

1) They are very expensive to manufacture as the assembly often involves seam welding thin plates (.004 - .01 inch). For example, the upgraded MTI Mod I engine requires over 2000 weld seams.

2) Preheaters are made of stainless steel which is also problematic because the material is susceptible to attack by acids formed in the exhaust gas of fuels containing sulphur.

3) Preheater designs do not enable cleaning when fouled.

To resolve these three problems, MTI is working on a ceramic recuperative preheater made from a material similar in nature to cordierite (mixed oxide ceramic made from M_6O, Al_2O_3, and SiO_2). The idea is to assemble the preheater in a prefired state followed by firing to provide the necessary plate to plate sealing for separation of the two gas streams.

Preheater considerations for the gas-fired heat pump are different than the automotive application. Errosion due to fuel bound sulphur and heat exchanger cleaning are not problems with gas-fired systems. The gas-fired heat pump requires a lower cost per unit area heat exchanger and must be longer life 25,000-50,000 hours. Due to these different requirements between the gas-fired heat pump and automotive applications, it is not clear that advantages of ceramic materials outweigh the disadvantages.

The problems of thermal stress due to the high temperature gradients (up to 10,000°F/in) during cyclic operation could have severe problems with the use of ceramics. In addition, joining the ceramic preheater to other components of hot end (air inlet and heat exhaust plenum) is complicated by dissimilar thermal expansion between metal and ceramic materials. A detailed discussion of the use of ceramics in the Stirling hot end is presented in Section 3.4.

2.2.3 Materials

Probably the most valuable research conducted under the automotive which can be applied to gas-fired Stirling heat pumps has focused on material reliability. Much of the data developed to predict 3500 hour design life for rupture stress levels can be used to predicted rupture strength for longer time periods required for gas-fired heat pumps (25,000-50,000 hours).[1,7,9]

The materials research is underway at NASA Lewis, MTI and United Stirling. All three programs focus on the high temperature materials for the heater head. NASA Lewis has concentrated on baseline creep rupture properties, oxidation resistance and hydrogen permeation. NASA Lewis has also developed data on candidate materials using simulated engine conditions on the heater head tubes. MTI is conducting fatigue evaluation and failure analysis. United Stirling has been creep rupture testing candidate materials with gas temperatures up to 1560°F. The results of this work has been useful in evaluating heater head materials for the gas-fired Stirling heat pump. This evaluation is discussed in Section 3.3.

In summary, information developed under the automotive Stirling research programs can be applied to the heat pump applications in three areas: 1) emissions; 2) air preheater design, and 3) material analysis. Burner technology cannot be transferred from the automotive to the heat pump application because of the fundamental differences in forming liquid and gaseous fuels. Among the three relevant subjects, the material research is most applicable to the heat pump application since data has been developed over a range of temperatures and stress levels which includes the conditions for the heat pump.

3.0. DISCUSSION OF KEY ISSUES IN LIGHT OF SYSTEM REQUIREMENTS

The key issues identified in Section 2 are:

- CO and NO_x emission levels;
- Seasonal emissions efficiency of combustor considering cyclic losses;
- Materials requirements for 25,000-50,000 hour life; and
- Use of ceramic materials in hot end components.

3.1 Emissions

The chemical kinetics of CO and NO_x formation are fairly well understood and technology to burn fuel with high preheat temperatures (1000-1400°F) with acceptable emission levels has been demonstrated in automotive Stirling programs. Although achieving minimal levels of CO and NO_x requires a delicate balance of flame temperature and excess air for gas-fired applications, the technology and know how is available to accomplish this goal. MTI has not yet done so because they have not incorporated EGR or CGR into their prototype. However, based on their accomplishments in the automotive program, they should be able to reduce NO_x for the gas-fired heat pump below 100 ppm. General Electric succeeded in operating the combustor as a separated component with acceptable NO_x and CO emission levels. However, integrated tests of the combustor indicated excessive CO which resulted from burner wall temperatures below the design. Apparently the test rig for the combustor performance testing did not can adequately simulate the thermal characteristics of actual operation. Although additional development would be required to modify the GE prototype to operate with acceptable CO and NO_x emissions, the work is technically feasible.

All combustor prototypes have demonstrated acceptable hydrocarbon emissions (under 50 ppm), since both prototypes have adequate residence time and excess air to result in near complete oxidation of unburned hydrocarbons.

3.2 Seasonal Efficiency

The gas-fired Stirling heat pumps are designed to operate in a cyclic mode.* The frequency of the cycles ($\dot{N}$) will depend on (1) the thermostat differential, (2) the house heat-up/cooldown rate during the heat pump on cycle (which is determined by the difference between heat pump output rate and the heating or cooling load in the house), and the house heat up/cooldown rate (proportional to load) during the off cycle. The heating and cooling load varies considerably over the course of a heating season. During the spring and fall, the heat level may only be about 10-20% the heat output rate. The system then comes on infrequently and stays on for only a small fraction of the time. In very cold or hot weather, the heat load approaches the output rate. As a result, the system will stay on for a larger fraction of the total time. When the load equals or exceeds the input rate, the system operates continuously.

Among the Stirling heat pump contractors, combustor efficiency has been assumed to be equal to the stack loss efficiency η_{SL}. The stack loss efficiency is defined as:

$$\eta_{SL} = 1 - \frac{\dot{Q}_{flue}}{\dot{Q}_{in}} \quad 100$$

The stack loss efficiency only considers the on-cycle flue loss and not the on-cycle jacket losses. In addition, the stack loss efficiency only considers operation at steady state; cyclic losses are ignored.

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*MTI has predicted fractional on time as outdoor temperature.[(2)] GE fractional run time was never estimated and will depend on nature of two-stage firing.

The stack loss efficiency can be calculated based on the flue temperature and excess air level. Stack loss efficiency for gas-fired combustors as a function of these two variables is shown in Figure 3-1. Note that to achieve stack loss efficiency of about 80%, exhaust temperatures must be in the range of 375-525°F, the exact value depending on the exact value of excess air. The higher the excess air level, the lower the required flue temperature to achieve the same efficiency. The heater head temperature in both the MTI and GE prototypes is 1250-1300°F. Since the temperature exiting the heater heat is 200-300°F above this value, heat must be recovered from the exhaust gas to reduce its temperature to 375-525°F in order to achieve acceptable efficiency. The recovered heat is used to preheat the combustion air. The required levels of combustion air preheat as a function of heater head exhaust gas temperature to achieve stack loss efficiency of 80% is shown in Figure 3-2. From the figure, it is evident that air preheat levels of 1300-1400°F are required for 1500-1600°F heater head exhaust gas temperatures to achieve stack loss efficiencies in the vicinity of 80%. This analysis is based on a simple energy balance around the preheater and neglects leakage.

The regime of operation for the MTI and GE prototypes are shown in the figure. Note that GE operates at a much lower heater head exhaust gas temperature than MTI (1300 vs. 1500°F) even though the heater head is at the same temperature (about 1300°F) for the two prototypes. GE can operate at a lower gas temperature because the heater head receives considerable radiative heat flux (7,000-12,000 Btu/hr ft^2) from the transpiration burner.

The temperature data reported by MTI and GE are in general agreement with Table 2-1. Reported stack loss efficiencies are usually optimistic by one or two percentage points.

Single Model for Part Load Combustor Efficiency

An approximate expression for the combustor efficiency (η_{HH}) corresponding to a given load factor can be derived when the cycling rate is constant. The combustor efficiency is defined as:

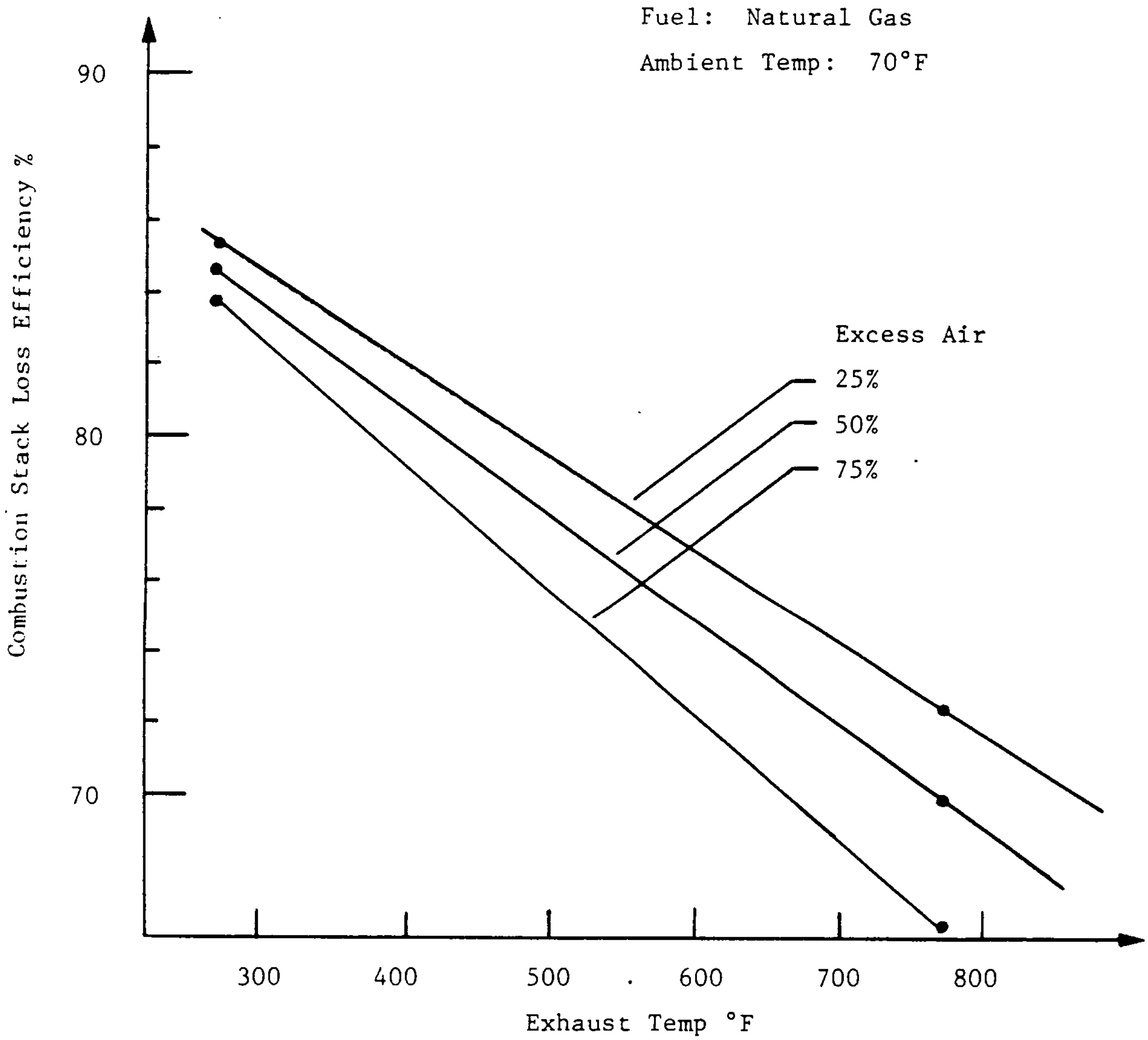

Figure 3-1. STACK LOSS EFFICIENCY AS FUNCTION OF EXHAUST TEMPERATURE AND EXCESS AIR LEVEL FOR EXTERNAL HEAT SYSTEM

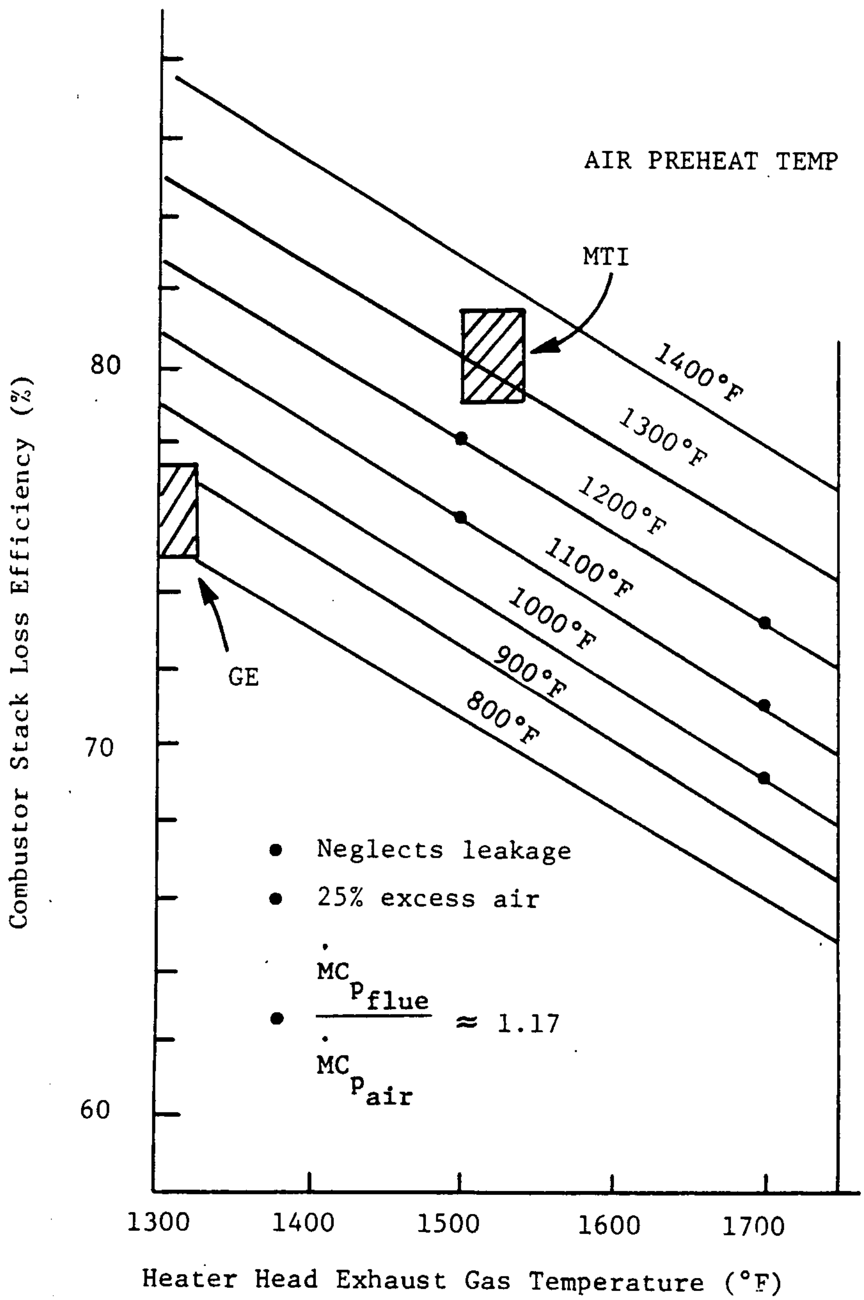

Figure 3-2. COMBUSTOR EFFICIENCY AS FUNCTION OF HEATER HEAD EXHAUST GAS AND AIR PREHEAT TEMPERATURE

$$\eta_{HH} = \frac{\dot{Q}_{HH}}{\dot{Q}_{IN}} = \frac{\text{Energy Transferred Across Heater Head}}{\text{Energy Consumption in Forms of Natural Gas}} \qquad (1)$$

An energy balance on the system for one cycle can be expressed as follows:

$$\dot{Q}_{IN}\, t_{ON} = Q_{HH} + \dot{Q}^{ON}_{LOSS}\, t_{ON} + <\dot{Q}^{OFF}_{LOSS}>\, t_{OFF} \qquad (2)$$

where: t_{ON} = the burner on-period per cycle

t_{OFF} = the corresponding off-period per cycle

$\dot{Q}^{ON}_{LOSS}$ = the heat loss from the system during the on cycle

$<\dot{Q}_{OFF}>$ = the average rate of heat loss from the system during the off-cycle

Note that $\dot{Q}^{ON}_{LOSS} = (1 - \eta_{SS})\, \dot{Q}_{IN}$ where η_{SS} is the steady state combustor efficiency.

Dividing both sides of Eq. (2) by the left hand side and rearranging, we obtain an expression for the combustor efficiency:

$$\eta_{HH} = \frac{Q_{HH}}{\dot{Q}_{IN}\, t_{ON}} = 1 - \frac{<\dot{Q}^{OFF}_{LOSS}>\, t_{OFF} + \dot{Q}^{ON}_{LOSS}\, t_{ON}}{\dot{Q}_{IN}\, t_{ON}} \qquad (3)$$

Substituting $\dot{Q}^{ON}_{LOSS}$, we find that the combustor efficiency can be expressed in terms of the steady state efficiency, and the standby loss:

$$\eta_{HH} = \eta_{ss} - \frac{<\dot{Q}_{OFF}>}{\dot{Q}_{IN}} \left(\frac{1 - x}{x}\right) \qquad (4)$$

$$\text{where} \quad x = \frac{t_{ON}}{t_{ON} + t_{OFF}} \tag{5}$$

The off-cycle loss rate is initially equal to the cool-down rate of the heat exchanger. The heat exchanger cools down due to jacket losses and off cycle flue losses.* If we represent the heat exchanger as a body with thermal mass C_{HX} (product of mass and specific heat), cooling off by natural convection with an effective heat-transfer coefficient of hA, we can derive the following equation for the variation of heat exchanger heat loss rate with time:

$$\dot{Q}_{HX} = \dot{Q}_{HX} e^{-t/t_{HX}} \tag{6}$$

where t_{HX} is the cool-down time constant defined as:

$$t_{HX} = \frac{C_{HX}}{hA} \tag{7}$$

and $\dot{Q}^{o}_{HX}$ is the initial value at burner shut-off (time zero).

Performing the required integration provides a closed-form expression for the average loss rate per cycle. This results in the following expression:

$$\langle \dot{Q}_{OFF} \rangle = \dot{Q}^{o}_{HX} \left(\frac{1 - e^{-t_{OFF}/t_{HX}}}{t_{OFF}/t_{HX}} \right) \tag{8}$$

$$\langle \dot{Q}_{OFF} \rangle = \frac{\dot{Q}_{HX}}{t_{OFF}} \left(1 - e^{-t_{OFF}/t_{HX}}\right) \tag{9}$$

---

*Even for the power burner, the system is not perfectly sealed during the off cycle and some air circulates through the system because of its low pressure drop.

The burner off-period per cycle can be expressed in terms of the cycling rate ($\dot{N}$), as follows:

$$t_{OFF} = \frac{1 - x}{\dot{N}} \tag{10}$$

Substituting equation (10) into equation (9), we arrive at:

$$\langle \dot{Q}_{OFF} \rangle = \dot{Q}^{o}_{HX} \left(\frac{1 - e^{(x-1)/\dot{N}t_{hx}}}{(1-x)/\dot{N}t_{hx}} \right) \tag{11}$$

The part load efficiency can then be calculated by substituting equation (11) into equation 4:

$$\eta_{HH} = \eta_{ss} - \frac{\dot{Q}^{o}_{HX}}{\dot{Q}_{IN}} \left(\frac{1 - e^{(x-1)/\dot{N}t_{hx}}}{x} \right) \dot{N}t$$

Estimates based on reported data on exterior surface temperature measurements for the combustor indicate that th: jacket loss ($\dot{Q}_j$) is about 2-3% of the input rate. Therefore:

$$\eta_{ss} = \eta_{SL} - \frac{\dot{Q}_j}{\dot{Q}_{IN}}$$

The stack loss efficiency (η_{SL}) can be estimated from Figure 3-1 based on the excess air and exhaust temperature.

Data presented at the Shaumberg Meeting(6) can be used to determine t_{hx} and $\dot{Q}^{o}_{HX}$. The data is sketchy, but indicates that the hot end assembly of the combustor cooled down about 200°C in one hour. Using this information together with the initial average hot end temperature of 1150°F t_{hx} and Q^{o}_{HX}, the calculated cooldown rate for the MTI hot end is shown in Figure 3-3. The analysis indicates that:

$$\dot{Q}_{hx} = 1520 \text{ Btu/hr}$$

$$t_{hx} = 2.5 \text{ hr.}$$

The part load combustor efficiency can now be calculated as a function of fractional on time (X) and cycling rate ($\dot{N}$). Table 3-1 summarizes the results for the MTI prototype.

Note that the part load efficiency is a weak function of cycling rate for a given fractional on-time. This is due to the fact that for off times of interest, the temperature time history of the hot end is almost linear due to the large heat exchanger time constant. Therefore $\dot{Q}_{OFF}$ is relatively independent of the off time.

To determine the net effect of this part load efficiency behavior on seasonal efficiency, we must consider the fuel consumed at each load factor which depends on the annual outdoor temperature history for a given location. The seasonal heating efficiency was determined from equal fuel usage at each of the three temperature bins for the Boston, Los Angeles, and Chicago:

$$\eta_{HH} = \frac{1}{3}\left(\eta(T_1)^{-1} + \eta(T_2)^{-1} + \eta(T_3)^{-1}\right)^{-1}$$

Figure 3-4 illustrates the part load efficiency versus load factor with different leakage rates for the MTI prototype. Note that for a stack loss efficiency of 77%, the seasonal heating efficiency in Los Angeles (assuming a typical leaking rate of 7%) is only 55%. The heating and cooling seasonal efficiency for Chicago is approximately 65%.

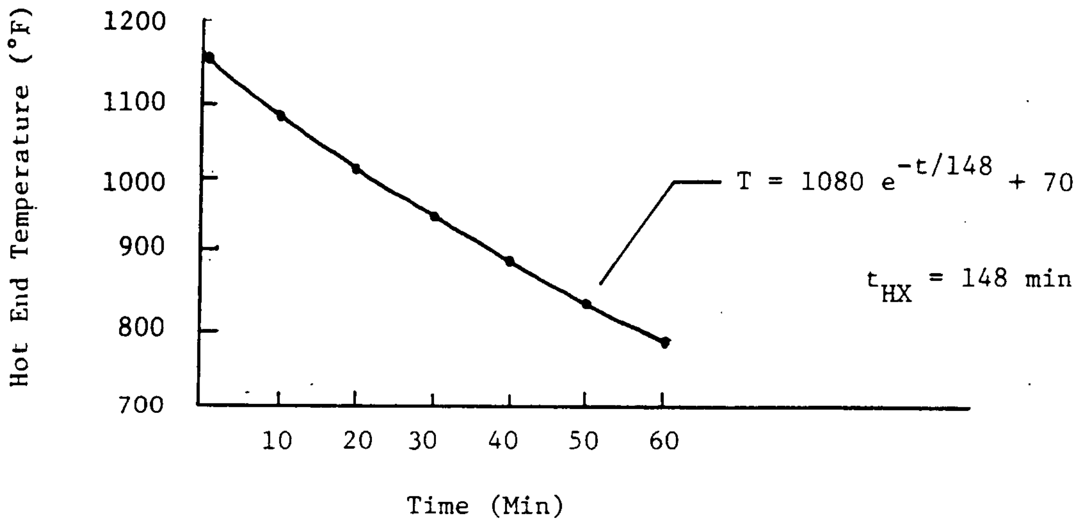

Figure 3-3. COOLDOWN OF MTI EXTERNAL HEAT SYSTEM

Source: Based on heat exchanger cooldown model and observation system cools 200°C in one hour (6).

Table 3-1

PART LOAD EFFICIENCY OF MTI COMBUSTOR DEPENDENCE ON FRACTIONAL ON TIME AND CYCLING RATE

Fractional On Time	PART LOAD EFFICIENCY				
	$\dot{N}$ cycles/hr				
X	1	2	3	5	10
1.0	73.4	73.4	73.4	73.4	73.4
.9	73.0	73.0	72.9	73.0	73.0
.8	72.5	72.5	72.4	72.5	72.5
.7	71.9	71.8	71.8	71.8	71.8
.6	71.1	71.0	71.0	71.0	71.0
.5	70.1	69.9	69.8	69.8	69.8
.4	68.5	68.2	68.1	68.0	68.0
.3	65.9	65.4	65.2	65.1	65.0
.2	60.9	59.4	59.5	58.2	59.0
.1	45.7	43.3	42.3	41.6	41.0

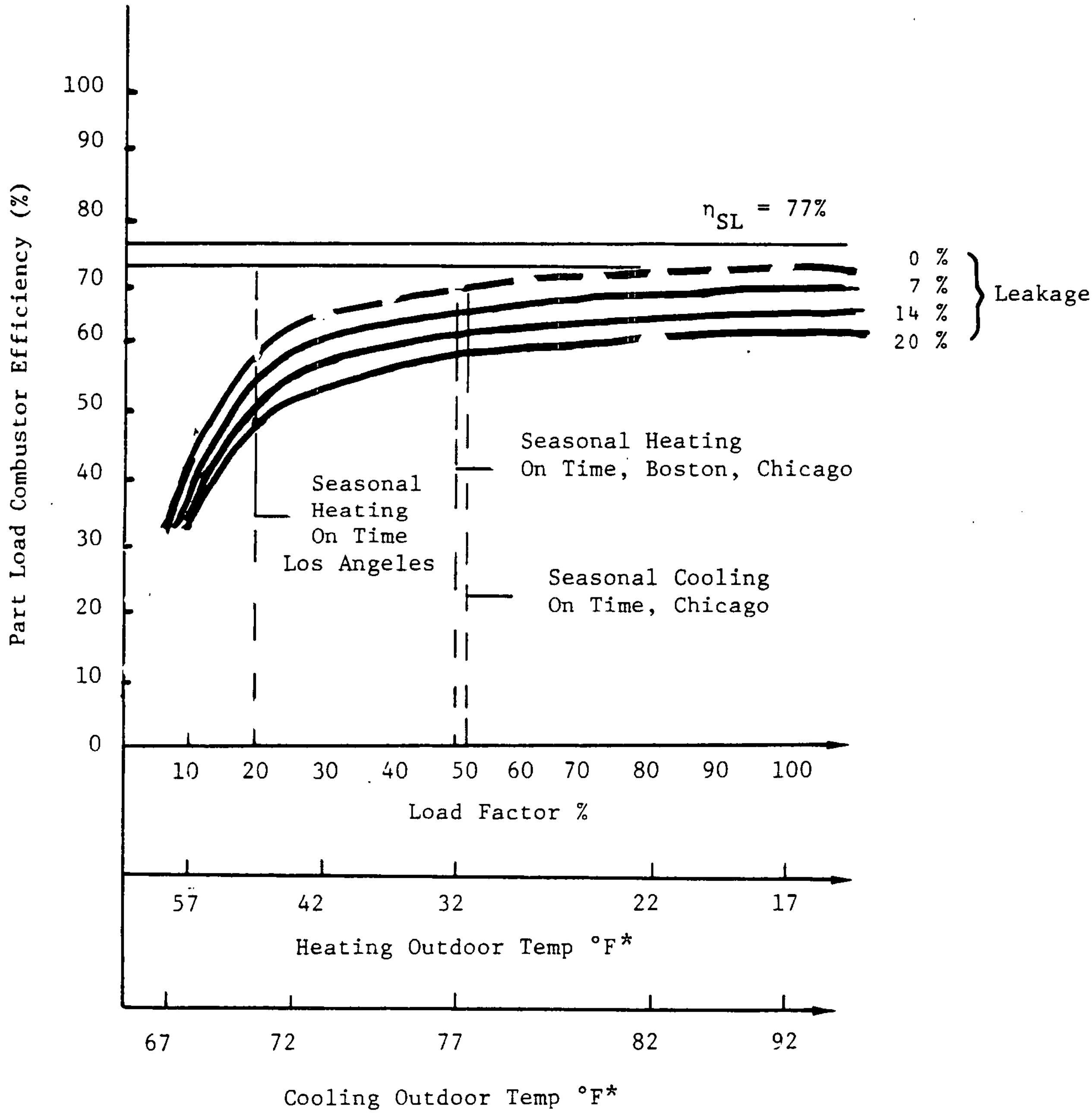

Figure 3-4. PART LOAD EFFICIENCY OF EXTERNAL HEAT SYSTEM VERSUS LOAD FACTOR

*Temperature for each load factor taken from Reference 2.

In summary, the off cycle losses from the combustor can significantly reduce seasonal efficiency below the stack loss efficiency sported by the manufacturers.

The magnitude of the effect can vary by 10-20 percentage points depending on geographic location. Although this analysis was completed from data for the MTI prototype, it generally applies to the GE unit as well. Prediction of seasonal efficiencies for the GE would require accounting of the effect of two stage firing on fractional burner on time as a function of heating and cooling load.

3.3 Material Composition of Hot End Components

For long term operation (25,000-50,000 hours) at 1300°F heater head temperatures and peak pressures of 1200-1500 psig creep life is a critical design constraint. NASA Lewis has focused on creep rupture properties and United Stirling is presently conducting creep rupture testing for candidate heater head materials.

Much of this work conducted under the automotive program can be applied to the heat pump applications. Materials and their composition which have been studied in the automotive program are listed in Table 3-2. In the automotive program, rupture strengths are presented at 3500 hours which is the design life for the automotive engine. Based on the work completed to date at NASA Lewis and United Stirling, CG-27 and Incoloy 901 are the only materials to withstand over 3500 hour endurance at 1500°F and 15 MPa pressure Hz in the simulator test rig. Since these tests are run in hydrogen materials which failed early from hydrogen embrittlement may have longer endurance times when used with helium in the HAHP application. NASA Lewis has classified materials on the test rig into categories of failure time as shown in Table 3-3. Through the use of correlations on the literature, the rupture strength for 10,000 and 50,000 hours for the heat pump application can be determined. For example, at 1600°F, the creep rupture curve for CG-27 can be approximated by the following relationship between creep rupture stress (σCR) and time to creep rupture:

$$\sigma_{cr} = 710\ (t_{cr})^{-.35}$$

Table 3-2

COMPOSITION OF ALLOYS USED FOR HEATER-HEAD TUBES

Alloy	Fe	Cr	Ni	Co	Mn	Si	Mo	Cb	W	Al	Ti	C	Other
	Composition, wt %												
N-155	30	21	20	20	1.5	0.5	3.0	1.0	2.5	--	--	0.15	0.15 N
CG-27	38	13	38	--	.1	.1	5.5	.6	--	1.5	2.5	.05	0.01 B
Incoloy 901	34	14	43	--	.45	.40	6.2	--	--	.25	2.5	.05	0.015 B
Inconel 625	3	22	61	--	.15	.3	9.0	4.0	--	.2	.2	.05	----
W545	54	14	26	--	1.5	.4	1.5	--	--	.2	2.9	.08	0.08 B
12RN72	47	19	30	--	1.8	.29	1.4	--	--	--	.5	.1	0.02 N
253 MA	65	21	11	--	.4	1.7	.40	--	--	--	--	.09	0.7 N
HS-188	1.5	22	22	40	--	--	--	--	14.0	--	--	.08	0.08 La
Inconel 750	7	16	73	--	.5	.2	--	1.0	--	.7	2.5	.04	----
Sanicro 32	43	21	31	--	.6	.47	--	--	2.8	.4	.4	.89	----
19-9DL	67	19	9	--	1.1	.6	1.3	.4	1.2	--	.3	.3	----
Inconel 718	19	19	52	--	.2	.3	3.1	5.0	--	.4	.9	.04	----
Inconel 601	14	23	60	--	.5	.2	--	--	--	1.4	--	.05	----
A286	53	15	26	--	1.4	.4	1.3	--	--	.2	2.2	.05	0.26 N
Sanicro 31H	46	21	31	--	.6	.55	--	--	--	.4	.5	.07	0.02 N
Incoloy 800H	46	21	33	--	.75	.5	--	--	--	.38	.38	.05	----
SS310	55	25	20										
Inconel 600	8	14	78										
Hastelloy	18	22	47										

Table 3-3

ENDURANCE TEST RESULTS
[Temperature, 820°C; pressure, 15 MPa)

Time to failure, t, hr				
t > 3500	2500 < t < 3500	1500 < t < 2500	750 < t < 1500	t < 750
CG-27 Incoloy 901	Inconel 625 W545	12RN72 Cold worked 12RN72 Annealed	253 MA HS-188 Inconel 750 Sanicro 32	19-9DL N-155 Inconel 718 Inconel 601 A286 Sanicro 31H Incoloy 800H

Table 3-4

MAXIMUM CREEP RUPTURE STRESS FOR 10,000 HOUR OPERATION

Alloy	Creep Rupture Stress for 10,000 hour operation (MPa) 1300°F
304 stainless	50
321 stainless	60
Multimet (N-155)	160
Inconel 617	200
19-9DL	90
A-286	105
Haynes stellite 31	200

Note that stress value depends on the pressure, diameter and wall thickness of heater tube:

$$\sigma = \frac{P(D-h)}{2h}$$

where: P = internal pressure

D = outside tube diameter

h = wall thickness

For tube diameters typical of the GE prototype (1/4" d, .028" wall) the stress value is $\sigma \sim 4P$.

Since peak pressures are in the range of 1000-1500 psig, peak stress values are about 4000-6000 psig (27-47 MPa).

The 10,000 hour creep rupture stress for 1300°F operation has been estimated for candidate heater head materials, and is summarized in Table 3-4. Note that 304 and 321 stainless steel are borderline given the maximum stress levels of the heater head. The 50,000 hour rupture strength has also been estimated for several heater head materials and is presented in Figure 3-5. Note that 304, 321 stainless steels are not appropriate materials for 50,000 hour creep rupture life for heater head temperature of 1300°F and pressures in the range of 1000-1500 psig.

The more exotic materials (N-155, Inconel 617), used in automotive applications have rupture strengths in excess of the requirement and may be promising from a creep rupture standpoint. A more detailed assessment is required however, to determine whether these materials are suitable. Safety factors of 1.5 are generally applied to account for the statistical variations in creep rupture gives limited data. For example, the mean predicted stress of 3,500 hour rupture level in air for N155 at 1400°F is 149 MPa. However, 90% confidence level indicates that the low is 77 MPa and high 257 MPa. Therefore, to ensure 90% confidence, the heater head would have to be designed for the lower value.

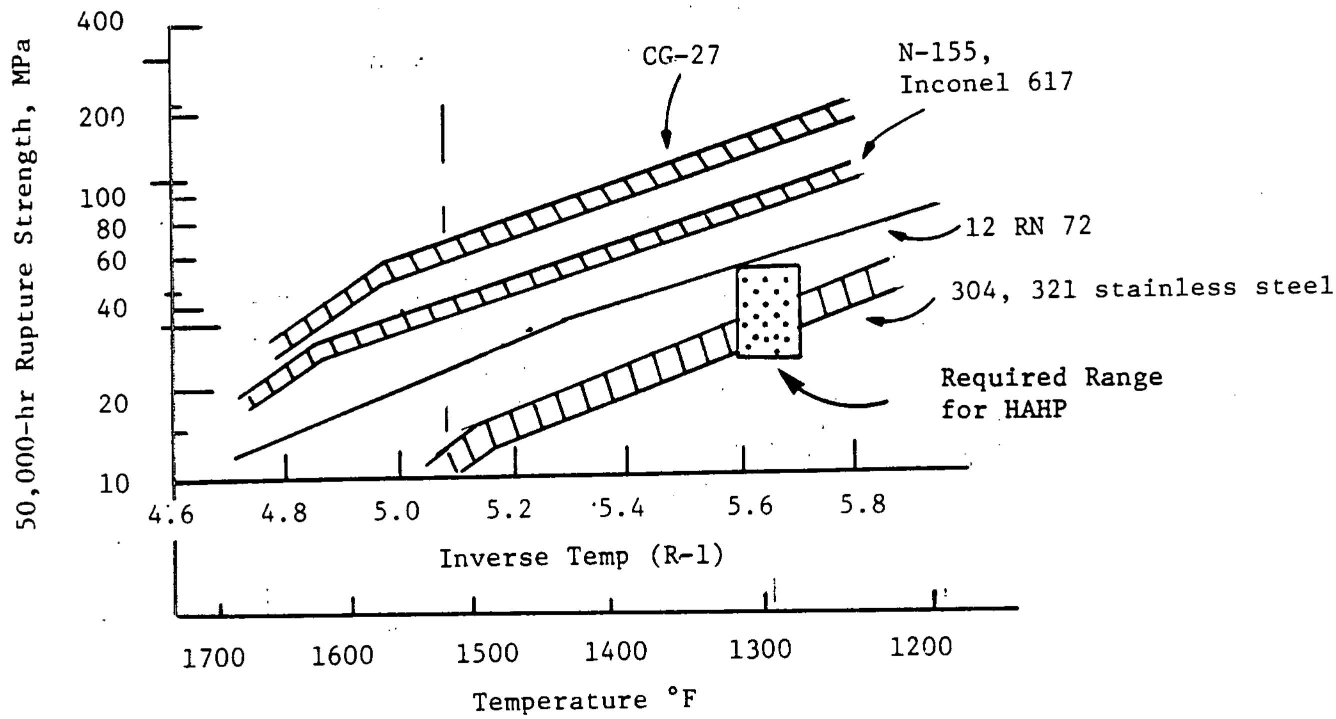

Figure 3-5. STRESS-RUPTURE STRENGTH OF CANDIDATE HEATER-HEAD TUBE ALLOYS

In summary, considerable materials work has been done on the automotive stirling progress related to creep rupture. Based on correlation developed on these programs, CG-27, Inconel 617, and N-155 multinet should be suitable with respect to creep rupture for 50,000 hour life for the gas fired Stirling heat pump. However, work must be completed for other material properties such as fatigue limits, ultimate tensile strength and yield strength before firm conclusions are made.

3.4 Ceramic Materials in Hot End Components

Ceramic materials are being considered for use in the hot end assembly for:

1) Burner liners - MTI turbulent burner;
2) Burner element - GE transpiration burner;
3) Air preheater - MTI and NASA automotive Stirling; and
4) Heater head - Garret, Sandian (Japan)

Ceramic materials and their max operating temperatures which are being considered, are listed in Table 3-5

Table 3-5

CANDIDATE CERAMIC MATERIALS FOR USE IN HOT END COMPONENTS

		Max Operating Temp (°F)
Alumina	(94 A_2O_3 - 2 Si O_2)	2500-3200
Mullite	(62 Al_2O_3 - 34 SiO_2	3000-3200
Alumina-Silica Fibers	(47 Al_2O_3 - 52 SiO_2	2300
Silicon Carbide	(SiC 80-90%)	2400
Cordierite	$MGO/Al_2O_3/S_iO_2$	2500

The following discussion of the use of ceramic materials is divided into subsections according to the application.

3.4.1 Burner Liners

The maximum preheat temperature is limited by the maximum temperature at which the burner temperature can be operated without structural failure. Since MTI operates at high flame temperatures (3500°F), a ceramic burner liner is required. Preliminary tests have indicated a failure mechanism due to the dissimilar growth rate between the Kawool liner and the stainless steel outer shell leading to separation, cracking, exposure and melting of outer shell. MTI has modified the design by installing the inner ceramic in compression loading. Although about 480 hours of successful operation has been demonstrated with this modification, if is doubtful that design lifetimes (25,000-50,000 hr) will be achieved with this design.

Since GE uses a radiant transpiration burner with low flame temperature (2950°F), blanket Kawool insulation is used as a liner. GE also utilized Kawool unsuccessfully at the combustion engine exhaust seal. Due to excessive wear characteristics, GE abandoned Kawool and reduced seal operating temperatures by installing an insulated flange between the combustor and the engine which permitted use of a conventional type seal consisting of a graphite impregnated PTFE seals.

In summary, ceramic materials have excellent temperature resistance for contact with hot gas, but their mechanical properties present problems. In particular, the difference in thermal expansion from metal make it difficult to line metal shells and vibration resistance is poor resulting in early failure.

3.4.2. Burner Element

General Electric HAHP prototype use a porous plate for the transpiration burner. Ceramic burners could substantially reduce the cost for the GE prototype. Preliminary estimates indicate manufacturing cost of 35¢ per burner versus $10 for metallic burners.

However, ceramic fiber burners (Cera blanket or Cerafoam) which were tested, had "Hot spots" on the outside surface due to nonuniformity of the material porosity. These hot spots melted after several hours of continuous operation creating multiple holes through the burners.

Flexible ceramic fiber materials are available with more uniform porosity than Cera blanket or Cerafoam. But their cost would be in the range of $6-8 per burner, which would substantially reduce their cost advantage. This would make ceramics less attractive considering their thermal stress problems.

3.4.3 Air Preheater

MTI is proposing a ceramic recuperator preheater for use in the Reference Engine System Design (RESD). The preheater is counterflow, corrugated plate heat exchanger made of a material similar to cordierite (mixed oxide ceramic made from MGO, Al_2O_3 and SiO_2). The ceramic preheater will: 1) substantially reduce the manufacturing cost as unit is assembled in preferred state and fired instead of making numerous welds; 2) ceramic material is more resistant to corrosion due to fuel bound sulphur; and 3) ceramic can be cleaned by baking off soot deposits.

The low cost feature is the primary benefit of ceramics for gas-fired heat pump applications since fouling and cleaning are not issues due to the negligible sulphur content of natural gas.

Based on ceramic preheater developments for industrial heat recovery, the use of ceramics appears to be encouraging. Hague has designed and developed a ceramic tubular heat exchanger made of silicon carbide for high-temperature application (1800°F - 2800°F). The company has installed this technology in the steel-forging industry and, since 1981, has extended its use to aluminum melting furnaces. Over 400,000 hours of commercial operating experience have demonstrated the high-temperature capabilities and resistance to corrosion of the heat exchanger.

GTE produces a ceramic cross-flow recuperator made of cordierite with an effectiveness of 65%. Over 800 of these units are currently in operation worldwide in heat treating, forging, melting, and other applications.

Babcock and Wilcox has designed a ceramic/metallic recuperator system and high-temperature burner that is being installed on a steel-soaking pit at their plant in Beaver Falls, Pennsylvania. A two-stage recuperator will preheat combustion air to 1850°F with flue gas temperatures of 2250°F.

AiResearch has developed a ceramic recuperator and high-temperature burner capable of operating with industrial flue gas temperatures as high as 2500°F. Installation of a commercial-size demonstration unit is being conducted on a steel-forging furnace owned by Cameron Iron Works of Dallas, Texas.

The DOE is currently sponsoring laboratory work in ceramic materials development and non-destructive evaluation techniques. In addition, work is being conducted at industrial facilities in the area of advanced concept heat exchangers.

GTE Sylvania, Midland Ross Corp., and Terra Tek have each demonstrated 1000-3000 hours of operation in high temperature flue gas with ceramic heat exchangers.

The most critical problem associated with the use of this technology for the gas-fired Stirling heat pump relates to the thermal cycling. The heatup rate of the ceramic heat exchangers has to be controlled to prevent thermal stresses from cracking the material. Silicon carbide offers the best resistance to thermal shock among all ceramics considered. If silicon carbide is used and the preheater is not allowed to cool down more than 300-500°F per off cycle, thermal stress due to cycling would probably not limit the life of the preheater.

NASA is giving considerable effort to developing ceramic heat exchangers. Their efforts in this area, described in recent automotive reports, are primarily motivated by a desire for reduced costs -- particularly in the air preheater. They displayed sections of a new ceramic air preheater module made by COORS.

This heat exchanger still suffers from severe thermal shock problems and usually fails in less than 50 cycles. In general, NASA does not display a high level of confidence in the ceramic heat exchangers based on available technology.

In summary, ceramic preheater developments for industrial heat recovery use have been encouraging. Thermal shock and differential expansion problems are minimized by controlling the heatup rate of the heat exchanger and utilizing silicon carbide which offers the best resistance to thermal shock. Considerable effort will be required to overcome these problems for the Stirling application as the mechanical problems are aggravated by the engine vibrator.

3.4.4 Heater Head

Limited work has been done on the use of ceramic materials for the heater head. Although the flexural strength of certain ceramics is high enough to consider as a candidate material (see Figure 3-6), work is required to define fatigue limit, and ultimate tensile strength for these materials.

Garret and Sandian (Japan) have claimed to develop ceramic heater heads and have offered to supply them to Sunpower for testing.

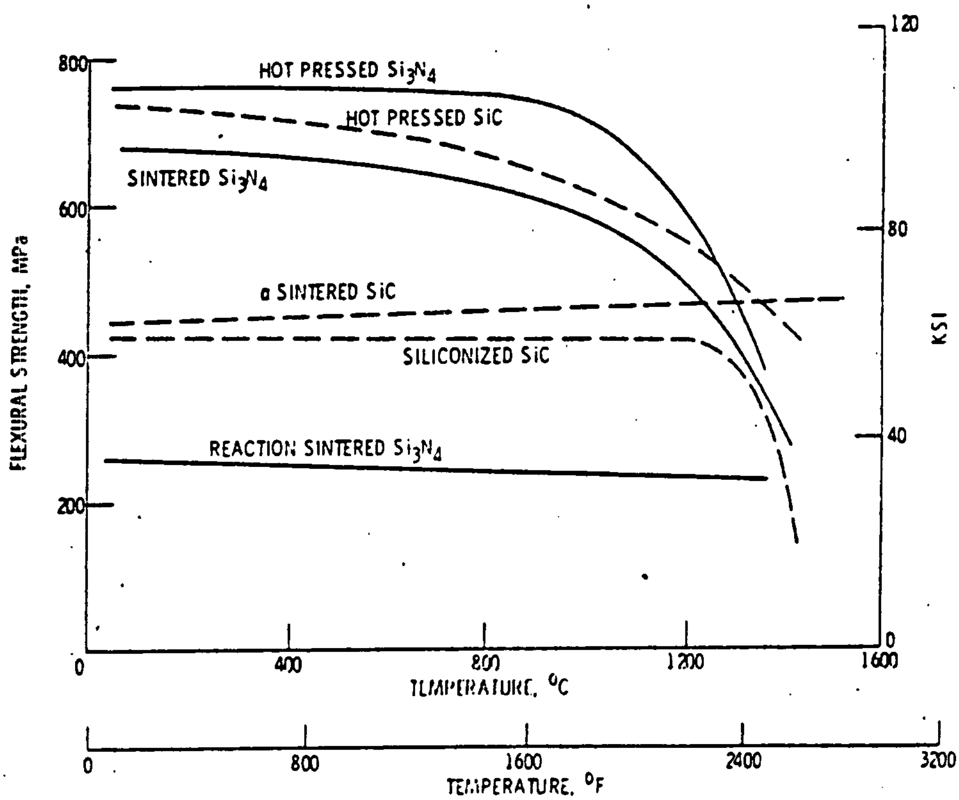

Figure 3-6. EFFECT OF TEST TEMPERATURE ON STRENGTH OF CANDIDATE CERAMICS PROCESSED BY DIFFERENT METHODS

4.0 CONCLUSIONS

Based on foregoing assessment of hot end components in the gas-fired Stirling heat pump, the following conclusions are summarized.

4.1 Efficiency

We have completed the first serious analysis of the impact of cyclic losses on combustor efficiency. Manufacturers have used steady-state or stack loss efficiency in all their performance calculations. On recommendation from DOE, MTI has incorporated a partload efficiency factor for the heat pump (.77 at 7% on time; .52 at 80% etc.), but have not applied a part load factor to the combustor.[2] DOE efficiency regulations for residential gas fired appliances requires that off-cycle losses be considered in the determination of seasonal efficiencies.[17] Results of our analysis on combustor efficiency indicate that seasonal efficiency can be 10-20% lower than steady state efficiency depending on load factor. Load factor depends on the weather, output rate of the heat pump, and heat transfer characteristics of the house.

4.2 Emissions

Meeting both CO and NO_x emissions for the Stirling application requires a delicate balance of excess air and flame temperature control. Neither MTI or GE have demonstrated operation at acceptable CO <u>and</u> NO_x emission levels. GE succeeded in achieving NO_x limits but very low flame temperatures caused quenching of CO resulting in CO levels above 400 ppm (ANSI limit) during integrated tests. The MTI prototype has low CO emissions but excessive NO_x (>1300 ppm). MTI will be incorporating EGR or CGR to reduce NO_x.

Based on work conducted in the automotive programs, emission control of CO and NO_x are technically feasible but will require further development to reduce the delicate balance of flame temperature and excess air under actual operating materials.

4.3 Materials

Stainless steels (304, 321) are not suitable for heater head materials for 25,000-50,000 hour life due to creep rupture limitations. Although Inconel 625, 617 and N-155 (multimet) exceed the creep rupture stress requirements for 50,000 hours life, their use is questionable given the safety factor of 1.5 which is suggested as a design guideline in the automotive programs. The safety factor of 1.5 is required to design within a 90% confidence interval of the 50,000 hour creep rupture stress.

Multimet (N155) and Inconels are expensive because of their chromium (15-20%) and cobalt (20% in N155) constituents. Based on a 50,000 hour rupture strength analysis completed in Section 3.3, only CG-27 would have the required creep rupture properties. 12RN72 and Sanuro 32 are on borderline for 50,000 hour operation and would not qualify if a safety factor of 1.5 were applied.

Ceramic air preheaters are not dictated by temperature requirements as the primary motivation is to reduce costs. Metallic air preheater can withstand the operating temperature, but would be difficult to manufacture. During the course of five years of ceramic preheater development for industrial waste heat recovery, considerable progress has been made resolving thermal shock and leakage problems. Their use requires careful control of the heatup rate to prevent thermal stresses from damaging the heat exchanger.

Work with ceramic materials in Stirling Hot End Components has been plagued with thermal stress/expansion problems. The use of ceramics in the Stirling is complicated by the vibration leading of the engine. It is too early to judge the technical feasibility of ceramics in the hot end components. The cycling operating environment needs to be more clearly defined. If the development of a ceramic air preheater is feasible, the substantial development is still required. In light of these problems, alternate approaches to manufacturing a low cost metallic heat exchanger should be pursued to determine if ceramics are the most promising route to achieving low cost.

4.4 Costs

Although a detailed cost analysis was outside the scope of this study, a few observations can be made. Cost estimates by GE and MTI of $110 to $150 are highly optimistic given the price of a special blower (hot exhaust for MTI and two-stage air flow rate control for GE), laborious assembly of air preheater, and use of expensive materials for heater head. Cost estimates assume the most optimistic developments and production rates inordinate with potential market share for the HAHP.

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