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A Technology Evaluation of the Stirling Engine for Stationary Power Generation in the 500 to 2000 Horsepower Range

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TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

TABLE OF CONTENTS (Continued)

TABLE OF CONTENTS (Continued)

LIST OF FIGURES AND TABLES

LIST OF FIGURES AND TABLES (Continued)

1.0 SUMMARY

1.1 Introduction

The current heightened concern over high energy costs and fuel availability has increased the interest in the Stirling engine as a potential prime mover for commercial, industrial and institutional power generation in the capacity range from approximately 500 to 2000 hp. Compared to its major competitors, the Diesel and gas turbine, the Stirling offers the potential for high efficiency, high reliability and long life, quiet operation, and most important, multi-fuel capability.

To date, the bulk of Stirling engine development effort has been devoted to automotive-type applications. By and large, these development efforts have been aimed at producing light-weight, high-specific power, and competitively priced engines for mobile power applications. In comparison, the stationary power generation market requires engines designed for high efficiency and long life, while accepting the inevitable trade-off towards greater bulk and higher cost. Therefore, before one can determine whether the Stirling engine may be a practical improvement over current prime movers for the aforementioned applications, the applicability of automotive Stirling engine technology to large stationary prime movers must be assessed.

This report documents the results of a study undertaken for the Division of Power Systems, for the purpose of assessing the potential and development status of the Stirling engine, and making recommendations for a possible program to develop 500-2000 hp stationary Stirling engines for commercial introduction by the late 1980's.

The principal tasks of the study were:

1. Assess the technical problems which must be resolved in order to develop commercially competitive and available stationary Stirling engines for multi-fuel operation.

2. Outline a program plan for conducting the development work necessary to advance Stirling engine technology to the point where multi-fuel stationary engines could be introduced commercially by the late 1980's.

The following subordinate tasks were also addressed during the course of the study and results are documented in Section 4.0 - Background and Section 5.0 - Technical Requirements - 500-2000 HP Stirling Engines:

1. Describe principles of operation of Stirling engine and history of developments.

2. Synopsis of recent and current automotive Stirling engine developments.

3. Outline differences in requirements and characteristics between automotive (mobile) and stationary Stirling engines.

4. Compare principal characteristics and advantages of Stirling engines for the stationary power generation applications with those of Diesel and gas turbine engines.

5. Discuss types of Stirling engines that should and should not be considered for the 500-2000 hp stationary engine application.

6. Outline typical design and operating characteristics for the recommended types of stationary Stirling engines.

7. Discuss the use of alternate fuels, particularly solid fuels including coal and coal-derived fuels.

8. Describe the expected characteristics of external heating systems for Stirling engines including both liquid and solid-fuel combustors.

9. Discuss the use of indirect heating systems to improve engine performance and increase flexibility in using solid fuels.

10. Identify the high cost components and where possible suggest approaches for reducing costs.

11. Identify the areas of high technological risk and recommend work needed to reduce these risks.

1.2 Methodology

Although an extensive literature review was conducted at the outset, much of the Stirling technology input to this study came from the background experience of the authors and from discussions with Stirling engine developers. Some of the authors (W. Percival, J. Smith) have been engaged in R&D work on Stirlings for several years. This knowledge was supplemented by fact-finding visits to the two principal Stirling engine developers; N.V. Philips, Eindhoven, Netherlands, and K.B. United Stirling AB, Malmo, Sweden. During these visits extensive discussions were conducted concerning the Stirling's potential for the intended application and the remaining technological risk areas where further work is required. Thus, it is hoped that the results and recommendations of this report will reflect the knowledge and concerns of those having the greatest exposure to Stirling engine technology.

1.3 Results and Conclusions

The principal conclusion from the study is that the Stirling engine indeed offers the potential to become a viable alternative to Diesel and gas turbine engines for stationary power generation in the 500-2000 hp range, primarily because it can be developed for operation on solid and other low grade fuels that cannot be utilized in Diesels and gas turbines. Although its multi-fuel capability is the principal attribute, the Stirling engine also offers significant advantages due to its low noise and emissions characteristics. Engine durability and maintenance characteristics are expected to be favorable even though there are some areas requiring further development before obtaining desired engine life. Stirling engine cost is expected to be somewhat greater than a corresponding Diesel, but it is believed that its ability to operate efficiently on low grade fuels will offset this cost penalty and create a viable market for the Stirling.

Detailed results of the various study tasks outlined above are presented in the corresponding sections of the main report and can be located easily by reference to the table of contents. The major conclusions reached during the course of the study are summarized below. These conclusions are segregated into five classifications as follows: General, Combustion System, Engine Design, Future Technology and Technological Risk Areas.

1.3.1 Conclusions - General

1. Stationary Stirling engines can operate with an efficiency essentially equal to that of Diesel engines of comparable size while producing lower levels of exhaust emissions and noise.

2. Stirling engines can be fired with solid fuels or other low grade fuels, such as residual oil, crop-residues and waste materials with development of suitable combustion systems and heat transport systems.

3. Stirling engines can readily be fired with low Btu gas because the high pre-heat temperature makes it possible to achieve stable combustion with reasonable furnace efficency.

4. In production, stationary Stirling engines burning distillate fuels are likely to be 25-50% more expensive than Diesel engines of similar size.

5. Stirling engines operating on solid fuels (coal) are likely to be 30-50% more expensive than Stirling engines running on distillate fuels.

1.3.2 Conclusions - Combustion System

6. When burning distillate fuels, Stirling engines emit lower levels of exhaust emissions than either Diesel engines or gas turbines.

7. Stirling engines can readily burn coal or other solid fuels - the most likely methods are believed to be stoker and fluidized bed combustion systems, although significant development effort is required.

8. Stoker-fired systems would probably need to use preprocessed coal to limit SO_x emissions, because SO_x stack gas cleanup equipment would be too expensive for single engine installations.

9. When burning solid fuels in a Stirling engine it is most likely that indirect heating means will be employed, i.e., an intermediate heat transport system to transfer heat of combustion to the Stirling engine heater head.

10. Effective indirect heating can be achieved by either a liquid metal heat-pipe system or by a pressurized helium or hydrogen convective heat transport loop.

1.3.3 Conclusions - Engine Design

11. The multi-cylinder double-acting engine configuration is believed to be the most suitable type for large stationary engines from an overall cost-effectiveness point of view.

12. 125 hp per cylinder is an ideal cylinder size for a line of engines to cover the 500-2000 hp range because a basic 4-cylinder engine module would provide 500 hp, and 1000, 1500 and 2000 hp units could be obtained by incorporating 2, 3 and 4 basic engine modules to obtain 8, 12 and 16 cylinder engines. Commonality of parts would help keep engine production costs down.

13. There is little risk in scaling up current automotive size double-acting engine designs to the 125 hp cylinder size because:

    a. Displacer-type Stirling engines have been designed as large as 400 hp per cylinder and built/tested as large as 90 hp per cylinder.

    b. Double-acting engines have been built and tested as large as 40 hp per cylinder - but the scale-up to 125 hp involves little risk since scaling rules developed for displacer engines apply equally well to the double-acting engine.

14. Stirling engines for stationary power are most likely to use helium as the working gas, particulary if indirect heating is employed.

15. In order to design an optimum Stirling engine, it is essential to have access to <u>validated</u> analytical design methods (e.g., computer routines). To the best of our knowledge, the only such analytical routines are available only through Philips and its licensees.

16. Near-term Stirling engines (late 1980's) will use metallic heater heads operating at temperatures ranging from 750-800°C.

1.3.4 Conclusions - Future Technology

17. Future developments in ceramics may offer the possibility of lower cost ceramic heater heads operating at higher temperatures (up to 1100°C) for improved efficiency and power.

18. While ceramic heater tubes may be attractive for use in fluidized bed combustion systems from a cost and corrosion resistance view-point, their heater temperatures will be restricted to below 1800°F to retain effectiveness of sulfur removal and/or to prevent ash fusion.

1.3.5 Conclusions - Technological Risk

The principal areas of technological risk are:

1. Solid Fuel Combustion Systems.

2. Indirect Heat Transport Systems.

3. Heater Head - Cost and Life.

4. Piston Rod Sealing.

5. Piston Seals.

6. Air Pre-Heater - Cost and Life.

While it is not an area of technological risk, regenerator cost reduction may also be required to make the large Stirling engine economically viable. These risk areas are summarized briefly in Section 2.0 (Technological Risk Areas) and discussed in more detail throughout Section 5.0 (Technical Requirements).

1.4 Recommendations

It is recommended that a program be undertaken by DOE to develop the technology for a large stationary Stirling engine capable of operation on solid fuels for possible commercial introduction by the late 1980's. This program should have two principal branches:

1. Large Stirling Engine Development - results would be applicable for engines operating on liquid fuels, as well as solid fuels and other heat sources.

2. Combustion System Development - should provide acceptable methods for utilizing low grade fuels, such as coal and coal derivatives, cultivated fuels, agricultural and forest waste, etc.

Further details of the recommended development program are contained in Section 3.0 (Program Plan).

2.0 TECHNOLOGICAL RISK AREAS

There are several areas where there still exists significant technological risks in developing commercially viable large stationary Stirling engines to operate on low grade fuels. Further development work is needed in these risk areas. These areas are listed below in approximate order of risk level. Each area is then discussed briefly in the paragraphs which follow. The risk areas are:

1. Solid Fuel Combustion Systems
2. Indirect-Heat Transport Systems
3. Heater Head - Cost and Life
4. Piston Rod Sealing
5. Piston Seals
6. Air Pre-Heater - Cost and Life

2.1 Solid Fuel Combustion Systems

Since no existing solid fuel combustion system is immediately applicable to the Stirling engine, this must be a prime area of attention if a practical coal-burning Stirling engine is to be developed. The two leading candidates from existing or emerging technology are stoker and fluidized bed combustion systems. The former system would be applicable to a direct-fired engine only if it could be equipped with a hot-side particulate collection system. Both systems are applicable if an indirect heat transport system is employed.

2.1.1 Stoker Systems

The technological risk areas for stoker-fired systems include particulates, gaseous emissions, air pre-heating, and control.

Particulates

If combustion products from a stoker-fired furnace are to be used directly to heat the engine heater head, it is desirable to have a furnace exit temperature as high as possible in order to achieve high heat transfer rates in the heater. In order to prevent slagging or sintering of fly ash on the heater tubes, it is necessary that either the gas temperature be below the fusion temperature of the ash or that essentially all of the ash be removed by a hot-side dust collection system. While there are coals which have ash fusion temperatures of 2800°F (1500°C), the fusion temperature of most coals is several hundred degrees below this temperature. Even at a temperature of 1500°C, the high fouling factor of the dust-laden gases would require significantly more heat transfer area in the heater head than in an oil or gas-fired unit, resulting in reduced specific power and efficiency. Thus, it is likely that a stoker furnace could be used for direct-firing only if a very effective hot-side dust collection system can be developed. The low percentage of entrained particulates from stoker burners may make it practical to use cyclone separators for this purpose.

Emissions of SO_x and NO_x

Unless preprocessed coal is used, the SO_x emissions from a stoker-fired engine would be as high as in conventional units. The high excess air and long residence time associated with stoker burners are conducive to high NO_x emissions also. These gaseous emissions would require some sort of post-combustion cleanup, which may be inordinately expensive for systems of small size. It is felt that little of the emissions reduction technology that is being developed for large utility and industrial boilers would be applicable to a system of this size, but that a new technology, possibly relating to automotive emissions reduction technology, would have to be developed.

Air Pre-Heat

The high heat input temperature of the Stirling engine results in high exhaust gas temperatures leaving the heater head. In liquid-fueled automotive versions, these high exit temperatures are effectively recuperated in the air pre-heater. However, a stoker burner cannot tolerate high air pre-heat because of its tendency to cake the coal bed. While it may be technically feasible to utilize the high temperature exhaust gases in some sort of a bottoming cycle, this would compromise the efficiency of the Stirling cycle, and in all likelihood would be economically unattractive. It is believed that the most practical solution is to use the waste heat to generate process or heating steam, which would be appropriate for industrial, commercial, or residential applications, but probably not for dispersed power applications. In fact, this pre-heat limitation might be viewed as an advantage, where the Stirling engine is to be employed in a topping cycle to improve the energy utilization efficiency of coal-fired process and heating boilers.

Operation and Control

Generally, stoker-fired boilers require more operating labor, but of less skill, than do pulverized or cyclone-fired boilers. The control of the combustion process is relatively simple and forgiving, and is thus, well suited to semi-skilled operation. However, if a coal-fired Stirling engine is to be a viable competitor with other alternative power sources, it must be capable of automatic operation with minimal operator requirements. Also, the simple and slow control afforded by manually-operated stoker-fired burners may be entirely unsuitable in applications where the Stirling engine is providing all of the power to a process, and thus, must be capable of rapid and precise load modulation. Thus, it is believed that a stoker-fired Stirling engine must be fully automated in all of its functions, including load modulation, coal feeding, combustion control and solid waste removal. While there are modern stoker systems which achieve the desired level of automation, these are employed in large utility and industrial boilers, and cannot be expected to apply to the size of system under consideration. Therefore, it is believed that considerable effort would have to be devoted to developing an economical and practical automatic control and materials handling system for a stoker-fired Stirling engine.

2.1.2 Fluidized Bed Combustion Systems

The areas of technological risk associated with fluidized bed combustion systems (FBC) for Stirling engines are particulate emissions, solid waste disposal, and indirect heating systems. Also costs will be a critical problem area.

Particulate Emissions

Stack particulate emissions are a problem with most types of fluidized bed combustion systems. Some initial test results indicate that a fairly large fraction of the fly ash is of a small enough size that it is ineffectively removed by mechanical dust collectors (Ref. 48, p 6-43). Furthermore, fluidized bed burners produce almost no SO_3, which is believed to be necessary for effective operation of electrostatic precipitators (Ref. 48, p 6-48). This problem is receiving considerable attention in the R&D programs presently underway, and if it is resolved, the solution would probably apply to this application as well.

Solid Waste Disposal

The solid waste produced by a fluidized bed burner is approximately three times greater than in other types of coal burners due to the addition of bed material. This presents problems both with regard to loss of combustion efficiency, and with regard to the problem of waste disposal. A coal-fired Stirling engine would produce approximately 250 lb/hr of waste per MW of power. Systems must be developed for the efficient (and automatic) removal of this solid waste; plus the larger problem of ultimate disposal or recycling must be considered.

Indirect Heating

In spite of the high heat transfer coefficients available in the fluidized bed combustor, the low bed temperature results in too low a heat transfer rate (heat flux) to consider direct-firing of the Stirling engine. Thus, either a sodium heat-pipe or a single-phase gas heat transport system must be employed. Although a sodium heat-pipe might be considered simpler from an operational standpoint, problems of its application to industrial use and its potential serviceability must not be overlooked. Due to the limited FBC heat transfer rate, both approaches would reqire the development of low cost, high temperature heat exchangers, and could benefit from advances in ceramic heat exchanger technology.

Regardless of which type of coal combustion system is utilized, it must be recognized that the cost and complexity of a coal-fired Stirling engine will be significantly higher than an equivalent gas or oil-fired engine. Whether or not the additional cost and complexity is accepted by the marketplace will depend to a great extent on the relative availability and cost of the competing fuels. While large industrial or utility users may accept or be forced to utilize the more capital-intensive fuel, it is more likely that the user of the Stirling engine in the size range under consideration will be less ready to accept a coal-burning engine unless the additional cost and manpower requirements are reduced to a minimum.

2.2 Indirect Heat Transport Systems

While it is recognized that the use of a heat-pipe in an indirectly-fired engine provides substantial performance benefits and may, in fact, be necessary in a coal-fired engine, its application introduces several areas of technological risk. The principal areas of risk that will require attention are as follows:

1. Material and Fabrication

Long service life of a high temperature liquid metal heat-pipe requires excellent long-term compatibility between the liquid metal and the materials of construction. While present evidence indicates no compatibility problems between sodium and the high temperature alloys (especially stainless steels), there is insufficient experience over long periods of time at a high temperature (greater than 20,000 hours at 800°C) to demonstrate that no long-term problems exist. Furthermore, these materials are expensive, and considerably larger amounts will be used in a heat-pipe than in a direct-fired heater head. Because of the extreme importance of maintaining high purity, stringent precautions will have to be taken during fabrication to assure cleanliness and adequate sealing of the heat-pipe. Thus, the heat-pipe is likely to be expensive both in terms of the materials used and the manufacturing process.

2. Serviceability

Two approaches to serviceability are a hermetically sealed heat-pipe which would require no service, or a mechanically sealed heat-pipe which would be accessible for service. The former approach entails the problem of assuring long service life in a non-serviceable component, plus the potential problem that a hermetic heat-pipe may interfere with serviceability of the remainder of the engine. The alternate approach introduces the problems of maintaining system purity and safety, while exposing a system which contains an extremely reactive medium such as sodium. One must question whether this would be practical in an industrial environment. The authors' believe that both approaches would have to be investigated in detail before a preferred approach could be selected.

3. Heat Flux Control

It is anticipated that temperature control for a heat-pipe furnace design, burning gas or liquid fuels, will resemble the control system now applied to a direct-fired engine, which is described in Section 5.2.7. The thermocouple for feedback control is preferably located on a heater tube, rather than on the heat-pipe. Control of solid fuel combustion is expected to be considerably more complex, with slower response times than now realized with conventional fuels. However, it is expected to operate in the same basic mode, with the temperature signal controlling air flow and the fuel flow slaved to air flow to maintain a nearly constant air-fuel ratio.

Industrial systems for sizing, storing, feeding and distributing solid fuels in a furnace are generally much larger than required for a single Stirling engine. Considerable study and development will be required before small systems can be applied successfully. Solid waste and particulate removal systems must also be designed.

4. Single-Phase Heat Transport Systems

In principal, a forced circulation, single-phase heat transport system can furnish many of the same advantages obtainable through an orthodox heat-pipe, while avoiding the problems associated with the high reactivity and the high purity requirements of liquid metals. However, the high working pressures needed to limit the pumping power, introduce additional problems associated with the safety of the high pressure gas, the additional cost of the high temperature material needed to contain the gas, and the problem of developing a reliable high-temperature/ high pressure circulator. Thus far, these risks have been judged greater than those associated with liquid metal heat-pipes by developers in the field, however, this choice should be re-examined with respect to the industrial application.

2.3 Heater Head

Design and fabrication techniques for metallic heater heads are currently well in hand for engines burning liquid fuels. High heater head costs continues to be a problem and there is certainly a need for further work on cost reduction methods. The main objectives are obtaining lower cost materials which can be fabricated by welding, rather than the present slow batch brazing methods. Minimizing the content of strategic elements, such as chromium and cobalt is important. The problems and risks nearly duplicate those in the gas turbine industry. Some of the iron-base metals developed for automotive gas turbines may be applicable to the Stirling engine.

Heater heads for use with solid fuel combustion have not been widely explored, so this remains an area of significant technical risk. New designs must be developed and optimized for use with solid fuels, both for direct-fired (if applicable) and indirect-fired systems. Heaters for possible direct-firing will need to allow for particulate fouling, and probably incorporate means for periodic cleaning by use of in-situ ash blowing devices. Heaters for indirect-firing must be designed to take full advantage of the "bonus effect" described in Section 5.3.2.1; i.e., they must be optimized for minimum void volume and allow operation at the highest possible tube wall temperatures. For hydrogen engines, the problem of hydrogen diffusion through heater tubes must be carefully considered so that it does not have an adverse effect on the heat transport system (e.g., poisoning of heat-pipe with non-condensible gases).

Heater tube materials should also be re-examined for use with solid fuel combustion systems. For direct-fired systems, the effect of flue gas constituents on tube corrosion and erosion must be considered, along with material costs and fabrication techniques. For indirect-fired systems, compatability with the heat transport medium is a new factor to be considered in choosing heater tube materials.

When hydrogen is used as the working fluid, the problem of hydrogen loss by diffusion through the hot metal components must be considered. Philips claims to have solved this problem by use of a special coating inside the heater tubes and other hot parts. Further work in this area is required. As noted in Ref. 33, diffusion appears to decrease with time, with present heater designs. While this is encouraging, further research is needed to determine the causes and what can be done to speed up the process, as for example, by a pre-treatment of the hot metal components.

2.4 Piston Rod Sealing

Although much progress has been made on improvement of Stirling engine rod seals, they have not yet demonstrated the 10-15,000 hour life required for stationary engines. Most of the recent progress has been made on sliding seals which to date, have achieved lifetimes of about

4,000 hours in double-acting engines. Further progress is needed. While is it important to prevent working gas from leaking out of the system, the most difficult part of the rod seal problem is preventing the lubricating oil from migrating up to the piston ring area. Further work is needed to develop fully satisfactory rod seals for both gas retention and oil control. New designs should be explored, as well as possibly new materials and fabrication methods to improve surface geometry. Although they have abandoned the roll sock seal for use on small engines, Philips believes that it might prove suitable for large stationary engines due to the lower seal operating temperatures and the larger (thicker) diaphragms involved. It is recommended that this type seal should at least be explored for use on the large stationary engines.

2.5 Piston Sealing

Piston ring seals for stationary Stirling engines are not developed to a fully satisfactory state. Ring life is still insufficient and needs to be improved from the present level of about 4,000 hours to a level of 10-15,000 hours. In addition, control of the "pumping" characteristics of polymer rings needs further study and development to ensure maintaining an even distribution of working gas among cylinders of a double-acting engine.

2.6 Air Pre-Heater

The major technological problems with the air pre-heater are service life and cost. These problems are common to both metal and ceramic pre-heaters and are common to both recuperators and to rotary regenerators. In contrast the heat transfer and hydrodynamic design of pre-heaters is well developed with adequate data available for standard geometric arrangements. However, additional design data will have to be taken and correlated for any new geometric arrangements of flow channels that are developed in the interest of lowering cost and/or improving life.

The major service life problems are from thermal fatigue and chemical attack and oxidation. The thermal fatigue problem results from local yielding due to thermal stress cycling. The thermal stress cycling occurs at start-up and shut-down, load changes and from regenerator thermal cycling in the case of rotary regenerator. Improved designs should be explored to solve the thermal stress problems.

The corrosion problem is different on the hot end than on the cold end of the pre-heater. On the hot end, various fuel impurities are able to diffuse through the oxide layer that normally protects the base metal. This problem is common to all structural metals operating at high temperatures in atmospheres with corrosive components in combination with oxygen. The cold end corrosion is associated with condensation of water and sulfur oxides on the pre-heater surface. This problem is controlled by maintaining stack temperature sufficiently high to avoid condensation in the air pre-heater. However, this is difficult or impossible on start up, during low load operation and when operating with very low ambient temperatures. Condensation on the pre-heater surfaces also aggravates the problem of particulate fouling, since it sticks the particles to the heat transfer surfaces. The selection of materials and designs to provide adequate corrosion protection requires further investigation.

3.0 PROGRAM PLAN

It is recommended that a program be undertaken to develop large Stirling engines for stationary power applications based on the utilization of non-petroleum fuels that are unsuited for use in Diesel engines. This program should have two principal objectives as illustrated in Figure 3-1; (1) large stationary engine development; and (2) solid fuel combustion system development. These two principal activities should proceed in parallel, and after successful hardware demonstrations of each objective, the programs should be combined to demonstrate the first generation solid-fuel stationary engine. Following that, a second generation engine optimized for the chosen combustion system should be developed and demonstrated.

Concurrent with this work it is recommended that this Stirling engine program should support and stay abreast of high temperature materials research work (e.g., ceramics) that could be of long-term benefit to Stirling engine advancement (post 1990). The following paragraphs provide more details of the proposed program.

3.1 Large Stationary Engine Development

There is currently no development work underway on large heavy-duty Stirling engines suitable for stationary power applications in the 500-2000 hp range. Present developments are concentrated on the automotive, truck and mining vehicle markets for engines up to 200 hp. It is not likely that any private companies will begin development of the large engines of the type considered in this study, because today there is inadequate incentive in the private sector to purchase such engines solely for their ability to burn solid fuels, and the future course of the fuel market is

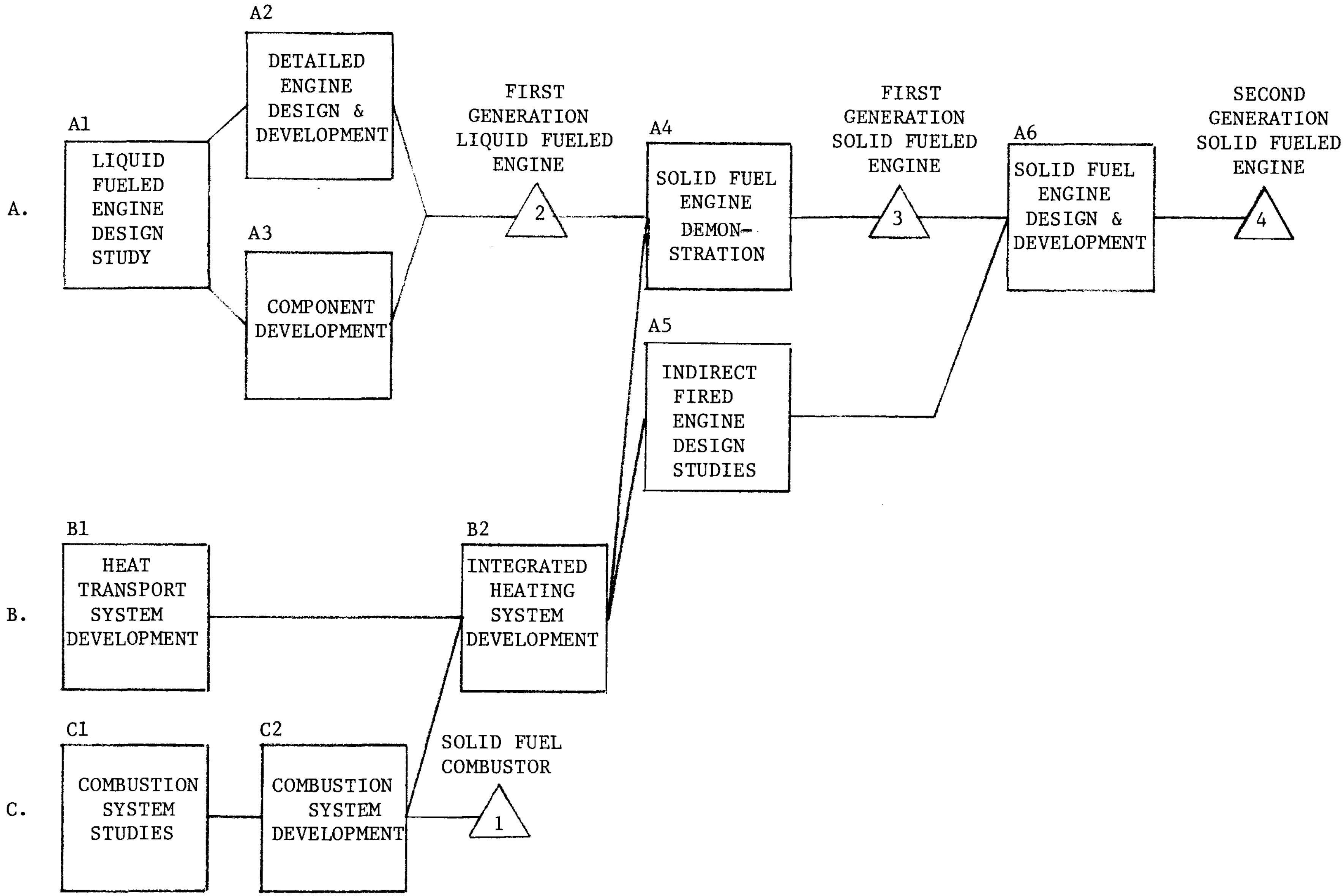
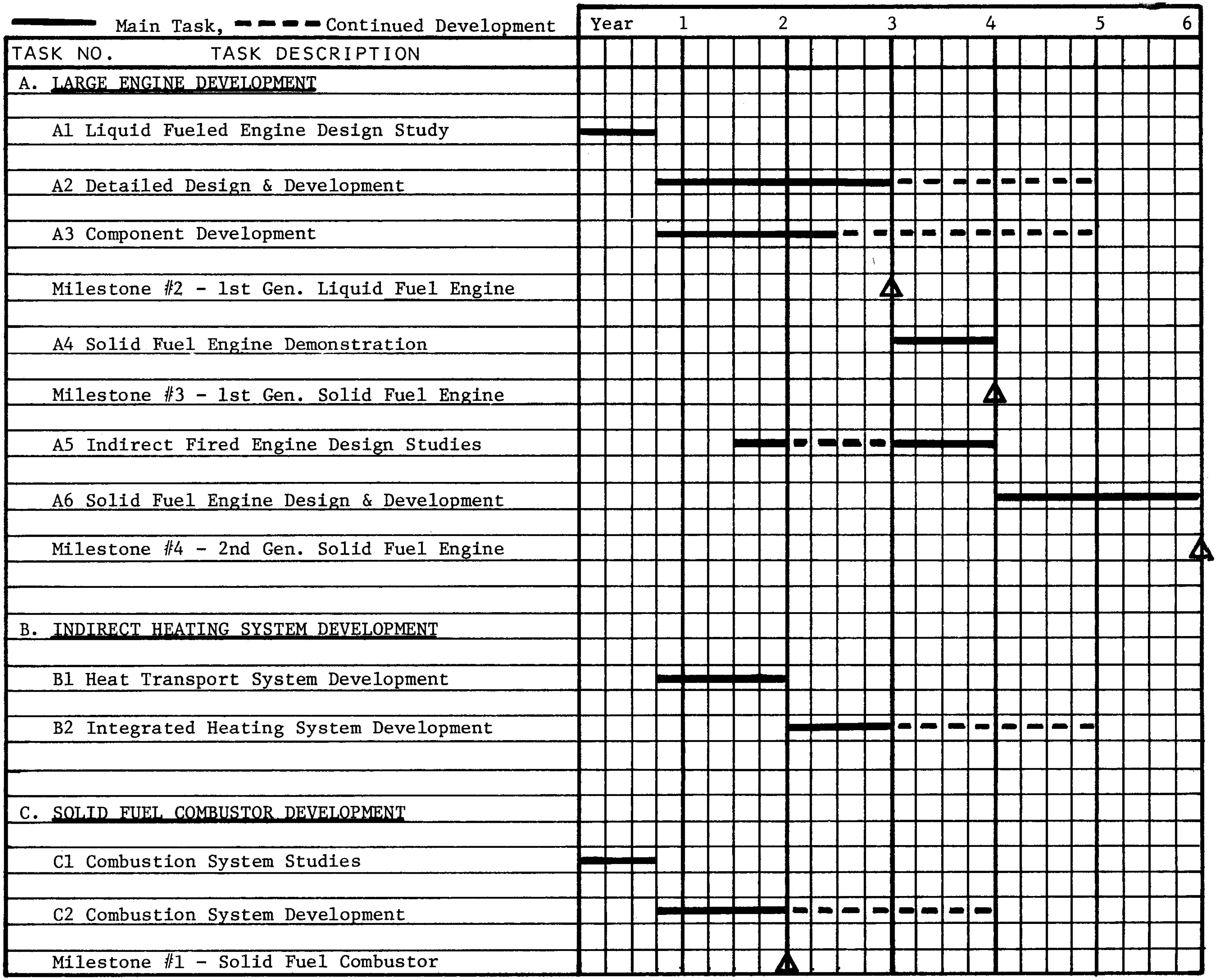


Figure 3-1 (a) Recommended Large Engine Development Plan

Figure 3-1 (b) Recommended Program Task Schedule



uncertain. Therefore, if such developments are to occur early enough to make the solid-fuel engine technology available when needed, government support will be required in the early phases.

Many of the technical risk areas for stationary Stirling engines are not necessarily related to the problem of burning solid-fuels, but rather are common to both liquid and solid-fueled engines. Therefore, it is recommended that the large engine design and development work start with the objective of developing an engine optimized for the use of liquid-fuels using current liquid-fuel combustion technology. This activity is illustrated in branch A of the program chart (Figure 3-1(a)). Proceeding along this course will enable the long lead engine program to progress a significant amount before the solid-fuel combustion technology is demonstrated.

This branch should begin with an engine design study to develop design concepts for a line of engines to cover the range of 500-2000 bhp. This would include a computer optimization for the thermodynamic design of the basic engine unit, as well as detailed design layouts of the engine mechanical arrangement. Preliminary, critical and final design reviews should be held during this phase with appropriate expert consultants in attendance to help assure the best possible design for the first prototype engine. As indicated above, the first engine should be designed for direct heating with combustion of liquid-fuels and the test program should be conducted with both petroleum distillate fuels and coal-derived liquid fuels.

Early in the engine design program, a study should be conducted of the relative advantages of hydrogen versus helium as the working gas for large engines. This study should include consideration of such factors as engine size, cost, and efficiency, as well as safety, regulatory, insurance, operator training and other institutional-type problems that may be associated with the use of hydrogen. A final selection on working gas for the first prototype engine would be made after the study conclusions are drawn.

A separate engine component development program should proceed in parallel with the main engine development. This program would concentrate on only those components that have been identified as high technical risk areas. New ideas for innovative solutions should be sought and programs should be funded independently of the main engine program. These component programs should be keyed to the main engine program and component developers should be required to work in close concert with the main engine developer to assure the applicability of the component work to the prototype engine. Component problems that should be considered for such work are:

1. Heater Head Cost and Durability - Metallic
   - Materials, designs
   - Fabrication techniques
   - Hydrogen diffusion
   - Thermal cycling
   - Corrosion
2. Piston Rod Seals
   - Sliding seals
     - Power consumption
     - Gas leakage
     - Seal life
     - Oil retention
   - Rolling seals
     - Material quality control
     - Life

3. Piston Seals

   - Life
   - Control of pumping behavior
   - Alternate materials/designs

4. Air Pre-Heater

   - Cost - materials/designs
   - Durability
     - Thermal cycling
     - Corrosion

5. Regenerator

   - Low cost designs that retain high performance

6. Development of Accurate Analytical Design Routines

After these tasks are well underway and some initial positive results are available from the solid-fuel combustion system development, design work should be initiated on a large stationary engine optimized for solid fuel combustion.

3.2 Solid Fuel Combustion System Development

Since a major advantage of the Stirling engine is its ability to burn a wide range of fuels, including solid-fuels, it is important to exploit this feature to reduce our dependence on petroleum-based fuels. Therefore, it is recommended that a major segment of the stationary engine program be devoted to development of combustion systems to utilize coal and coal-derived fuels, as well as cultivated renewable fuels and agricultural and forest residues.

The combustion branch of the program should start with a study of the various methods for direct combustion of coal, as well as the possibilities for burning coal derived fuels including coal-liquids and low-Btu gas. Ideas should be solicited for innovative approaches to coal utilization in Stirling engines. The evaluation of various attractive approaches should start with studies comparing the characteristics, problems, and potential advantages for various methods and, where necessary, might also include some simple "proof-of-concept" experiments.

Following this comparative evaluation, the more promising methods should be selected for hardware development leading to coal-based combustion systems to be integrated with the large engine. It is presently believed that this activity should explore at least the following methods for coal utilization:

1. Direct Combustion of Coal and Other Solid Fuels
   a. Atmospheric Fluidized Bed Combustion
   b. Pressurized Fluidized Bed Combustion
   c. Stoker-Fired Coal Combustion (both refined and unrefined coal should be considered for these systems)
2. Combustion of Coal-Derived Liquid-Fuels
   a. Consider various liquid coal products and the advantages/disadvantages of their use. Focus on lower cost coal liquids that cannot be used in Diesel engines due to cetane ratings.
3. Low Btu Gas From Coal
   a. On-site coal gasification for low Btu gas should be explored since a Stirling engine can easily operate on this fuel.

All candidate combustion systems must provide acceptable characteristics in the following areas:

1. Combustion efficiency
2. Emissions
3. Waste disposal
4. Heat transport system life/fouling
5. Pre-Heater life/fouling
6. Automatic operation
7. Load following
8. Safety
9. Equipment cost
10. Maintenance requirements
11. Operating labor requirements
12. Reliability
13. Auxiliary power requirement

3.3 Indirect Heat Transport System

Due to limitations in heat transfer rates and problems with fouling and corrosion, it is most likely that coal combustion systems will employ indirect heating and require some form of heat transport system interposed between the combustor and Stirling engine heater head. Since the critical design features of the heat transport system depend more on the engine heater head design than on the combustion system design, the development of this subsystem can begin early and proceed in parallel with the large engine development and coal combustion system development. The two schemes most likely to be suitable for indirect heat transfer are the liquid metal heat-pipe and the high pressure helium gas flow loop. Both systems should be studied for possible application and then one or both developed for use on the large engine program. Some of the important factors to be considered in the evaluation and comparison of candidate heat transport systems are:

1. Performance
   - Heat flux into heater head
   - Excess temperature required at combustor
   - Parasitic power requirements
2. Cost
3. Safety
4. Reliability
5. Maintainability

It is recommended that the work on indirect heat transport systems proceed in parallel with combustion system development, and after successful testing on both the coal combustion and heat transport systems, they should be combined to demonstrate the integrated system before installing it on the large engine.

3.4 Coal Burning Engine Program

An engine optimally designed to utilize solid-fuels is likely to be somewhat different from the optimized liquid-fueled engine, principally because the liquid-fueled engine will most likely be a direct-fired hydrogen engine, whereas the solid-fueled engine will probably be an indirect-heated helium engine. The heater head designs will be quite different, the indirect engine having lower void volume; as a result it may be desirable to change other engine design parameters for an optimized solid-fueled engine.

As mentioned earlier, a design study for an optimized solid-fueled engine should be initiated at the time the specific approaches are selected for the combustion system and heat transport system (see Figure 3-1). This engine design should be continually updated throughout the test and development period for the liquid-fueled engine, so that improvements derived from that experience can be incorporated into the second-generation engine design. It would even be desirable to delay the final "design freeze" until after some experience is obtained with the first-generation (liquid-fueled) engine operating on the solid-fuel combustion systems (Task A4, Figure 3-1). After the design freeze, the fabrication, test and development phases should proceed as rapidly as possible to provide the earliest possible decision point on the technical and commercial viability of this engine.

3.5 High Temperature Materials Research

Ceramics appear to offer an attractive potential for both reduced cost and improved efficiency. However, it is the authors' opinion that the perfection of ceramics for heater heads at the higher temperatures being suggested will require extensive and costly research and development. A great deal of funding has already been made available for gas turbine ceramic work, and all of this technology being developed will be available for use in future Stirling engines. Two of the candidate materials, silicon carbide (SiC), and silicon nitride (Si_3N_4) have very good high temperature properties; however, it would not be productive to concentrate on their use in large Stirling engines as part of this program. These materials are being evaluated in less-demanding applications, and even in these situations, there appears to be much research and development required before practical use of hot ceramic components. Moreover, it is the authors' opinion that the feasibility of industrial Stirling engines can be more forthrightly demonstrated using less exotic, current technology, high-temperature wrought alloys. This is not to say that ceramics are unfeasible, but that their introduction is more properly a part of an advanced engine development program.

The stationary Stirling engine program should have a task segment to monitor high temperature materials developments currently underway, and should engage in Stirling engine component test and development only after promising results are achieved in the gas turbine ceramics program.

Funding Requirements

Estimated funding required for each task of the program is shown on Table 3-1. In making these estimates it has been assumed that intensive development of automotive size engines will continue and the stationary engine program will be able to derive substantial benefit from the automotive program. Total program costs over a period of 6 years are estimated at \$12-22×10^6.

TABLE 3-1

ESTIMATED STIRLING ENGINE DEVELOPMENT COSTS

	($\$10^6$)
A. LARGE ENGINE DEVELOPMENT	
A1 - Liquid Fueled Engine Design Study	0.3 - 0.5
A2 - Detailed Design and Development	3.0 - 5.0
A3 - Component Development	2.0 - 3.0
A4 - Solid Fuel Engine Demonstration	1.0 - 2.0
A5 - Indirect-Fired Engine Design Studies	0.3 - 0.5
A6 - Solid Fuel Engine Design and Development	2.0 - 5.0
B. INDIRECT HEATING SYSTEM DEVELOPMENT	
B1 - Heat Transport System Development	0.5 - 1.0
B2 - Integrated Heating System Development	0.4- 0.8
C. SOLID FUEL COMBUSTOR DEVELOPMENT	
C1 - Combustion System Studies	0.4 - 0.6
C2 - Combustion System Development	2.0 - 4.0
TOTAL	11.9 - 22.4

4.0 BACKGROUND ON THE STIRLING ENGINE

The modern Stirling engine is a heat engine which operates on a closed regenerative thermodynamic cycle and is similar to engines built in 1815. The working fluid undergoes a cyclic process including compression at low mean temperatures and expansion at high mean temperatures. The temperatures of the gas are changed during the cycle by a positive displacement of the gas from a hot space through a regenerative heat exchanger to a cold space or vice-versa. Heat is added to the gas at high mean pressure and temperature. and rejected at low mean pressure and temperature such that there is a net conversion of heat to work.

The Stirling is different from the familiar gasoline or Diesel internal combustion engine in two respects; first, it uses the same working fluid over and over again; and second, the heating and cooling takes place by heat transfer through metal walls. In other words, the Stirling operates on a closed cycle and combustion takes place externally, as in the Rankine cycle powerplant. The Stirling differs from the Rankine in that the working fluid does not change phase; it is always a gas, usually hydrogen or helium.

All Stirling engines which have actually operated are reciprocating piston devices which operate without valve gear. Two pistons per cylinder are required for single cylinder engines; while engines of 3 to 8 cylinders may employ one double-acting piston per cylinder.

The Stirling engine can operate from any source of sufficient heat at the proper temperature. Normally, heat from the combustion of a distillate fuel is utilized; but other liquid, gaseous or solid fuels, as well as solar heat, stored heat, isotopic heat or a nuclear reactor heat source might be used. For most applications, the engine can be considered to have two distinct fluid circuits. The internal circuit is the engine thermodynamic circuit, filled with working gas at an elevated pressure. The components are two variable volume cylinders and three heat exchangers, called the heater, cooler and regenerator. The external circuit usually includes a blower to supply combustion air, a fuel pump and nozzle, a combustion chamber and an air pre-heater which recovers heat from the exhaust gases. Figure 4-1 shows a single cylinder Stirling engine and its components nomenclature.

4.1 The Ideal Stirling Cycle

The ideal Stirling engine cycle is shown in Figure 4-2. The hot side of the ideal engine is always maintained at a constant temperature T_h and the cold side at a temperature T_c. At the start of the cycle (position 1), all of the working gas is in the cold space cylinder at temperature T_c. As the cold piston starts compressing, the gas is continually cooled such that the compression process (1-2) is isothermal at a temperature T_c.

Process 2-3 is a movement of both pistons to displace the gas from the cold space to the hot space. Moving through the regenerator, the gas picks up heat stored in the regenerator and is heated up to the temperature T_h before it emerges into the hot space. The heating of the gas causes a rise in pressure; however, since there is no change in volume, no net work is generated during the displacement process.

Process 3-4 is an expansion stroke at elevated temperature. During the stroke, heat is supplied to the gas to keep the temperature T_h constant. Since this work stroke is carried out at higher gas pressure than the compression stroke, the work output of process 3-4 is greater than the work required for the compression process 1-2, and the engine produces net power.

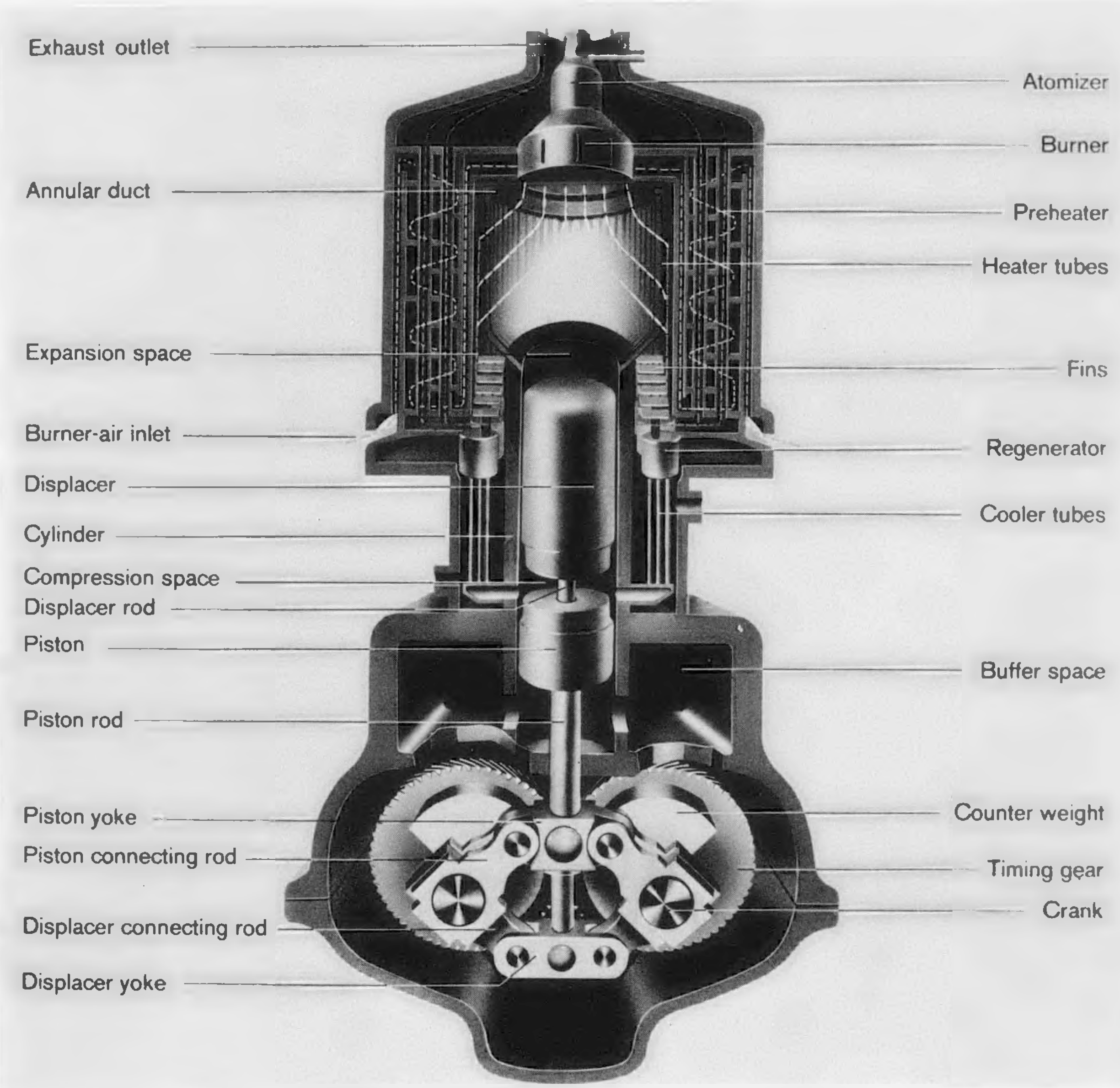
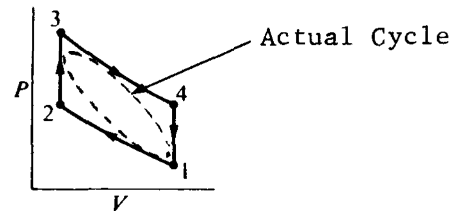


Figure 4-1 Stirling Engine Nomenclature (courtesy Philips)

(a)



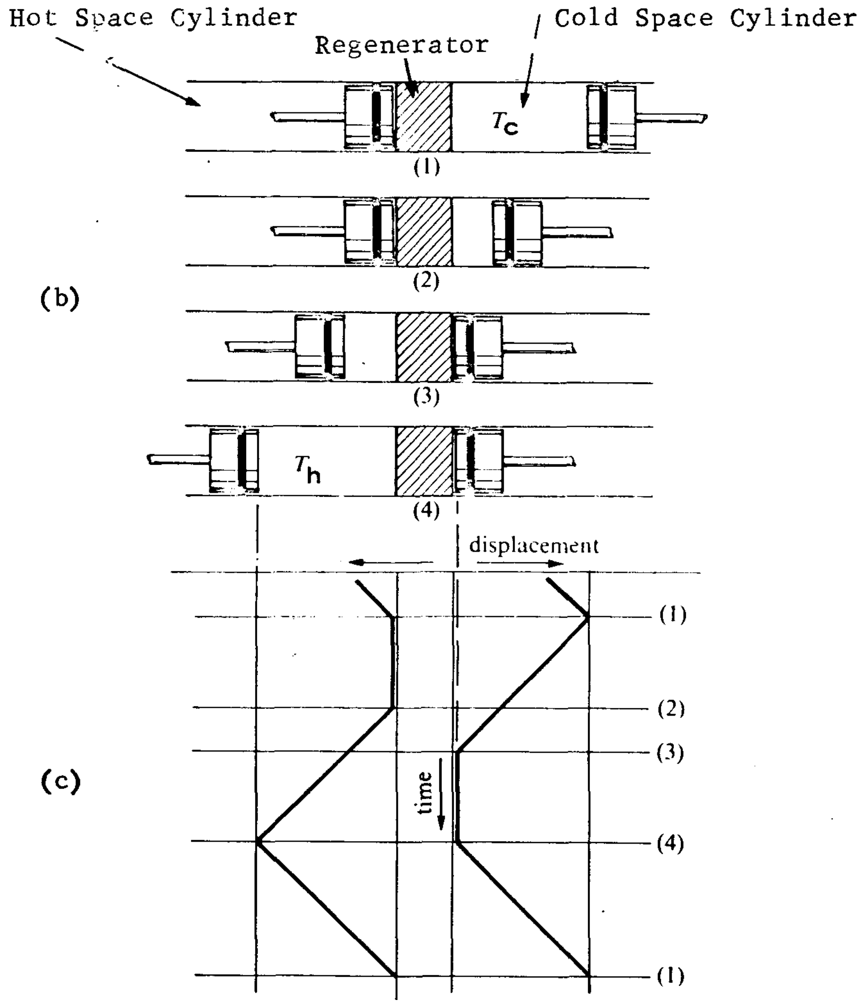


Figure 4-2 <u>The Ideal Stirling Cycle</u> (ref. 36)

(a) P-V diagram
(b) Piston arrangement at the terminal points of the cycle.
(c) Time - displacement diagram

Process 4-1 is a movement of both pistons to displace the expanded gas back into the cold space. Passing through the regenerator, the gas is cooled to the temperature T_c and the regenerator is recharged. The gas pressure goes down, but since the volume is constant, no net work is done during the process.

Thus, the ideal cycle is composed of four thermodynamic processes requiring heat transfer:

Process 1-2 - isothermal compression; heat transfer from the working fluid at T_c to an external heat sink.

Process 2-3 - constant volume displacement from cold space to hot space; heat transfer to the working fluid from the regenerator matrix.

Process 3-4 - isothermal expansion; heat transfer to the working fluid at T_h from an external heat source.

Process 4-1 - constant volume displacement from hot space to cold space; heat transfer from the working fluid to the renerator matrix.

Since the heat transferred in process 2-3 has the same magnitude as in process 4-1, the only heat transfer between the engine and its surroundings are: (a) heat supply at T_h and (b) heat rejection at T_c. This heat supply and heat rejection at constant temperature satisfies the requirement of the second Law of Thermodynamics for maximum thermal efficiency. In addition, in the ideal cycle, all heat transfer and gas flows are reversible, thus the efficiency of the ideal Stirling cycle is the same as the Carnot cycle, i.e., $\eta = (T_h - T_c)/T_h$. The principal advantage of the Stirling cycle over the Carnot cycle lies in the replacement of two isentropic processes by two constant volume processes, which greatly increase the area of the P-V diagram. Therefore, to obtain a reasonable amount of work from the Stirling cycle, it is not necessary to resort to very high pressures and swept-volumes, as in the Carnot cycle.

4.2 Actual Stirling Engine Cycle

The ideal Stirling engine cycle is very difficult to carry out in practice due to the difficulty in achieving reversible constant volume heating and cooling processes and reversible isothermal compression and expansion processes. Practical engine mechanisms involve rapid and continuous motion of the pistons, typically approaching sinusoidal motion, with approximately 90° phase displacement between cold and hot pistons. Consequently, the gas displacement processes overlap with the expansion and compression processes, thereby rounding the sharp corners of the ideal P-V indicator diagram. Also, the practical engine operates at significant speed which requires heat transfer and gas flow at significant rates. The required temperature and pressure differences result in imperfect regeneration and reduced pressure on expansion. In addition, the limited heat transfer surface area in the hot and cold cylinders causes the expansion and compression processes to be intermediate between isothermal and isentropic (no heat transfer) processes, probably more nearly isentropic.

The non-ideal piston motion and losses due to finite engine speed combine to reduce the indicated work below that of the ideal cycle by rounding the corners and narrowing the diagram, producing an overall diagram as shown on Figure 4-2. Well designed engines can achieve an indicated efficiency of about 65-70% of the Carnot efficiency, but when mechanical losses, combustion system losses, and auxiliary power requirements are considered, the net engine efficiency is typically only about 50% of the Carnot efficiency. See Appendix C for a more detailed discussion of the Stirling engine efficiency.

4.3 Types of Stirling Engines

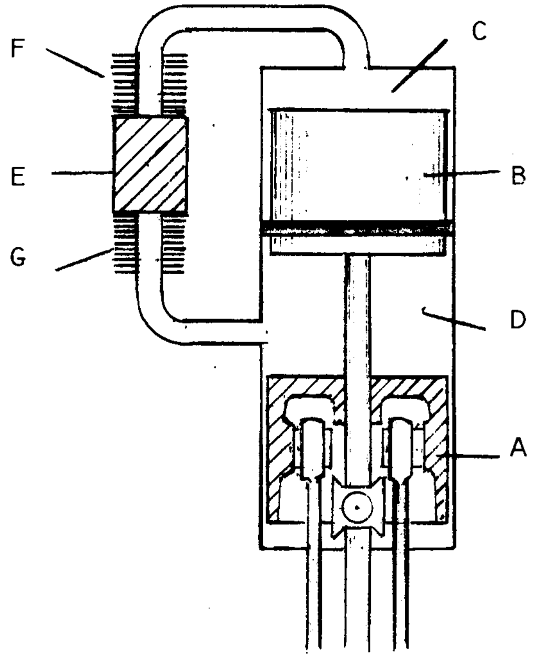
The mechanical arrangement of modern Stirling engines can be divided into two basic classes -- displacer and double-acting machines. Displacer type engines require two pistons for each power unit and may be made with either multiple cylinders or a single cylinder (when both displacer and power pistons are incorporated into the same cylinder). Double-acting engines, by contrast, must always be made with from 3 to 7 cylinders per power unit, the most common being 4 cylinders.

4.3.1 Displacer Engines

The displacer-type Stirling engine involves one power piston and one displacer piston for each power unit (Figure 4-3). These pistons operate approximately 90° out of phase. The power piston is provided with substantially gas tight sealing rings and operates with significant pressure differences across it, while extracting net positive work throughout the course of a complete cycle. By contrast, the displacer piston simply displaces gas back and forth from the hot (expansion) chamber through the heater, regenerator and cooler to the cold (compression) chamber. The only pressure difference across the displacer piston is due to the small flow resistance through the heater, regenerator, cooler circuit. As a result, the displacer piston does not require very effective sealing rings. There is essentially no net work produced or required by the displacer except that required to overcome the small friction forces and flow resistance.

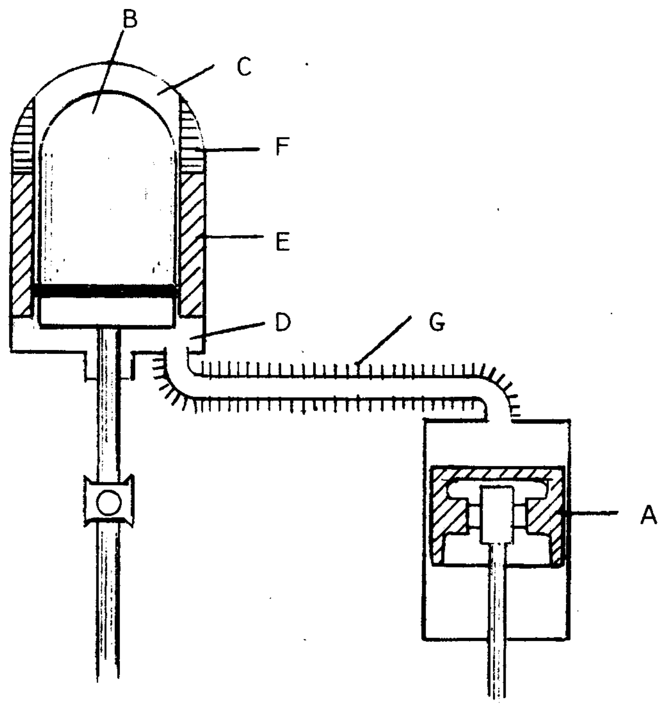
There are many possible mechanical configurations for displacer-type engines, but the two principal variants are:

1. Power piston and displacer in same cylinder (Figure 4-3a).
2. Power piston and displacer in separate cylinders (Figure 4-3b).



A - piston
B - displacer
C - hot space
D - cold space
E - regenerator
F - heater
G - cooler

(a) Piston - displacer in the same cylinder



(b) Piston - displacer in separate cylinders

Figure 4-3 Two Basic Arrangements of Displacer-Type Stirling Engines

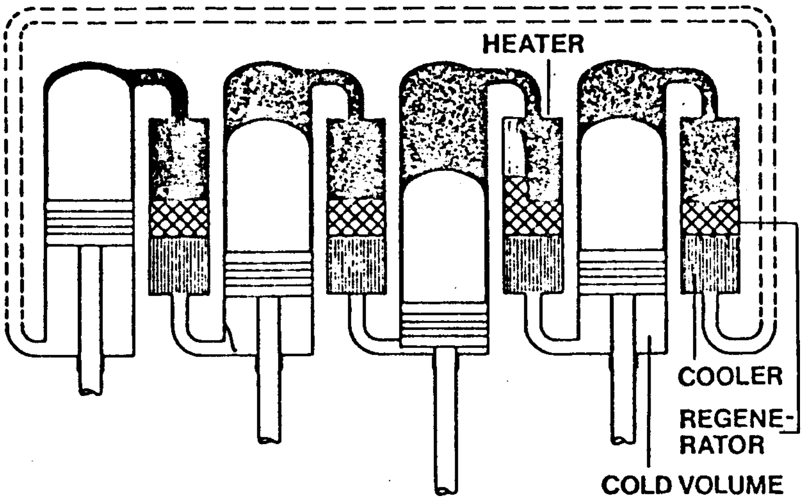
Practical engines are type 1, for example the original engine of Robert and James Stirling, Figure 4-10, and early Philips engines, Figure 4-1, because the type 2 engines require more dead volume in the cold space, and hence, have lower efficiency and specific power. Type 2 engines are considered only as a means to avoid the complex co-axial piston displacer drive required in the type 1 engine.

4.3.2 Double-Acting Engines

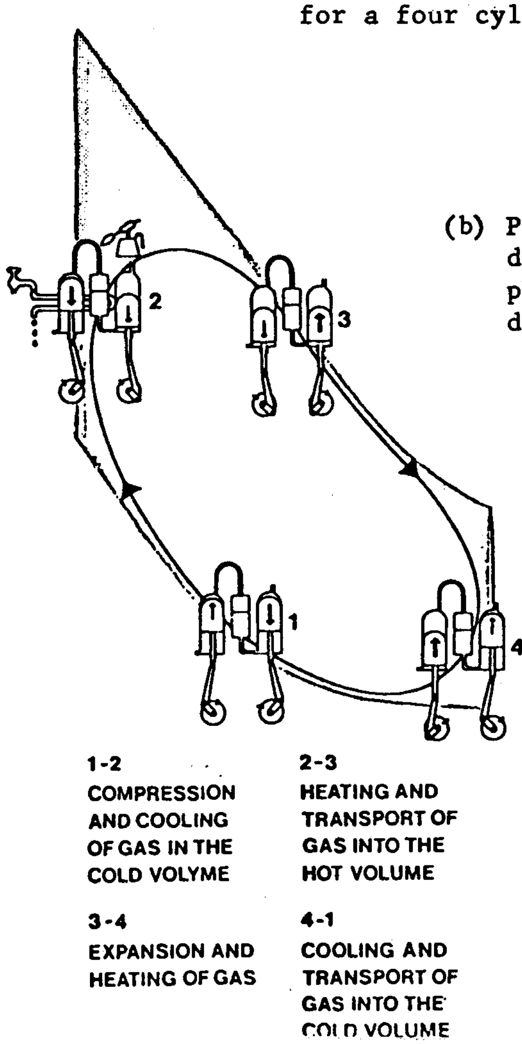
Rinia (Ref. 1) at Philips invented the multi-cylinder double-acting Stirling engine scheme early in the Philips development program. Although it was abandoned soon afterward due to sealing and lubrication problems, recently it has been under development for use in mobile power systems because it offers the most compact light-weight, and lowest cost arrangement.

Each of the multiple double-acting cylinders has a hot space on the head end and a cold space on the rod end, Figure 4-4. There are the same number of Stirling cycle modules as cylinders with the hot space of a cycle module in one cylinder and the cold space in an adjacent cylinder. Thus, the hot space of one cylinder is connected through a heater, a regenerator and a cooler to the cold space of an adjacent cylinder to form a cycle module, as is illustrated in Figure 4-4 for a 4-cylinder engine. The cycle modules are symmetrically phased in the engine cycle. Thus, in the 4-cylinder engine; the pistons are phased 90° and consequently, the 4 gas pressures of the 4-cycle modules are phased 90°. This principle can be applied to engines having from 3 to 7 cylinders; however, maximum efficiency is achieved with from 4 to 6 cylinders.

Double-acting Stirling engines have been configured with cylinders in-line, in a Vee arrangement and in a co-axial or "barrel" arrangement. Drive mechanisms have included a single crankshaft for the in-line



(a) Arrangement of heat exchangers and cylinders for a four cylinder double-acting engine



(b) Principles of operation for a double-acting engine (working phase related to the PV diagram of a real engine

Figure 4-4 Double-Acting Stirling Engine Principle (courtesy USS) and Vee, double-parallel crankshafts for the co-axial with 4 cylinders, and both Z crank and swashplate drives for 4 or more co-axial cylinders in a barrel arrangement. These configurations are illustrated in Figure 4-5, 4-6 and 4-7. The relative merits of these various configurations are discussed further in Section 5.1.2, and in Refs. 33 and 36.

4.3.3 Opposed-Piston Engines

A discussion of Stirling engine mechanical configurations would not be complete without mentioning the opposed-piston configuration illustrated in Figure 4-8. This engine has two single-acting pistons in separate cylinders, one for the hot space and one for the cold space. As in the double-acting engine, the cycle module consists of the hot space connected through the heater, the regenerator and the cooler to the cold space.

The opposed piston concept has several possible mechanical arrangements as illustrated in Figure 4-9. The simplest of these is the Vee which has been used in many small machines (Ref. 13) and investigated in several design studies.

A single module opposed piston engine is frequently attractive for small engines with a pressurized crankcase and rotary shaft seal, but is undesirable for large stationary engines. When a pressurized crankcase is employed, the opposed piston module performs essentially like one module of a double-acting machine, with similar performance characteristics. However, large engines cannot have a pressurized crankcase without suffering a severe weight and cost penalty. With an atmospheric crankcase the piston ring friction, wear and seal leakage become excessive due to the high ΔP across the rings and it is necessary to employ a cross-head and pressurized buffer space below each piston (as in the single cylinder displacer engine - Figure 4-1), with a piston rod seal for working fluid retention. Once these

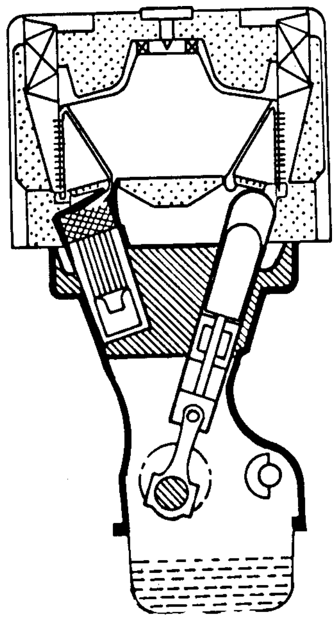


Figure 4-5

Vee Design - Single Crank

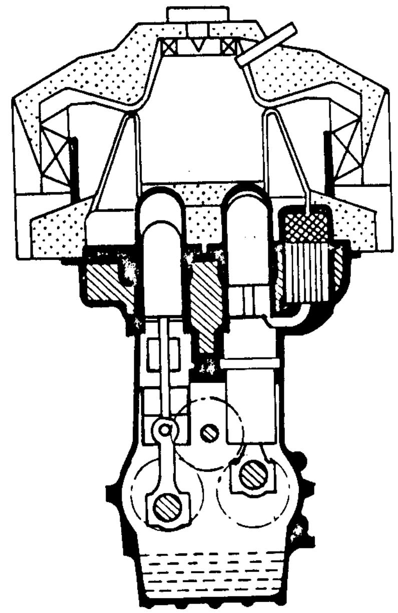


Figure 4-6

Parallel Cylinders
Twin Cranks

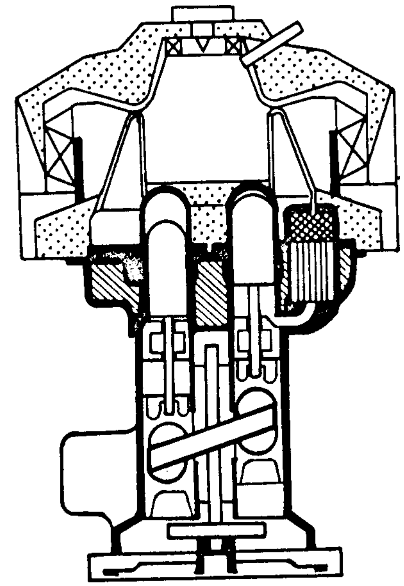


Figure 4-7

Swashplate Configuration

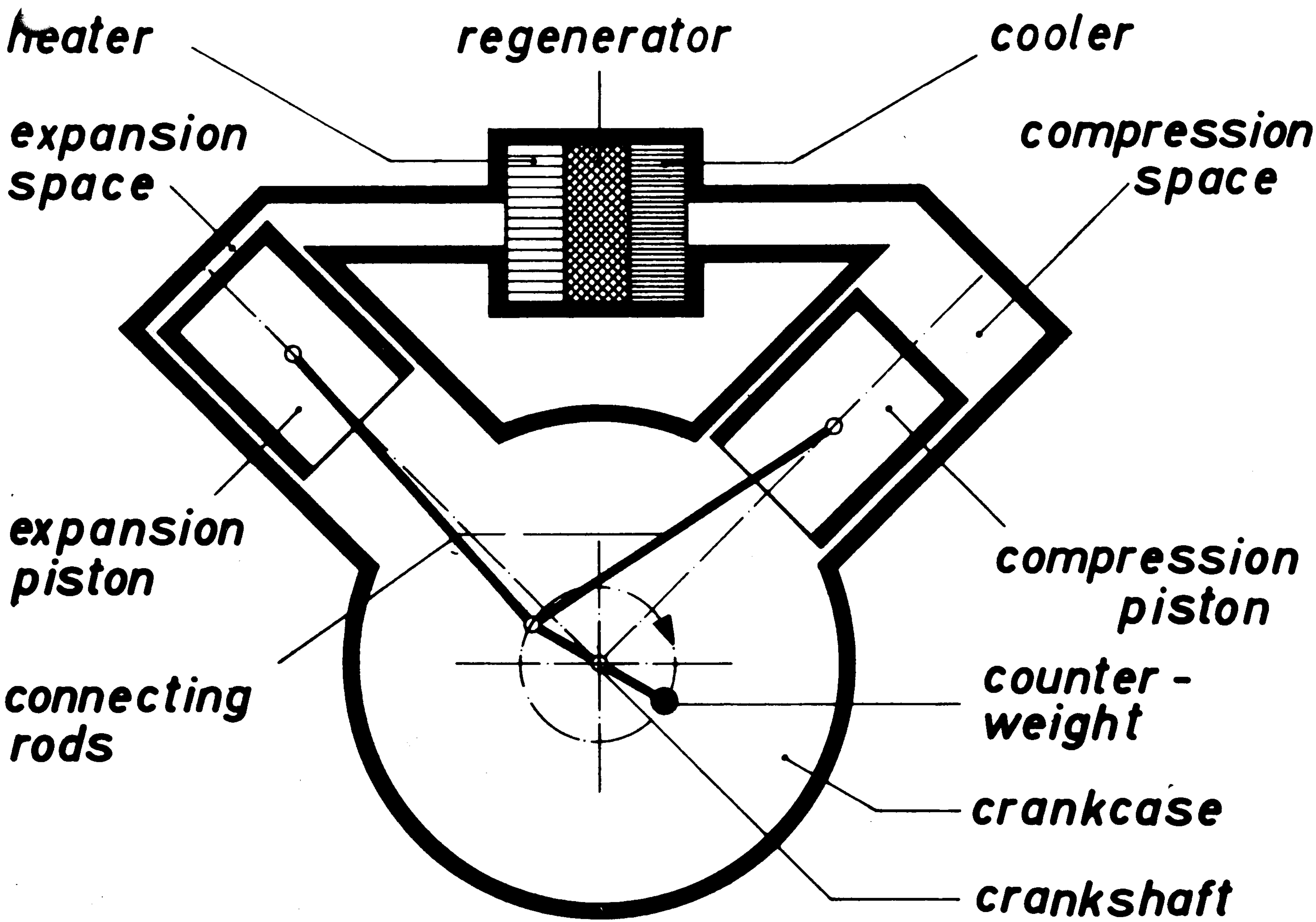
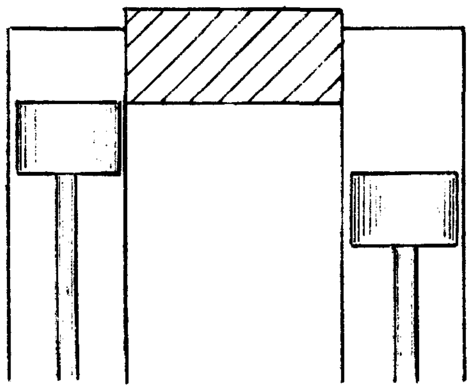
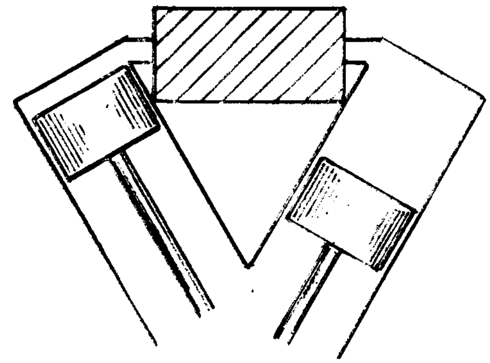


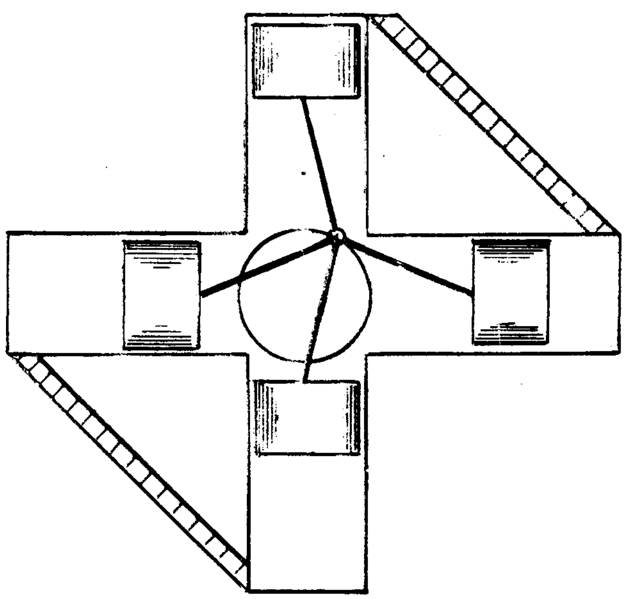
Figure 4-8 Opposed - Piston Configuration



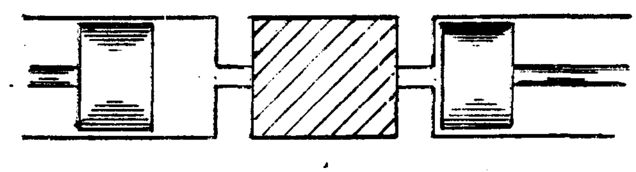
Parallel Cylinder



Vee Cylinder



Rotary Cylinder



Opposed Piston

Figure 4-9 Alternative Arrangements of Single-Acting Two-Piston Engines features are added, we have all the elements of the double-acting engine and would therefore benefit from using the double-acting principle instead.

Stirling engines have been suggested that utilize rotary positive displacement mechanisms (Ref. 36, p 58). Although this arrangement has a compact mechanism, it has two serious flaws. First, high temperature sliding seals are required; and second, metal parts move continuously from the hot space to the cold space without thermal regeneration, thereby producing a high heat leak.

For a more complete discussion of Stirling engine mechanical arrangements, refer to Walker (Ref. 36), and Finkelstein (Ref. 3).

4.4 Historical Development of Stirling Engines

In the late eighteenth, and early nineteenth centuries, several hot air engines were invented and developed (Ref. 3). The closed cycle regenerative engine, which is the subject of this discussion was invented by Robert and James Stirling in 1816. The original Stirling engine, a single cylinder displacer type, is illustrated in Figure 4-10.

Subsequently, throughout the nineteenth and early twentieth centuries, thousands of engines of varying configurations and up to 300 hp were in commercial use. These engines were safer than steam engines of the period, and their efficiency and reliability were reasonable compared to steam engines of the same period. Nonetheless, they were completely displaced, along with their steam counterparts, by the arrival of the internal combustion engine.

Figure 4-11, shows some other configurations of early Stirling engines, which include the two types of displacer engines and the opposed piston engine previously described.

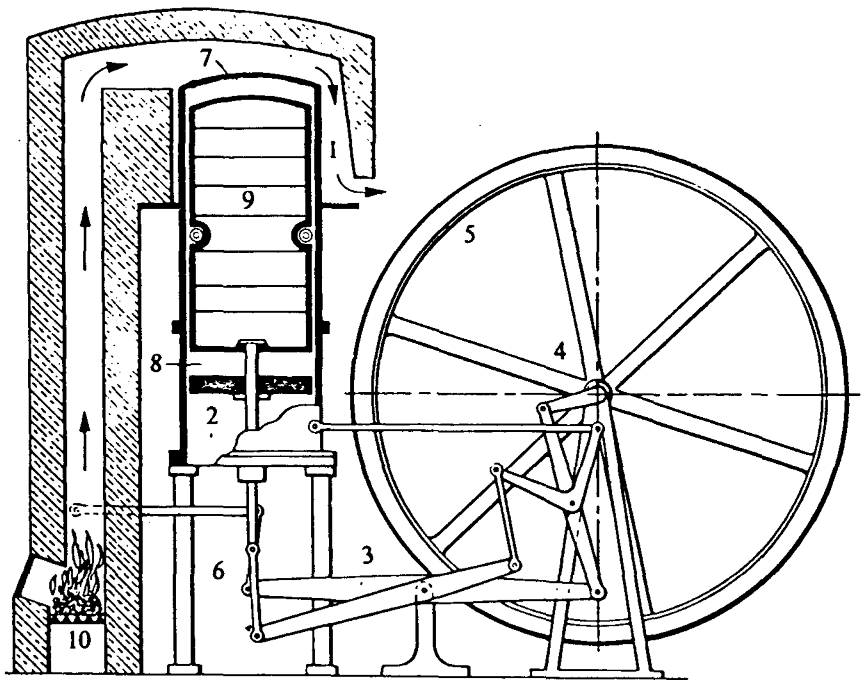


Figure 4-10 The Original Stirling Engine. Reproduction of a Drawing Showing the First Stirling Engine, From the Original Patent Specifications of 1816. Such an Engine Was Used in 1818 for Pumping Water .

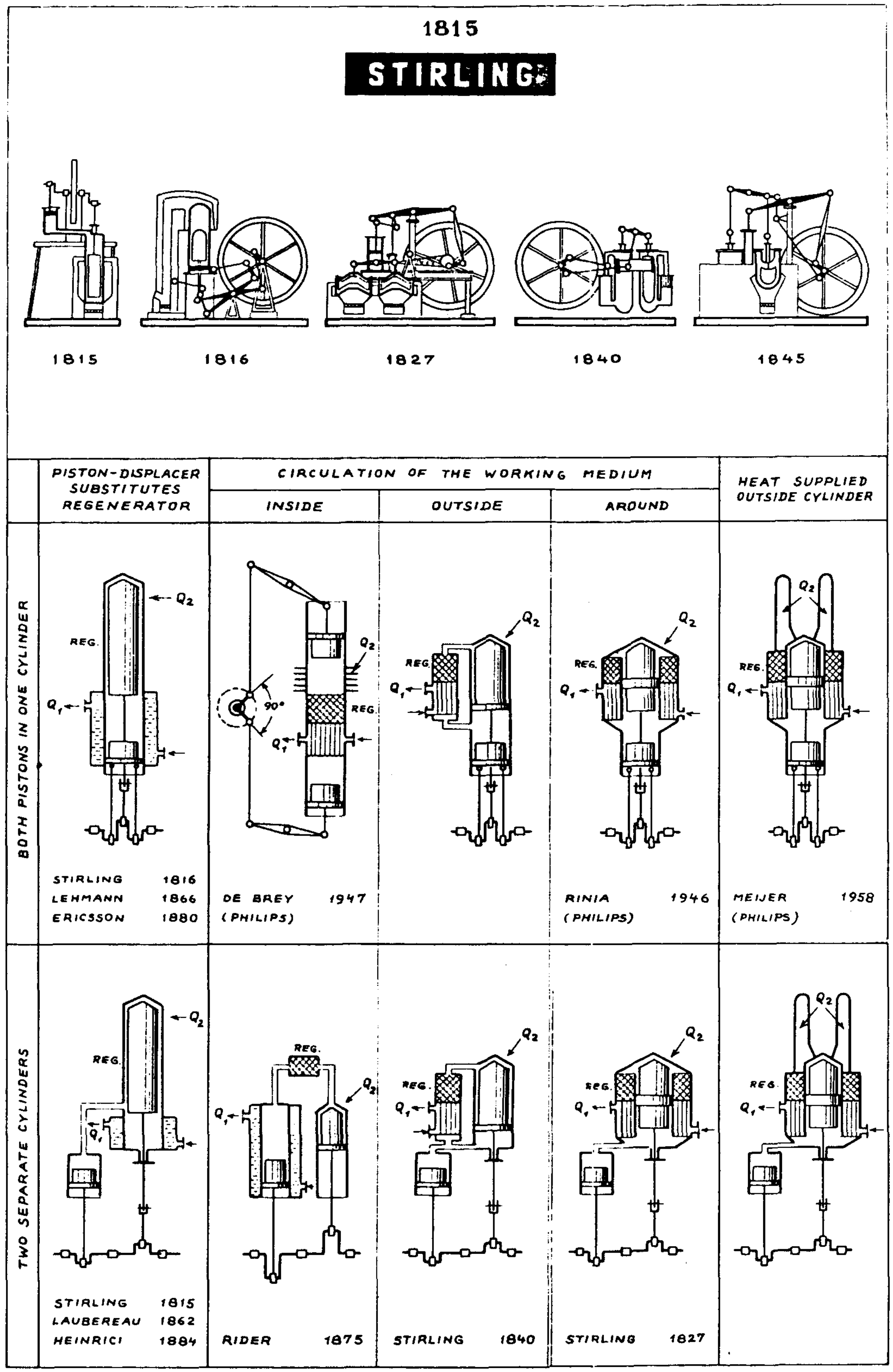


Figure 4-11 Schematics of Engine Types (courtesy Longman Group Ltd)

4.4.1 Philips Contributions

About 1938, the Philips Research Laboratories at Eindhoven, Holland, first became interested in the hot air engine to drive a generator unit for operating radio receivers in remote areas. Several small hot air engines were built (Ref. 1). A schematic diagram of one of these early Philips engines is shown in Figure 4-12. It can be seen that this engine has a pressurized crankcase and a displacer which is actuated by a rocker-arm driven from a point on the piston connecting rod.

In 1948, the Philips Stirling engines were still operating on air as the working fluid with a pressurized crankcase and an overall thermal efficiency of about 15%. With the invention of the transistor, battery operated radios became practical and the small Stirling driven generator lost most of its potential market. By 1952, work on Stirling engines at Philips was nearly stopped.

Gradually, a new group of engineers and physicists interested in cryogenics began to replace the old group. They concentrated on achieving a low temperature device instead of an engine. Hydrogen replaced air to reduce pumping losses. Improved heat exchangers and regenerators were constructed and a more accurate cycle analysis was developed. By 1954, the new group succeeded in making liquid air with the Stirling cycle machine. About the same time a new drive mechanism with twin crankshafts, called the "Rhombic drive", was invented by Dr. Meijer (Ref. 61) to solve the balancing and phasing problems in single cylinder displacer engines (Figure 4-13). With encouragement from Philips' management, they made a new start on the more difficult task -- development of a commercial Stirling hot gas engine.

Philips considered the rhombic drive to be a major advance because: (1) it provided the possibility for single cylinder machines to be completely balanced; (2) it eliminated the friction and wear caused by side-

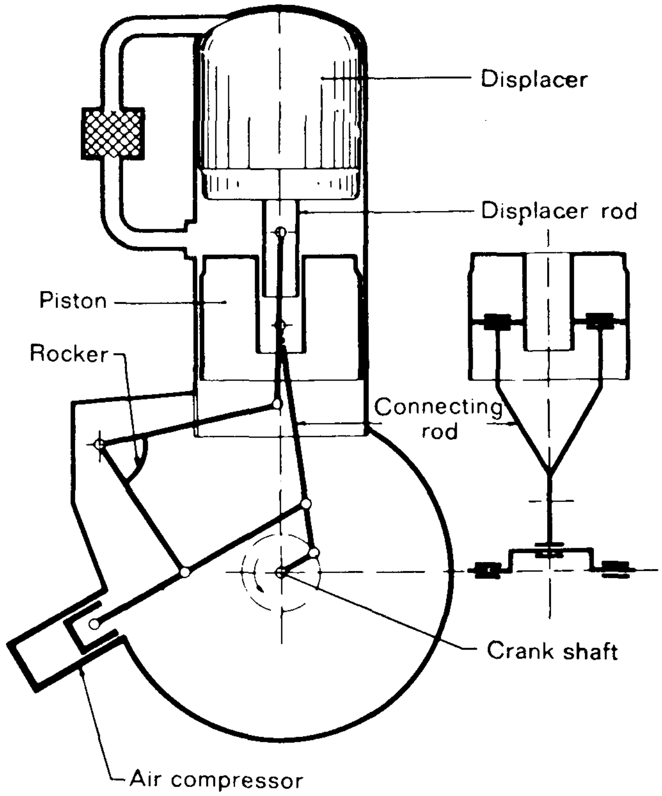


Figure 4-12 Schematic of a Crank Drive of an Early Philips Engine

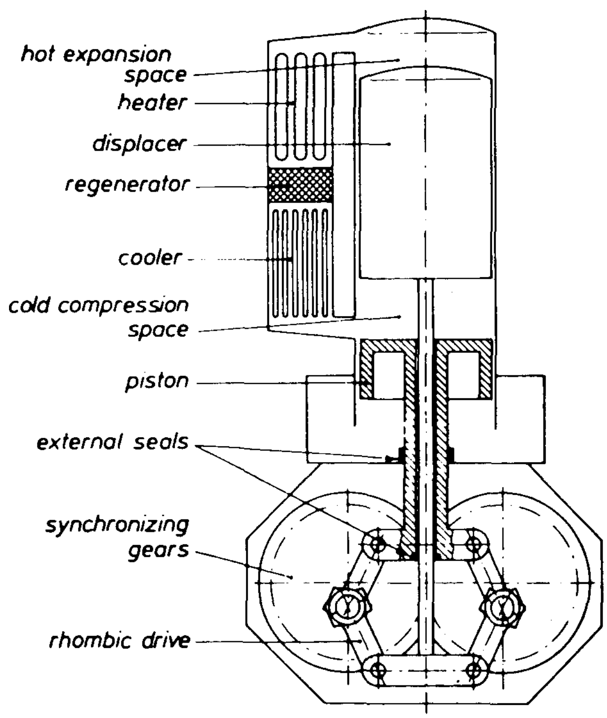


Figure 4-13 Schematic of Displacer Type Engine - The Rhombic Drive
(courtesy Philips)

thrust of the piston and displacer crosshead; and (3) it eliminated the need for a pressurized crankcase. In spite of these advantages, the rhombic drive is a rather complex, bulky and expensive mechanism.

Throughout the fifties and sixties, Philips concentrated their engine development efforts on the rhombic drive displacer-type engines. Both single cylinder and multi-cylinder designs were built and operated, although most of the research effort was devoted to single cylinder versions. During this period, Philips and their licensees built approximately 40 single cylinder rhombic engines. Table 4-1 shows specifications for these engines. Most were in the 8-15 hp range, several in the 30-40 hp range and approximately 6 were in the 80-100 hp range. During the same period, about 8 multi-cylinder rhombic drive engines were built to demonstrate the application of the single cylinder building block to larger engines. Several engines were installed in buses, boats, garden tractors, etc., to demonstrate the Stirling engines' desirable characteristics (Refs. 12, 14, 16, 24).

One of the major strides made by Philips during this period was the development of detailed analytical methods for Stirling engine cycle and component design. This work began in the late 1950's, and was constantly improved until by the late 1960's, it was possible to optimize engine designs analytically with good agreement between analyses and test results. The basic calculation procedures, loss analyses, and computer programs have remained highly confidential and proprietary to Philips, and are available only to their licensees.

Throughout this period, Philips also did much pioneering work on development of engine designs, heater head designs and construction techniques, piston and drive rod sealing methods, regenerator design, combustion and control systems, helium and hydrogen containment, sodium heat-pipe development, etc. Out of this work, Philips developed a significant patent estate and much proprietary "know-how" relating to Stirling engines. Since they were the only one working in this field during that period, they developed a virtual monopoly position as a result of broad patent coverage on Stirling engine features. This position led to licensing agreements with several other companies who became interested in Stirling engine possibilities.

4.4.2 Licensees; 1958 - 1972

Philips' success with cryogenic machines and engines spurred the interest of General Motors Corporation, which executed license agreements with Philips in 1958. The agreements provided for a 10 year information exchange with provision for mutual licensing of patents related to Stirling engines. GM subsequently engaged in Stirling engine development until they lost interest in 1969 and cancelled their agreements in 1970. Most of the GM work during this period, was carried out at their Research Laboratories, but significant work was also performed at the Electromotive and Allison Divisions. The collaboration between GM and Philips during this period resulted in significant advances in the development of practical Stirling engines.

In 1968, the West German Diesel engine manufacturers MAN-MWM secured a license to build Stirling engines and in the same year, a Swedish consortium, United Stirling AB also signed a license agreement with Philips. These two groups have been pursuing Stirling engine development since that time.

After the GM agreement was terminated in 1970, the Ford Motor Company became interested in Stirling engines as a potential clean engine for automobiles. They signed a license agreement with Philips in 1972 and

TABLE 4-1 a

CHARACTERISTICS OF SINGLE CYLINDER DISPLACER TYPE STIRLING ENGINES BUILT BY PHILIPS AND GENERAL MOTORS UP THROUGH 1970

	10-36 Engine	GPU 2 Engine	GPU 3 Engine	3015 (Calvair)	I-98 Engine	I-98 Engine	4S1210 Navy 4 Cyl.	EMD 1-S1050	EMD 2W17A
Max. HP	7.47	7.3	11.2	40	21	25	380	75	138
@ RPM	3600	3600	3600	2500	3000	3500	1500	1500	1800
Max. Eff.	26.3	28.03	26.5	39	30	33	35	28	28.4 HHV
@ RPM	1800	2400	1900	1400	2200	1200	750	1200	900
Heater Wall (°F)	1400	1400	1400	1270	1300	1300	1202	1270	1100
Water in (°F)	75	126	100	70	60	60	90	100	100
Working Gas	H_2	H_2	H_2	H_2	H_e	H_2	H_2	H_2	H_2
Mean Press (psi)	1000	1000	1000	1560	3200	3200	1500	1436	1100
No. Cyl.	1	1	1	1	1	1	4	1	2
Bore (in)	2.362	2.375	2.75	3.47			5.70	5.70	6.50
Stroke (in)	1.238	1.238	1.238	2.37			2.90	2.90	3.20
Displ. (in^3)	5.44	5.47	7.32	22.3	5.97	5.97	296	74.0	212.0
HP/In^3	1.37	1.33	1.53	1.79	3.52	4.19	1.28	1.01	0.65
Wt. (lbm)	127.4*	90*	127*	550*	200	200	5000	2300**	3800**
L (in)	14	12	14	17.5	14	14	74	36	36.25
W (in)	14	16.5	15.5	17	14	14	40	27.5	62.25
H (in)	28	26.5	28	37.25	30	30	76	65	84.63
Vol. (Ft^3)	3.17	3.04	3.51	6.42	3.4	3.4	130.0	37.2	111.0
HP/Ft^3	2.36	2.40	3.19	6.25	6.18	7.35	2.92	2.02	1.24
#/HP	17.1	12.3	11.30	13.75	9.52	8.0	13.20	30.60**	27.50**
BMEP (psi)	151	147	168	284			339	268	143

* Bare engine with pre-heater.
** Without flywheel.

TABLE 4-1 b

CHARACTERISTICS OF MULTI CYLINDER STIRLING ENGINES DESIGNED BY PHILIPS AND GENERAL MOTORS THROUGH 1970

	Philips Boxer	Philips In-Line	GMR In-Line	GMR In-Line	GMR VEE 1	Swash Plate	Torpedo***
Max HP	120	200	148*	129	2	22	690 (Gross)
@ RPM	3000	3000	2000	2000	5000	2400	3000
Max. Eff.	(36.5)		33	30	28.2	28.5	52 (No comb. loss)
@ RPM	3000		1500	1000	3200	2400	1200
Heater Wall (°F)	1292	1292	1400	1400	1300	1264	1500
Water in (°F)	104	104	125	125	170	150	60
Working Gas	He	He	H_2	He	H_2	He	H_2
Mean Press. (psi)	1720	3140	1500	1500	1500	1500	3500
No. Cyl.	4	4	4	4	1	4	5
Bore (in)	3.26	3.26	4.00	4.00	1.18	1.57	3.40
Stroke (in)	1.97	1.97	1.83	1.83	1.26	1.57	2.30
Total Displ. (in^3)	66.0	66.0	92	92	1.38	12.1	104.0
HP/In^3	1.82	3.04	1.61	1.40	1.45	1.82	6.64
Wt. (lbm)	850	880	1000	1000	24.6**	200	600
L (in)	59	44.5	41	41	10	30	21 (Inc. gear)
W (in)	39.7	17.3	18	18	10.5 dia.	12 dia.	19 dia.
H (in)	17.3	37.9	34	34			
Vol. (Ft^3)	23.5*	16.9*	14.5*	14.5*	.50	1.96	3.44
HP/Ft^3	5.1	11.82	10.2	8.9	4.00	11.21	201.0
#/HP	7.10	4.40	6.75	7.75	12.30	9.10	.87
BMEP (psi)	240	400	318	277	115	299	815

* Volumes include accessories.

** With ring for isotope heat but less isotope capsules.

*** All data w/o comb. system.

began a joint program to develop a compact, double-acting swashplate engine for automobiles. All of these latter three licensees are currently engaged in Stirling engine development. Their activities are discussed briefly in the following section while the GM work, now discontinued, is discussed below, and is more completely reported in (Ref. 33).

General Motors

The incentive for General Motors to develop the Stirling engine in 1958 was the interest shown by various GM Divisions in marine propulsion, locomotive power and generating sets, as well as military and space applications. More specifically, the Allison Division anticipated an Air Force contract for a Stirling solar-heated satellite powerplant, while Cleveland Diesel Engine Division believed the Stirling could compete with the Diesel for river and harbor workboat propulsion, as well as for submarines. There was no interest by anyone at that time in road vehicle power. It was believed that cost, bulk and weight would be excessive; also that higher heat rejection would make it impossible to install radiators. Consequently, GM made no investigation of Stirling vehicle propulsion until 1962.

GM received considerable encouragement from the U.S. Army Engineer Laboratories, Ft. Belvoir, Va., to develop small quiet generator sets to replace existing internal combustion engine units. Consequently, GM Research Laboratories (GMR), soon embarked on a program to develop such units based upon a 8-10 hp single cylinder rhombic drive engine. Much of the early work at GMR was done on this small single cylinder engine. Basic work proceeded on several fronts, such as piston rings, and piston rod seals; low-cost, high-effectiveness regenerator materials; engine friction reduction; heater head materials, fabrication techniques and durability; liquid fuel combustion development, including exhaust emissions reduction; air preheater design and cost reduction; noise/vibration studies;

and engine design analysis. Steady progress was made, and in 1964 the first Army Ground Power Unit designated GPU 2-1 completed its 500 hour test at Ft. Belvoir. A revised design, called GPU 3, was delivered to the Army in 1966. These units are described by Heffner (Ref. 6).

GM also became interested in the possibilities of multi-cylinder Stirling engines for other applications and did a great deal of work toward the design of multi-cylinder machines. The rhombic drive was not believed to be attractive for multi-cylinder engines due to its complexity, close tolerance requirements, and excessive bulk and cost. General Motors, at their Electro Motive Division in the period 1964 - 1968 built several interesting designs of their own in the attempt to eliminate the objections of the rhombic drive. One, shown in Figure 4-14 included variable phase angle with direct reversing, for marine applications. It was a V-8 displacer-type design with the goal of about 800 Bhp. Only 4 cylinders were ever installed, and it achieved about 360 Bhp. A sketch of the engine cross section is shown in Figure 4-15. The power pistons were connected through crossheads to a single crankshaft on the engine center line. The displacers were driven from a separate crankshaft by means of a rocking lever, one set being built in each side of the V-8 crankcase. A differential gearing arrangement enabled the displacer shafts to be varied with respect to the main crankshaft.

Several other configurations for large engines were investigated by GM in their Electro Motive Division. Sketches of two of these are shown in Figure 4-16 and 4-17; inverted 'W', and 'W' designs. The 'W' engine (Figure 4-17), was built as a "one cylinder" version to prove feasibility, with the goal of a multi-cylinder engine; the inverted engine was never built. A double-acting vertically oriented power piston drove through crossheads to a central crankshaft, while two displacers

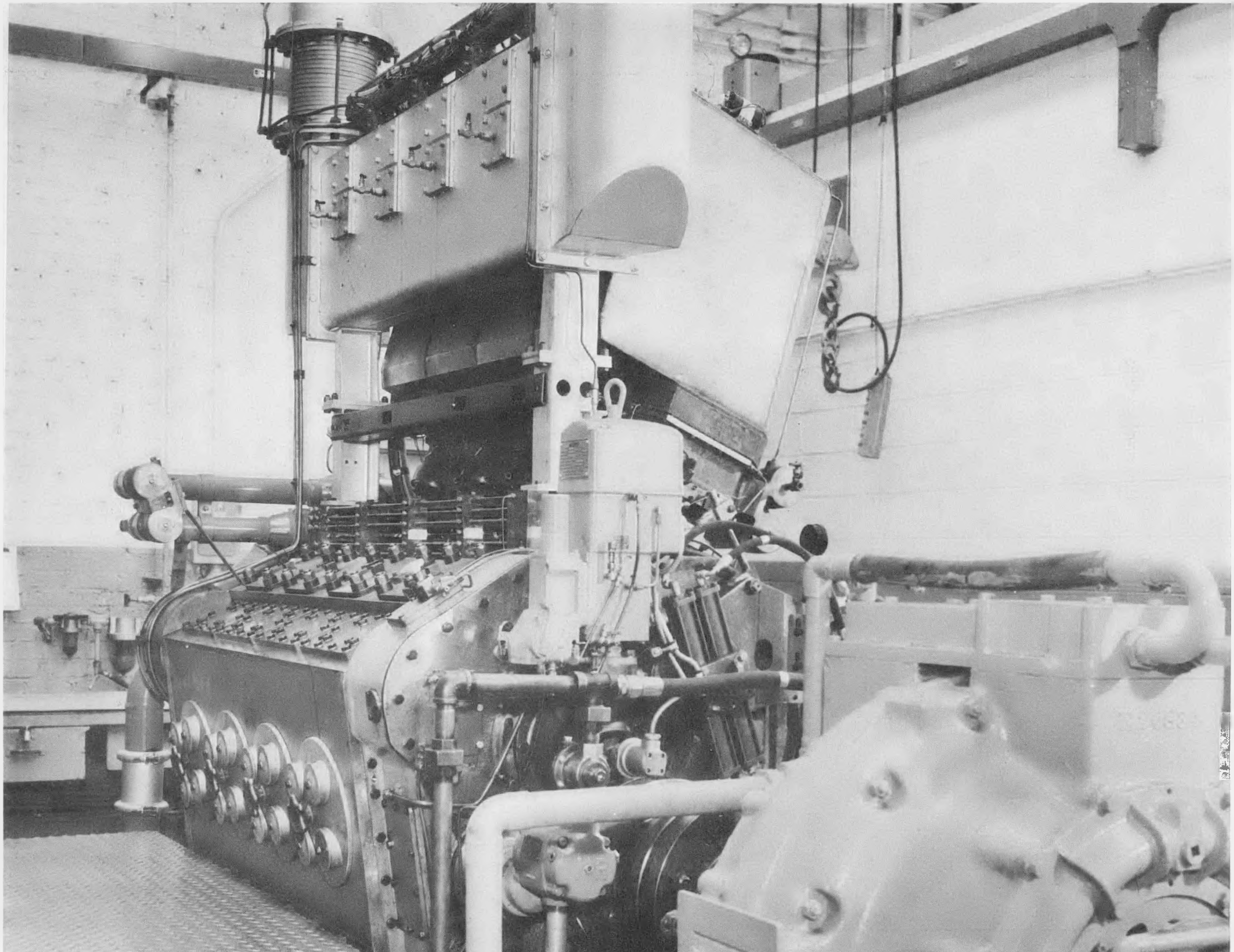


Figure 4-14 EMD V-8 Variable Phase Angle Displacer Engine ; the engine as shown with only four cylinders developed 360 BHP. (courtesy GM-EMD)

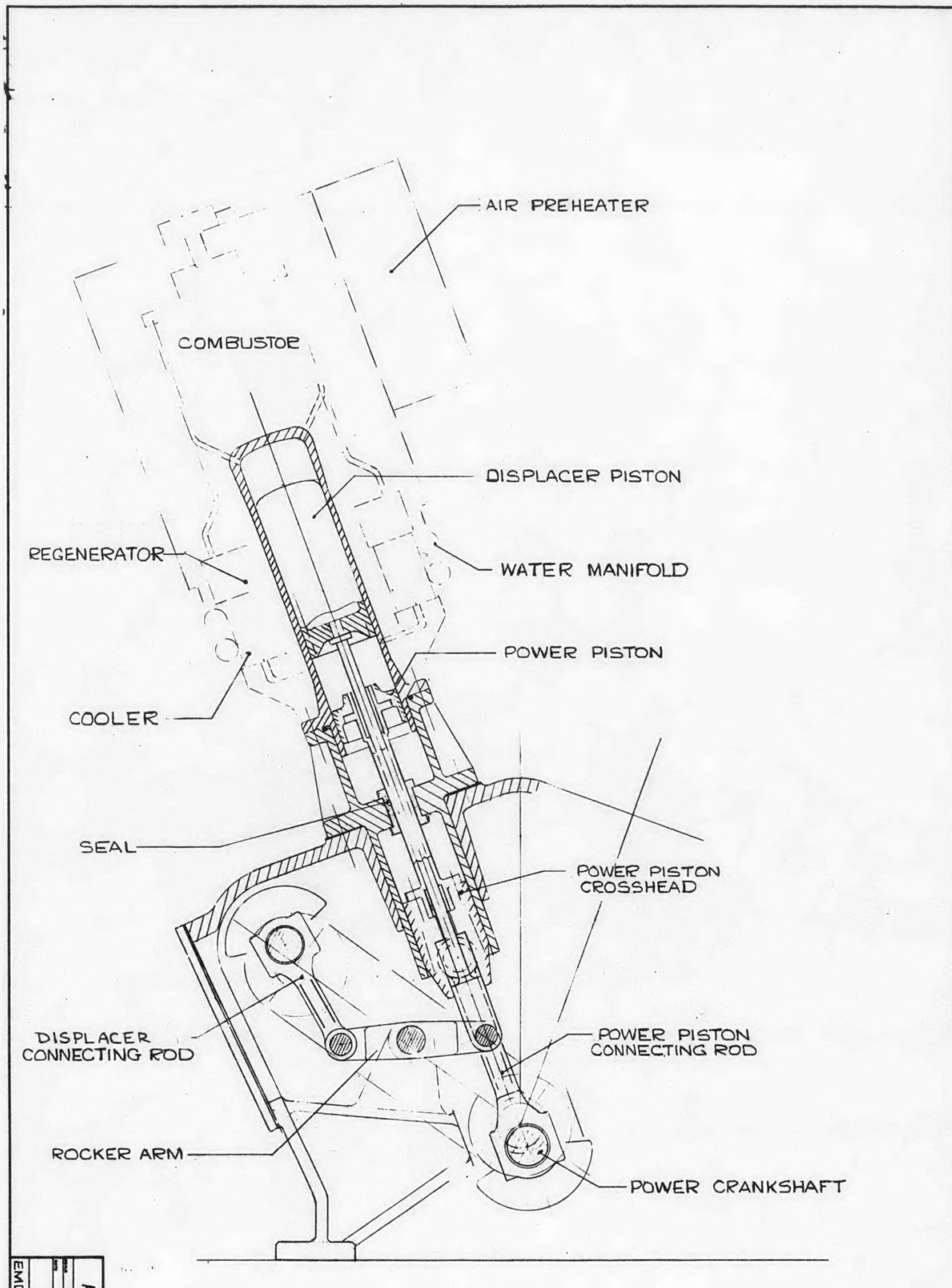
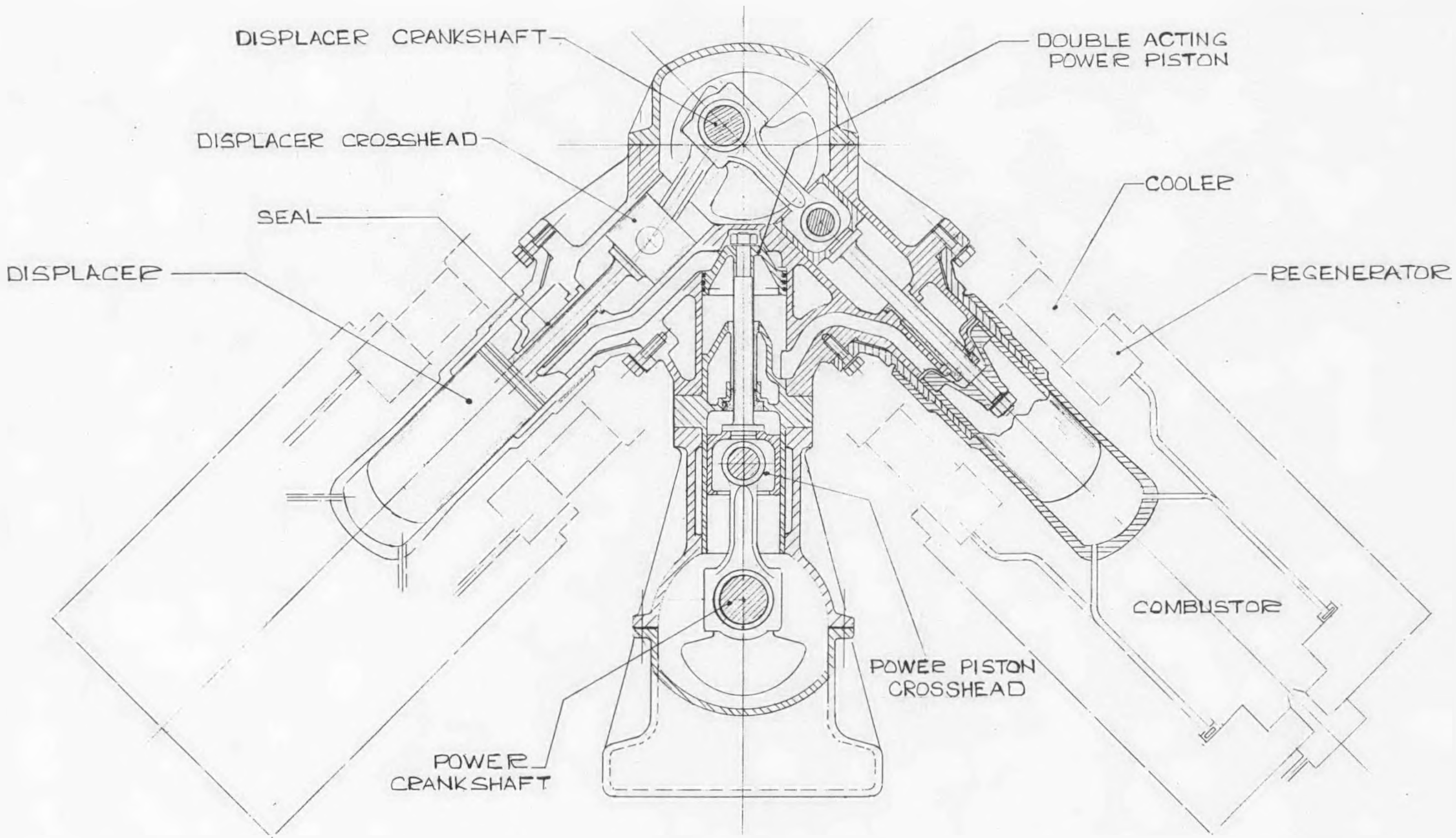


Figure 4-15 Sketch of EMD V-8 Variable Phase Angle Engine Configuration



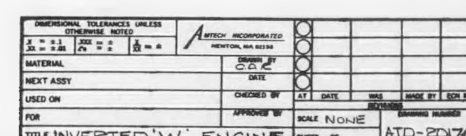


Figure 4-16 EMD Inverted 'W' Displacer Engine Design

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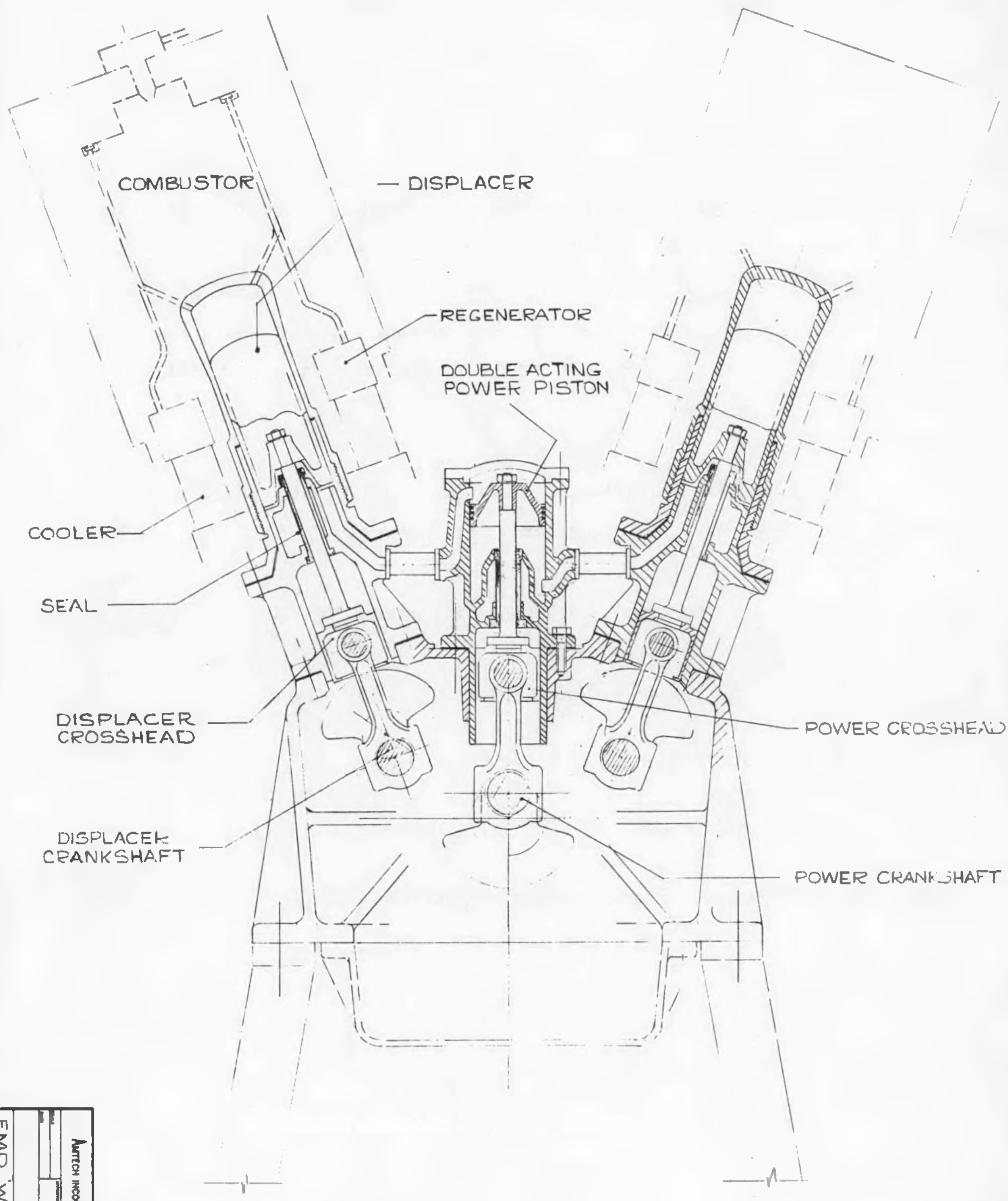


Figure 4-17 EMD 'W' Displacer Engine Design

EMD 'W' ENGINE

in the sides of the 'W' were each driven by their own crankshaft. Variable phase angle was not incorporated in either of the 'W' designs. The design was a progressive step on one hand because it eliminated one power piston in a 2-cylinder displacer engine; but it had the disadvantage of all so-called "separated cylinder" designs, because the cold clearance space was divided between 2 cylinders and therefore could not be reduced to the desirable lowest value achieved by a co-axial type displacer or a double-acting engine.

Other compact engine studies were performed by GMR from which they concluded that: (1) double-acting engines would be nearly half the specific volume of rhombic drive engines; and (2) double-acting swash-plate, in-line, and horizontally opposed engines would all be about the same size when "packing crate" volumes are considered and all auxiliary and accessory equipment is included.

In 1969, GM lost their interest in the Stirling engine and in 1970, they terminated their licensing agreement with Philips.

The following problems were reported by GM:

- Sealing problems around the piston rods.
- Weight and bulk.
- Low piston speed limit (1000 ft/min at that time).
- High temperature material costs.

4.4.3 Current Development Activities

There are three principal development activities currently underway:

1. Ford-Philips

2. United Stirling

3. MAN-MWM

as well as several other smaller activities by both small and large companies.

4.4.3.1 Ford-Philips

The Ford-Philips program is the most heavily funded Stirling engine development project currently active. This project has been supported by Ford since 1972, and more recently by the ERDA office of Advanced Automotive Propulsion Systems (Ref. 49). Under this program, Philips is developing a 170 hp 4 cylinder double-acting swashplate engine (Model 4-215) and working with Ford on the installation, test evaluation and development of the engine in a Ford Torino. A cross section of this engine is shown in Figure 4-18 and a cutaway pictorial view of the engine is shown in Figure 4-19. In addition, a study was made of an 80-100 hp sized (4-98, 84 hp) automotive engine (Ref. 50).

The development program on the 4-215 engine has resulted in a number of technical accomplishments; however, there are still some serious deficiencies. Some major innovative accomplishments are as follows:

- 50% reduction in specific weight when compared with previous Stirling engines.

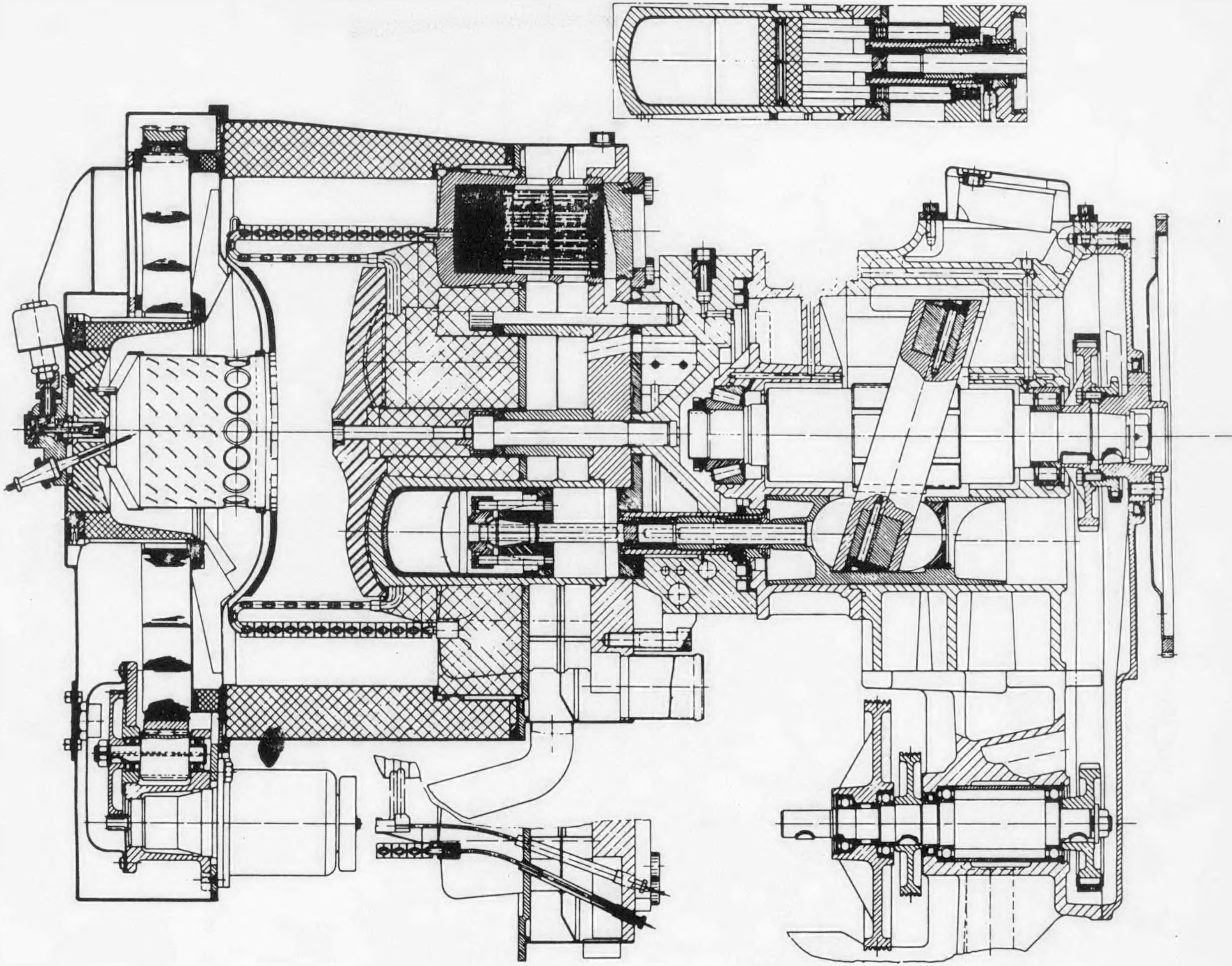


Figure 4-18 Cross-section of the 4-215 D.A. Engine (courtesy Philips)

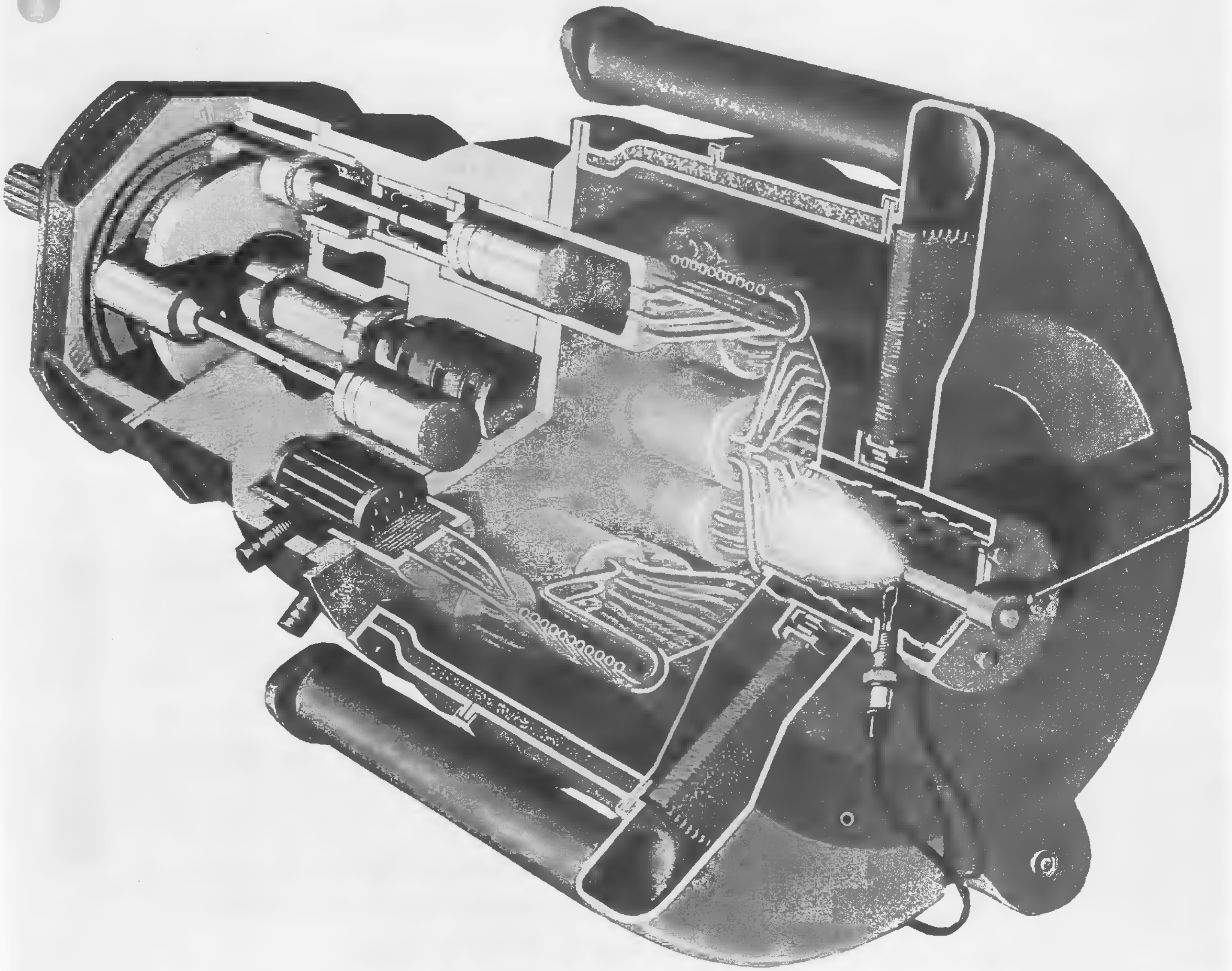


Figure 4-19 Cutaway Pictorial View of Ford-Philips Swashplate Engine

- 200 atmospheres H_2 working gas pressure versus 150 atmospheres for previous engines.

- Improved air/fuel ratio control system (for heater head temperature control) to satisfy vehicle dynamic performance requirements.

- Improved power control system to satisfy automotive demands.

Some of the deficiencies of the automotive system at its present state of development, as expressed by Ford at the recent IECEC meeting in Washington (August 1977), and at the Contractors Coordination Meeting in Ann Arbor (October 1977) are as follows:

- Demonstrated 20% lower fuel economy than baseline gasoline powered 1977 LTD II vehicle (12.6 mpg versus 15.5 mpg).

- Stirling vehicle measured emissions (particularly NO_x and CO) are substantially higher than predicted.

- Start up time is significantly higher than predicted (24 seconds measured versus 15 seconds objective for key-on to drive away).

- Vehicle weight is 106 lbs over baseline LTD II.

- High cost and reliability concerns.

In recent years, Philips have demonstrated that the Stirling engine can be operated on solid fuels and coal derived fuels. They have operated a Stirling engine on fluidized bed coal combustion and coal derived liquid fuels produced in the United States.

The Ford-Philips team has recently executed a contract with the U.S. Department of Energy (formerly ERDA) for a $160 million cost sharing type development program extending over the next 8 years. Details of the objectives and schedule for this program was revealed at a recent DOE Contractor Coordination Meeting in Dearborn, Michigan (Ref. 51). Some of the major objectives are:

1. Near term objective is to develop a Stirling engine which exhibits a 30% fuel economy advantage over a comparable 1976 spark ignition engine and can meet the statutory emission requirements of the Clean Air Act as amended.

2. Long term objective is to develop an advanced engine which will exhibit a 60% fuel economy advantage over a comparable spark ignition engine, and will meet or better the statutory Clean Air Act emission requirements.

4.4.3.2 KB United Stirling (Sweden) AB & Company

Since 1968, KB United Stirling (USS) has been pursuing Stirling engine development with a goal of adapting it for commercial production - "to realize its potential as a reliable, economical and environmentally acceptable means of converting energy". In 1972, USS decided to concentrate their work exclusively on the double-acting engine because it is simpler, more compact and less expensive, and it can be adapted to various types of drive systems (Ref. 38). At that time, USS conceived, designed and built the first double-acting "V" engine having

4 cylinders, a single conventional crankshaft, as well as short interconnecting ducts between adjacent cylinders. This engine has proven to be a very useful building block from which to investigate new heater designs, sealing methods, control systems, combustion systems and accessories. It has undergone extensive laboratory testing and has been demonstrated in 2 passenger cars; shown in Figure 4-20, is a double-acting V-4 engine installed in a Ford Pinto. In October 1977, road testing began on a 100 hp, V-4 Stirling engine installed in a truck (Figure 4-21).

Recently, USS decided to develop a new line of engines directed towards production of these engines for mining equipment, total energy systems and road vehicles. A dual crankshaft engine (or U-engine) was chosen as opposed to a V-engine, as the best choice for production from manufacturing cost considerations. Shown in Figure 4-22 is a photograph of the P-40 (40 kW) engine without air preheater. Fabrication of the new P-75 (76 kW) engine is due to be completed early in 1978 after which testing will begin.

The development work at USS has followed a somewhat different path than that of the Ford-Philips group, with complimentary achievements in Stirling technology. Some of the accomplishments are as follows:

- A piston rod sliding seal (as opposed to roll sock seal) life in excess of 4000 hours.
- Demonstration of the utility and practicality of a conventional engine configuration for the Stirling engine.
- Measured peak engine efficiency of 37%.

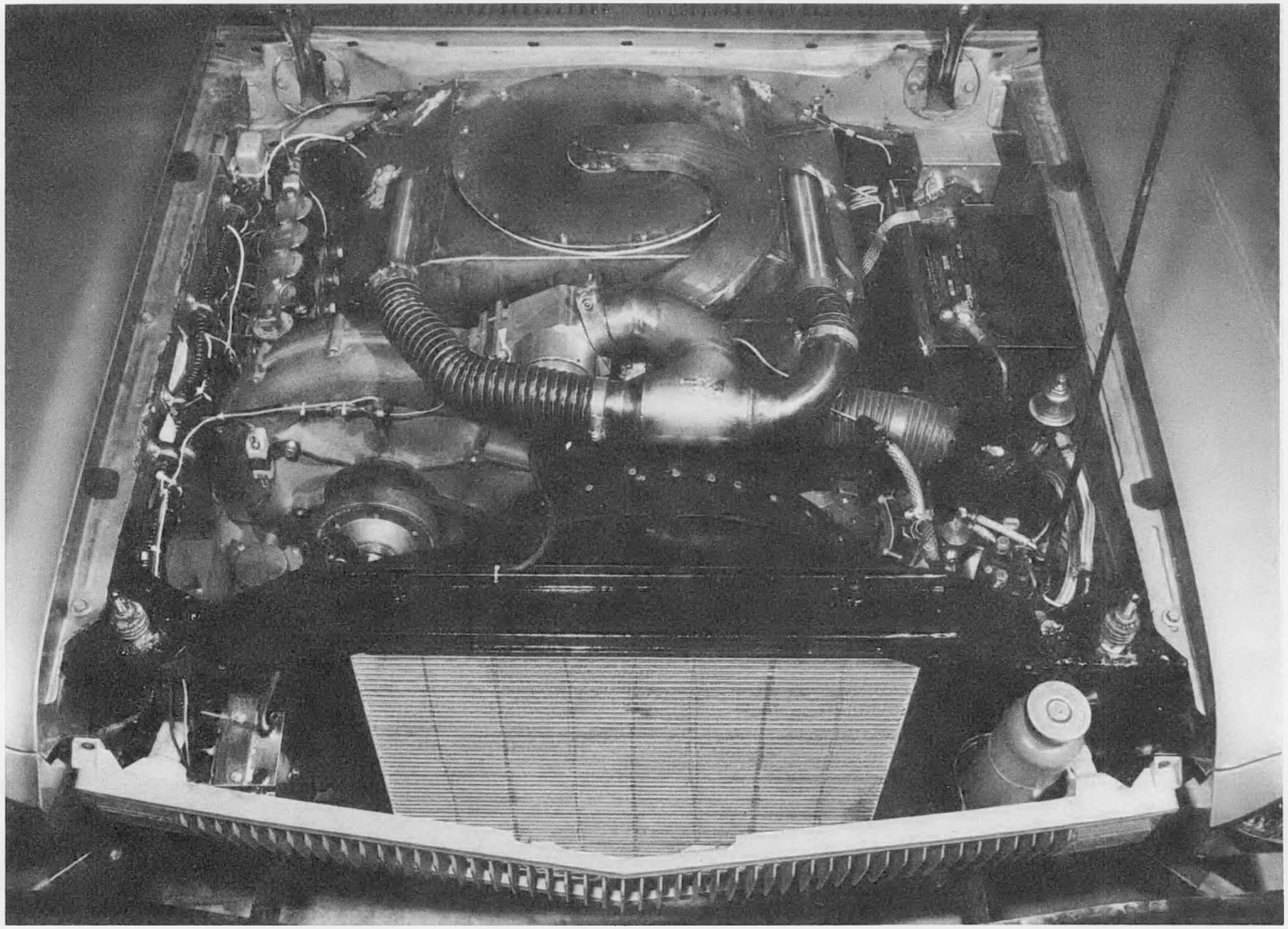


Figure 4-20 Experimental Double Acting V-4 Engine Installed in a Ford Pinto.

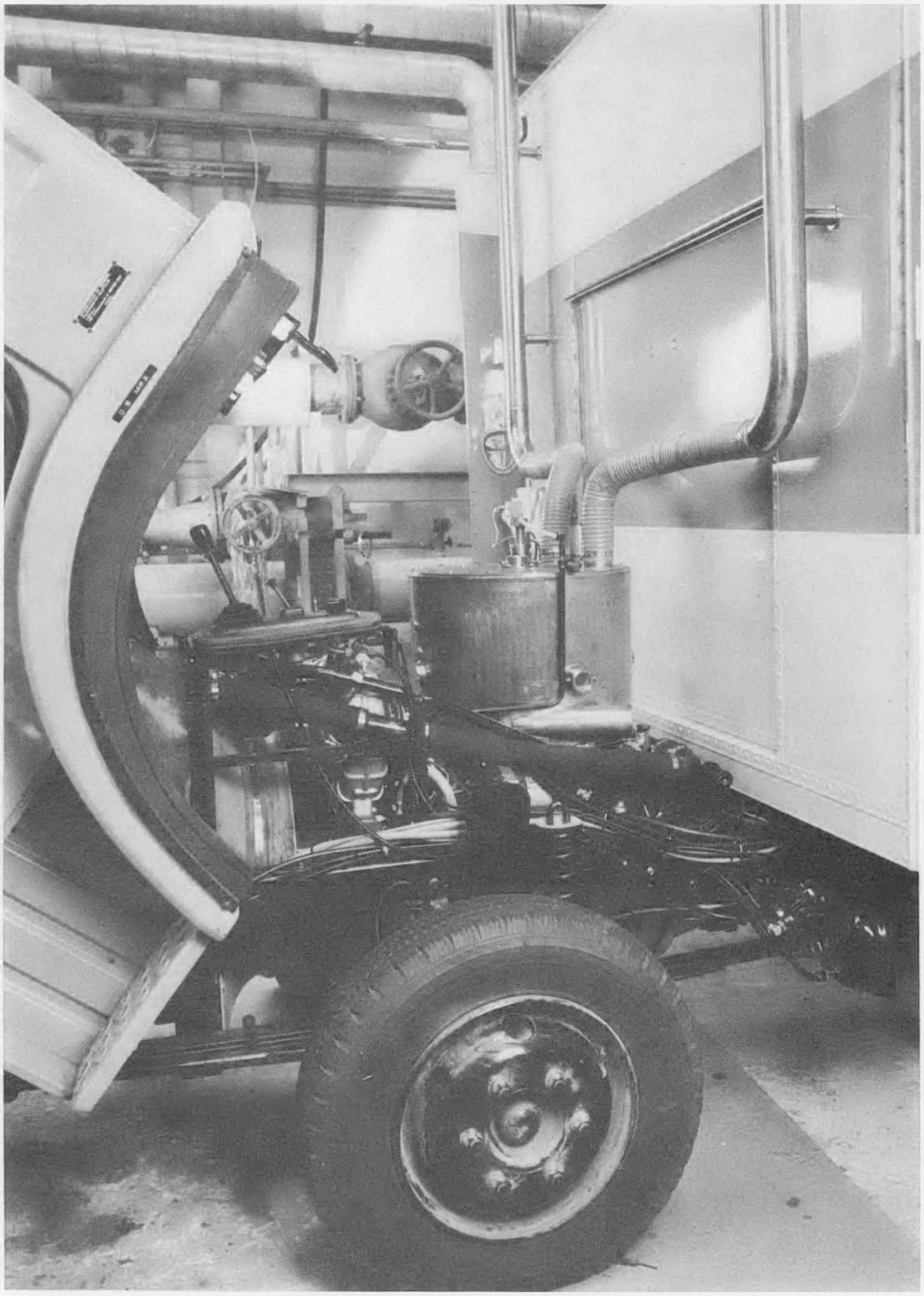


Figure 4-21 A 75kW Stirling Engine Installed in a Truck
(courtesy United Stirling)

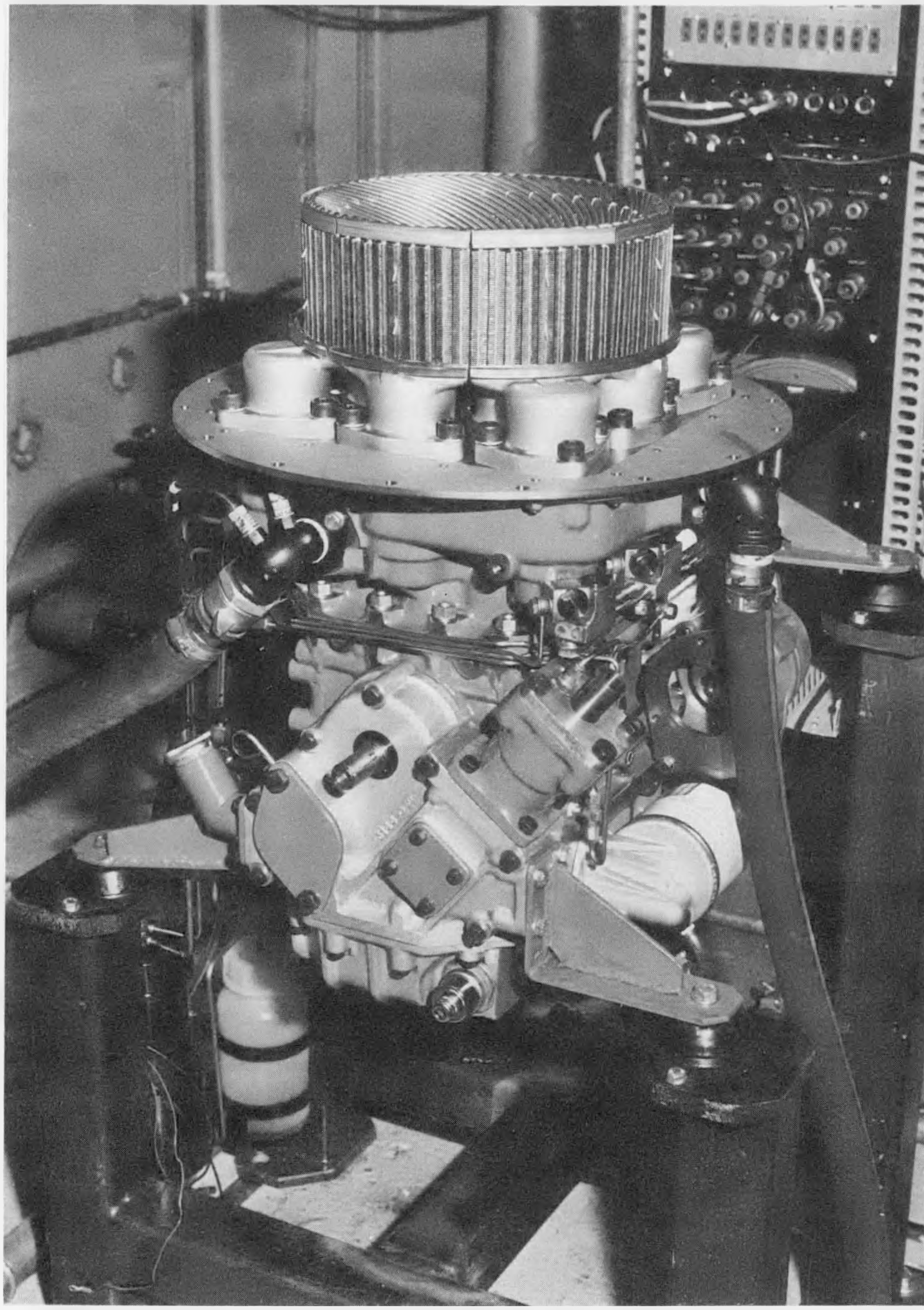


Figure 4-22 A 40 kW Stirling Engine (P-40) With Combustor, Preheater, and Insulation Removed (courtesy USS)

- Good power control and response in an automotive vehicle.

- Improved lower cost heater head design.

They continue to undertake development work aimed at producing lower cost and producible components with improved life. Areas which are receiving further development attention are:

- Improved combustor design for reduced emissions.

- Lower cost material/design heater heads.

- Improved sliding seal life (8000 hour goal).

Preliminary steps have been taken to introduce a Stirling engine to the market. United Stirlings' parent firm FFV are preparing to tool up for manufacturing a small (8 kW) Stirling engine driven generator set for mobile homes. Field tests of several units are currently underway and are set up by a new marketing firm called Stirling Power Systems Corporation located in Ann Arbor, Michigan.

In October of 1977, the NASA-Lewis Research Center announced that they were negotiating with a team consisting of United Stirling of Sweden, American Motors and Mechanical Technology, Inc., Latham, New York for a development program similar to the Ford-Philips contract in content and size. The goals of the program are to establish a data base of technical and economics information to aid the automobile industry in making a decision to put the Stirling in production.

4.4.3.3 Other Development Activities

In addition to the development activities described above, numerous companies, universities and government laboratories are engaged in various aspects of Stirling engine development. The following is a list of the organizations that the authors believe are active in the field:

Companies

1. Aerojet Liquid Rocket Co., Sacramento, CA - (J. Moise)
2. Alberta Gas Trunkline, Calgary, Alberta, Canada - (M. Benza)
3. Daimler-Benz, Stuttgart, West Germany
4. D-Cycle, Inc., Richmond, VA - (G. Davoud)
5. ERG, Inc., Oakland, CA - (G. M. Benson)
6. FFV, Eskilstvna, Sweden - (Parent of United Stirling)
7. Fairchild Industries, Germantown, MD - (A. Schock)
8. Ford Motor Co., Dearborn, MI - (N. D. Postma)
9. General Electric Co., Valley Forge, PA - (R. Tharpe)
10. General Electric Research Laboratories, Schenectady, NY
11. Josam Mfg., Co., Michigan City, IN - (N. E. Polster)
12. Kinergetics, Inc., Tarzana, CA - (K. Cowans)
13. M.A.N.-MWM, Augsburg, West Germany - (F. A. Zacharias)
14. Mechanical Technology, Inc., Latham, NY - (B. Goldwater)
15. N.A. Philips, Briarcliff Manor, NY - (A. Daniels)
16. N.V. Philips, Eindhoven, Netherlands - (C. L. Spigt)
17. Richland Energy Laboratories of McDonnell Douglas, Richland, WA - (R. P. Johnston)

18. Stirling Power Systems Corp., Ann Arbor, MI - (L. Johansson)

19. Sunpower, Inc., Athens, OH - (W. T. Beale)

20. Trans Computer Associates, Los Angeles, CA - (T. Finklestein)

21. United Stirling (Sweden) AB & Co., Malmo, Sweden - (B. Hallare)

Universities

1. Kings College, London, UK - (A. Organ)
2. Massachusetts Institute of Technology, Cambridge, MA - (J. L. Smith, Jr.)
3. Technical University of Denmark, Lyngby, Denmark - (B. Qvale)
4. University of Calgary, Alberta, Canada - (G. Walker)
5. University of Reading, Reading, UK - (P. Dunn)
6. Univeristy of Washington, Richland, WA - (W. R. Martini)
7. University of Witwatersrand, S. Africa - (C. J. Rallis)

Government Laboratories

1. Argonne National Laboratories, Argonne, IL - (R. E. Holtz)
2. Jet Propulsion Laboratory, Pasadena, CA - (F. Hoehn)
3. NASA LERC, Cleveland, OH - (M. Krasner)
4. National Bureau of Standards, Gaithersburg, MD - (D. A. Didion)
5. U.K.A.E.R.E., Harwell, UK - (E. Cooke-Yarborough)

The organizations listed above have approximately 500 workers in private companies, plus approximately 75 workers in universities and government laboratories working in the Stirling field. Besides that work which has already been described in previous sections, brief descriptions of the activities of some of these organizations are described below.

For further information, the reader is referred to the listing in Walker's book (Ref. 36) and to a compendium of current Stirling activities to be published shortly by Professor W. R. Martini of the Joint Center for Graduate Study, Richland, Washington, under a Department of Energy grant.

General Electric Company

Two groups within General Electric, one in Valley Forge, Pennsylvania, and one in Schenectady, New York, are working in the Stirling area. One project sponsored by the American Gas Association is to develop a heat-actuated heat-pump using a free-piston, free-displacer Stirling engine. Another project is a Stirling radio-isotope power system using plutonium-238 oxide as the fuel, which is being done jointly with N.A. Philips. The Schenectady group is working on ceramic heater heads, and reportedly plans to operate a small rhombic drive engine on helium at 1300°C using a silicon-carbide heater head. G.E. was recently awarded a DOE contract to develop a Stirling "universal test engine" in conjunction with Philips.

M.A.N. - MWM

M.A.N. began working on Stirling engines in 1968 under license from Philips. Very little information is known about their recent developments. They had been working on Stirling engines heated by chemical combustion (Li-SF_6), and also on double-acting, heavy-duty automotive engines. Some of their work is believed to be for military applications.

Mechanical Technology, Inc.

MTI has been working on free-piston (Beale-type) Stirling engines for both solar and radio-isotope power generation. They are estimated to have approximately 20 people working on Stirling projects, and are believed to be expanding their capability in preparation for a major automotive Stirling development program in collaboration with United Stirling (Sweden).

North American Philips Corporation

N.A. Philips is the U.S. subsidiary of NV Philips. They are investigating total energy applications for Stirling engines under a DOE contract, besides working with General Electric on the Stirling radioisotope power system. In recent years they have engaged in many specialty Stirling engine projects in collaboration with NV Philips.

Sunpower, Inc.

Sunpower, Inc., was founded by Professor William T. Beale of Ohio University, Athens, Ohio. They have done a great deal of work on free-piston type Stirling engines for both electrical power generation, heat-pumps, and air-conditioners. Much of their work was funded by the American Gas Association to develop a free-piston Stirling engine-driven air-conditioner using an inertial compressor. It is believed that Sunpower is no longer doing this work, but instead is supporting General Electric in a DOE/AGA-sponsored program with a similar objective.

Trans-Computer Associates

TCA is headed by Dr. Theodor Finkelstein, who has been active in the Stirling work for many years. TCA's primary contributions have been in the area of developing computer programs for design and analysis of Stirling engines.

The foregoing listings and summaries are intended to give a sample of some of the additional efforts in the Stirling engine development field. While these additional development activities involve the efforts of several hundred investigators, the reader should not be misled into over-estimating the engine development capability in this country. With the possible exception of Ford Motor Company, who has relied on Philips for their Stirling engine know-how, no U.S. company has an engine development program to match the scope and experience of the European companies (Philips, United Stirling, M.A.N.-MWM), all of whom are Philips licensees.

4.5 Comparison of Automotive Versus Industrial Application Requirements and Engine Characteristics

In recent years, the primary thrust of Stirling engine development has been towards automotive applications. While much of the progress in automotive Stirling engines applies as well to industrial engines, it must be emphasized that there are significant differences between the requirements for automotive Stirling engines versus industrial engines. Similarly, success or failure of the Stirling engine in one area of application may not hold true for the other. Since this assessment relies on automotive Stirling engines as examples of modern Stirling development, this section considers some of the important similarities and differences so as to put extrapolations between the two into the proper perspective. A summary of the desirable engine characteristics for automotive and industrial applications is given in Table 4-2.

Engine Size

Typical automotive engine power levels are in the range of 50-200 hp, versus the 500-2000 hp industrial engine power levels of interest here. This lower power level typically results in lower specific weight and specific volume for automotive engines due to the scaling laws (see Section 5.2.2). Thus, most examples of automotive Stirling engines fall into the specific weight and volume class of small high-speed Diesel engines, or, in the case of the Ford/Philips passenger automobile engine, even into the range of light-duty spark-ignition engines. Furthermore, because of the importance of compactness in automotive engines relative to industrial engines, the design trade-off for automotive Stirling engines is weighted in favor of light weight and small volume to the detriment of efficiency. A similar trade-off is made in high-speed automotive Diesels in comparison to large, stationary, industrial Diesel engines.

TABLE 4-2

COMPARISON OF APPLICATION REQUIREMENTS

	Automotive	Stationary
Power Rating	50 - 200 HP	500 - 2000 HP
Compactness	Very Important	Not Important
Low Weight	Very Important	Not Important
Efficiency	Part Load Efficiency Is Important	High Load Efficiency Very Important
Heat Rejection Load	Important	Secondary Importance
Required Rotational Speed	About 2500 - 5000 RPM	500 - 1000 RPM-Usually Constant
Required Response Time	Fractional Second	Fractional Second to Few Seconds
Noise	Important	Secondary Importance
Low Emissions	Important	Not as Critical, but Highly Desirable
Fuels	Distillate	Residual or Solid Fuels Preferred
Durability	About 4000 Hours	Over 15,000 Hours
Cost	Initial Cost Very Important	Life Cycle Cost Very Important

Efficiency

In vehicle applications engine volume and weight are very important, because they have a significant influence on total vehicle size, weight, fuel conomy and cost. High efficiency is more important at part load than at full load since most of the operating time occurs at part load. Hence, automotive engine designs usually strive for compactness and low specific weight (2 to 6 lb/hp) at the expense of full load efficiency so long as reasonably good part load efficiency is attained. These considerations lead to the selection of high-speed, light-weight designs which have inferior efficiency and life as compared to industrial engines.

Stationary engines, on the other hand, are not similarly penalized by weight and volume (except as they influence cost). As a result the more important considerations such as high efficiency and durability/reliability lead to the selection of larger, heavier, slower speed engines that are more conservatively rated for operation near their best efficiency point.

Furthermore, on automotive engines most auxiliaries are engine-driven, usually at fixed speed-ratio, with the usual result that the engine auxiliaries tend to be over-driven at full load, and thus consume excessive power. Since industrial engines often operate at a constant speed, and/or can afford to have their auxiliaries independently drive, auxiliary power consumption tends to be a smaller fraction of gross output, with the result that brake efficiency is higher for a given gross engine efficiency.

Rotational Speeds

Automotive Stirling engines usually involve maximum rotational speeds ranging from 2500 to 5000 rpm. This high speed results principally from the need to minimize engine volume and weight. Maximum piston speeds are pushed close to the design limit (about 900-1200 ft/min) where engine efficiency begins to degrade significantly.

For stationary Stirling engines where high full load efficiency is important, lower piston speeds will be necessary (e.g., 500-600 ft/min). As a result of the lower piston speeds and the larger piston sizes maximum rotational speeds for large stationary engines are expected to fall in the range of 600-1200 rpm. This is similar to the speed range of large stationary Diesel engines.

Heat Rejection

Because Stirling engines must reject all of their waste heat through a heat exchanger, they have a more demanding heat rejection requirement than Diesel or gas turbine engines. Moreover, since the efficiency of the Stirling engine improves substantially as the heat sink temperature is reduced, (Figure 5-12), it becomes important to minimize this temperature. Automotive engines must reject heat to atmospheric air through a radiator. In order to minimize engine cooling water temperature it is necessary to have a large radiator surface and/or expend a substantial amount of power to drive the cooling fan; thus, the heat rejection problem is rather severe. An optimum automotive cooling system operates with a cooling water temperature of about 70°C (160°F), at 30°C ambient.

By contrast, in the stationary Stirling engine the heat rejection problem is considerably simplified. Cooling water may be supplied at substantially lower temperature (30°C) from rivers, lakes, ponds, or a wet cooling tower system. This not only provides about 10-15% higher engine efficiency but also reduces the size and cost of the heat exchangers and reduces the pumping power requirements.

Control Response Requirements

Automotive engines require fast response (order of .1-.2 seconds time constant) due to rapid load changes which occur during typical maneuvers. This type of response has been achieved in automotive Stirling engines. Ordinarily large stationary engines do not require as rapid response to load changes. However, for electric power generation applications where close speed regulation is required, ($\pm$ 1/4 to 1/2%) and sudden load changes occur, similar response characteristics may be required. Small Stirling engine generator sets have already demonstrated this type of response characteristic and it is believed that large engines will also be capable of the required response.

Emissions and Fuels

The exhaust emissions requirements for automotive engines are currently very stringent and are expected to become even more stringent in the future. The interest in the Stirling engine for automotive use is based primarily on its superior emissions characteristics, and its fuel economy potential. Its low noise characteristics are also of particular interest in automotive use. Petroleum based gasoline and distillate fuels are used exclusively for automotive versions due to the ease of handling these fuels and to the stringent exhaust emissions regulations.

In stationary engine applications the exhaust emissions requirements are less severe. Distillate fueled Stirling engines easily meet the requirements, as do current Diesel and gas turbine engines. Stirling engines burning solid fuels and residual fuels are also expected to meet the current and expected statutory emissions regulations, although with some fuels, sulfur oxides and particulates may become a problem. Although noise is not as critical in the stationary application, the low noise characteristic of the Stirling engine will be a welcome bonus.

Cost

Since Stirling engines have never been mass produced there is considerable uncertainty regarding comparative costs. One study (Ref. 58) concluded that a 150 hp automotive Stirling engine would cost $1610 compared with $1320, and $1490 for equivalent spark-ignition and Diesel engines. This is about $10/hp. Large stationary Diesel engines typically cost from $100-150/hp. It is the opinion of the authors that the large stationary Stirling engines will cost about 25-50% more than their Diesel counterparts when operated on liquid fuels and 50-80% more when operated on solid fuels.

Durability

There is a substantial difference between the durability requirements of light-duty automotive engines versus heavy-duty industrial engines; the former typically requiring a useful life of up to 4000-5000 hours, while the latter application seeks three times this amount between major overhauls. In this regard however, even automotive Stirling engines appear to have excellent durability characteristics.

4.6 Comparison of Stationary Engines

In comparing a future industrial Stirling engine against existing industrial engines, one is faced with the problem of comparing a hypothetical Stirling engine whose characteristics have been extrapolated from smaller automotive research engines, to existing production engines which have had the advantage of substantially more technological development. While the authors have attempted to make the comparisons neither overly conservative nor overly optimistic, the possibility of some inherent bias in the comparisons must not be overlooked.

For purposes of comparison, we have selected the reference application to be either an industrial electric generator set or an industrial prime mover (traction, pumping, etc.). The current reciprocating engine is represented by a medium-speed, 4-cycle turbocharged and intercooled Diesel engine*, either direct or indirect-injected. Representative industrial gas turbines are primarily non-regenerative, gas or distillate fuel-fired. However, some regenerative versions have also been referenced for comparison. It must be borne in mind that most gas turbine characteristics are presented without considering the reduction gear, thus giving the appearance of much lower specific volume and specific weight than in the "as installed" condition.

Representative engine characteristics for industrial Diesel engines, industrial gas turbines and Stirling engines (mainly light-duty and heavy-duty automotive), are shown in Tables 4-3, 4-4 and 4-5, respectively. Emission characteristics are shown separately in Table 4-6. Finally, a synopsis of typical engine characteristics is given in Table 4-7.

Diesel Engines

The Diesel engine is the most efficient prime mover in the relevant power range of 500-2000 horsepower. Turbocharged and intercooled engines reach brake thermal efficiencies approaching 40% based on lower heating value

* This is not to imply that gas and dual-fuel engines are unrepresentative; but the Diesel can be chosen to represent the broad class of reciprocating industrial engines without loss of generality.

TABLE 4-3

INDUSTRIAL DIESEL ENGINES [1]

Manufacturer	Model	Power HP [2]	SFC lb/hp-hr	RPM	Weight lb	Dimensions LxWxH-in	Box Vol ft^3	ft^3/hp	lb/hp	Comments
Alco	8V251G	1,620	0.35	1,000	26,000	139x67x101	544	0.33	16.0	Turbocharged, aftercooled, not incl H-X
Caterpillar	D-399-V-16	1,100	0.38	1,200	16,125	118x60x79	324	.29	16.1	Heat Exc. cooled, not incl H-X
	"	1,100	0.42	1,200	--	164x87x100	826	.75	--	Rad. cooled, with radiator
Deutz (Ger)	BAGM528	1,000	--	1,000	13,670	127x49x92	331	.33	13.7	Direct injected
M.A.N. (Ger)	6ASL25/30	1,470	0.36	1,000	25,340	136x47x91	331	.23	17.2	
English Elec	8RK3C	1,760	0.35	900	27,850	--	--	--	15.8	
	16YJX	1,530	--	1,500	15,300	105x53x81	261	.17	10.0	Deltic
Nohab (Swed)	F28V	1,800	0.35	1,000	23,600	131x71x90	484	0.27	13.1	Turbocharged, aftercooled, not incl H-X
Sulzer (Swiss)	ASL25/30	1,760	0.35	900	31,740	172x48x78	373	0.21	18.0	" " " " "
GMT (Italy)	AL230-8	1,600	0.35	900	20,500	157x38x90	311	0.19	12.8	" " " " "

[1] Diesel and Gas Turbine Worldwide Catalog, 1976, Ed., Vol. 41, published by Diesel and Gas Engine Progress.

[2] Continuous rating with 10% overload permitted for 1 hour.

TABLE 4-4

INDUSTRIAL GAS TURBINES

Manufacturer	Model	Power HP	SFC lb/hp-hr	Power Turbine RPM	Output Shaft RPM	Weight lb	Dimensions LxWxH-in	Box Vol ft^3	ft^3/hp	lb/hp	Comments
AirResearch	831	500	0.73	42,000	1,800	1,250	65x41x34	52	.105	2.5	Engine and Gearbox [1]
	831P	490	0.56	42,000	--	2,600	124x45x47	152	.31	5.3	Inc. exh. recuperator [1]
Avco-Lycoming	TF12A	1,150	0.65	18,500	--	950	51x30x43	39	.033	.87	Engine Only [1]
	TF14B	1,450	0.62	18,500	--	950	51x30x43	39	.027	.67	" " "
	TF25A	2,250	0.63	14,000	--	1,100	50x30x43	37	.017	.49	" " "
	TF35	2,800	0.58	14,000	--	1,150	--	37	.013	.41	" " "
Allison	501K14	1,940kW	0.81 lb/kWh	14,000	--	1,260	86x24x30	36	.019	.65	Engine only [1]
	GT-404	325	--	--	2,873	1,700	47x28x40	31	.094	5.2	Automotive, Regenerative [1]
Rolls-Royce	GN1201	1,020	0.65	19,000	--	353	73x13x21	11.5	.011	.35	Engine Only [1]
Ruston	TA1750G	1,740	0.83	--	1200-1800	14,500	188x88x84	804	.46	8.3	Engine,Gearbox,Base,Control [1]
Solar	T1001S	1,160	0.63	22,300	--	1,250	70x45x44	80	.069	1.08	Not aircraft deriv.,eng.only [1]
	Saturn	1,200	--	--	--	8,230	122x70x76	376	.313	6.9	Mechanical drive package [2]
Ford	3600 Ser.	450	0.42	--	3,000	1,700	41x36x42	36	.08	3.8	Regen G.T., Automotive [2]
United Airc. of Canada	ST6J-70	510	0.67	--	2,200	350	62x19x19	13	.025	.7	Simp.cyc.,free turb.,inc.gearbox
	ST6T-75	1,300	0.65	--	6,600	730	66x44x32	54	.041	.56	" " " " " " [2]
Waukesha	T400	400	--	25,270	2400-3600	695	57x28x34	32	.08	1.7	GT only with gearbox [2]
	T400	400	--	25,270	2400-3600	1,400	58x40x46	62	.15	3.5	Complete drive module [2]

[1] Gas Turbine International, Vol. 14, No. 3, May-June, 1973.

[2] Diesel and Turbine Worldwide Catalog, 1972, Ed., Vol. 37, published by Diesel and Gas Engine Process.

TABLE 4-5

STIRLING ENGINE SPECIFICATIONS

Manufacturer	Model	Power	SFC lb/hp-hr	RPM	Weight lb	Dimensions LxWxH-in	Box Vol ft^3	ft^3/hp	lb/hp	Drive/Type	No. Cyl	Working Fluid	Comments
Philips	4-1385	400	.37-.42	1,300	2,870	55x35x36	39	.097	7.2	Swashplate	4	H_2	Bare engine
	8-500	400	--	1,900	--	44x27x42	29	.072	--	Swashplate	8	H_2	Bare engine (24)
	4-235	200	.46	3,000	1,670	49x20x43	25	.127	8.4	Rhombic	4	H_e	Bus engine, bare
	4-215	170	.43-.57	4,000	750	--		--	4.4	Swashplate	4	H_2	Incl all aux (Ref. JPL)
	--	275	.32-.46	1,600	--	59x27x51	48	.173	--	Rhombic	4	H_e	Bus eng incl all aux (14)
United Stirling	P-40	55	.4	4,000	396	20x20x31	7.2	.13	7.2	D.A.,Twin Crank	4	H_2	Fan-to-flywheel
	P-75	100	.37	2,400	770	36x23x39	18.7	.187	7.7	D.A., V-4	4	H_2	" " "
	P-150	200	.37	2,400	1,430	--	--	--	7.2	D.A., V-4	8	H_2	" " "
Representative Design	--	2,000	.34	850	32,000	143x50x68	281	.14	16	D.A.,Twin crank	16	H_e	Incl. indirect heat transfer system, but not combustor (liquid or solid fuel)

TABLE 4-6

TYPICAL ENGINE EMISSIONS

	Grams/hp-hr NO_x	CO	HC
A. PRODUCTION ENGINES			
Diesel:			
4-Cycle, Direct Injected (57)	7-20	1-10	1-10
4-Cycle, Precomb. Chamber (57)	4-7	1-2.5	.05-6
2-Cycle, Direct Injected (57)	15	2.5-6	.8-1.2
4-Cycle, Direct Inj., Comprex Supercharged (56)	3.5	--	.7
4-Cycle, Precomb. Chamber, Nat. Asp. (60)	3.2	2.7	1
4-Cycle, Precomb. Chamber, Turbocharged (60)	3.5-4.2	1.7-2.2	.7
4-Cycle, Direct Inj. Nat. Asp. (60)	3.5	6	3.5
Gas Turbine:			
Non Regen., Gas-Fired (57)	1-3	--	--
B. R&D ENGINES			
Stirling:			
United Stirling (37)	NO_x+UHC - 2.1	1.1	--
Philips (24)	NO_x+UHC - .53	2.2	--
Philips (16)	.4-.7	.4-1.1	.01-.02
with 25% EGR	.15	--	--
Gas Turbine:			
Regenerative, Oil Fired [1]	.6-5	3-6	.3-.6
Regenerative, Oil Fired [2]	6	.6	.06
Non Regen., Oil Fired [3]	2	.4	.2
Diesel:			
Variable Throat Prechamber (59)	2-3.5	--	--

[1] Derived from Ref. 58, pp 5-26, 5-27, assuming BSFC = .65 lb/hp-hr.

[2] Derived from Ref. 39, p 2, assuming BSFC = .65 lb/hp-hr.

TABLE 4-7

SUMMARY OF 500-2000 HP INDUSTRIAL ENGINE CHARACTERISTICS

Characteristic	Stirling	Diesel	Non-Regenerative Gas Turbine
Engine Speed - RPM	800-1200	900-1500	12,000-40,000
SFC - lb/hp-hr	.35-.4	.35-.4	.6-.7
Weight - lb/hp			
Engine Only	13-18	10-18	.3-1.0
With Gearbox	--	--	.6-8.0
Volume - ft^3/hp			
Engine Only	.2-.3	.2-.3	.01-.07
With Radiator/Gearbox	.3-.7	.3-.7	.03-.5
Emissions g/hp-hr			
NO_x	.5-2.0	2-10	.5-10
CO	.5-2.0	1-5	.5-10
UHC	.01-.05	.1-10	.05-.5

of the fuel. Part load efficiency is excellent, with many engines maintaining 35% efficiency even at one-third load.

The specific weight ranges from about 10 lb/hp to about 18 lb/hp (fan-to-flywheel, less radiator), with the lighter weights being typical of the smaller, higher speed, and turbocharged versions, and the heavier weights being typical of larger, lower speed, and naturally-aspirated versions. High-speed Diesels are about half as heavy, but are not considered "industrial", since they do not have the life characteristics required. Fuel economy ranges from approximately .35 lb/hp-hr to approximately .42 lb/hp-hr, generally with the heavier engines showing better efficiency, although there are exceptions. Engine speed ranges from approximately 900 rpm up to 1500 rpm, again with the lower speed engines tending to be more efficient.

Diesel engine emissions are characterized by relatively high oxides of nitrogen due to the high temperature and pressure of the combustion process. Indirect-injected Diesels tend to have lower NO_x than direct-injected versions due to the rich primary combustion conditions. Both indirect-injection and exhaust gas recirculation reduce NO_x emissions; however, both have an adverse effect on fuel economy. Carbon monoxide and hydrocarbon emissions tend to be reasonably low, although there are many exceptions. Relative to the Federal Emissions Standards for heavy-duty engines of 5 g/hp-hr of NO_x + HC and 25 g/hp-hr of CO, it is clear that there is a potential problem only with NO_x. However, many existing engines already meet the NO_x standard, and although some sacrifice in performance may be necessary, Federal Emissions Standards should not inhibit the popularity of heavy-duty Diesel engines.

Gas Turbines

Although gas turbines are significantly less efficient than Diesels, their light weight and low volume, ease of installation, high reliability, and lower first cost enable them to compete with Diesels, especially in instances where weight and volume are important, and in low load factor applications where fuel economy becomes less important. On the other hand, operation and maintenance of gas turbines requires more highly skilled personnel than for Diesel engines, and with an efficiency about half as great as Diesel engines, the operation of gas turbines is considerably more expensive than for Diesels, especially in base load situations.

There has been considerable interest recently in combined cycle gas turbines in which a Rankine bottoming cycle is used to boost overall plant efficiency. However, it is the opinion of the authors that this option is not practical for engines in the size range of interest. On the other hand, in applications that have a need for high-temperature process heat (or high-pressure steam), waste-heat recovery may be a viable option.

Table 4-7 shows that non-regenerative industrial gas turbines have specific fuel consumptions about 1.5-2 times that of Diesels. Specific volume and specific weight for bare engines are as much as an order of magnitude lower than for Diesels, which explains in part the lower cost and ease of installation. It must be pointed out, however, that except where noted, the specific weight and volume values are for the bare engine, and do not include the gear-box, engine base, or controls. After these are included, the specific weight and volume of most gas turbines fall into the lower range of Diesel engines.

Table 4-4 also shows some regenerative gas turbines, including automotive versions. These are seen to have better fuel economy, but higher weight and volume. Roughly speaking, these engines fall into the weight and volume class of automotive Diesel engines, but have poorer fuel economy.

The emission values shown in Table 4-6 for gas turbines show that emissions typically are half as high as for Diesel engines, with some examples in about the same range. Due to their higher combustor inlet temperature, regenerative gas turbines emit about twice as much NO_x as non-regenerative engines. Similarly, it is found that oil-fired gas turbines produce about twice as much NO_x as gas-fired engines (Ref. 57).

Because of the recent interest in automotive gas turbines, there has been considerable activity to reduce NO_x emissions down to the levels required for light-duty vehicles (of the order of 1 g/hp-hr). To this end, several low NO_x homogeneous and variable-area gas turbine combustors have been developed which appear to meet the lower NO_x values, as well as lower CO and HC (Refs. 39, 40, 41, 42, 43, and 58). Likewise, there is considerable current interest in higher temperature ceramic gas turbines which promise better fuel economy. However, this technology is still in its early stages of development, and is not believed to be ready for commercial introduction for at least the next decade.

Stirling Engines

The characteristics of some light and heavy-duty automotive Stirling engines are shown in Table 4-5. Specific fuel consumption is slightly higher than the average of Diesel engines, but about the same for Diesel engines of similar speed and specific power. Specific weight and volume are lower than for industrial Diesels, and fall into the range of high-speed Diesel engines of the same power levels. As with Diesel engines, higher-speed, more compact Stirling engines tend to be less efficient than larger, slower engines. Also included in Table 4-5 are the specifications for the preliminary design of a "representative" 2000 hp stationary Stirling engine (as described in Section 5.2.3). This is a large, slow speed engine that is comparable in size, weight and efficiency to corresponding industrial Diesel engines.

It is difficult to make an unbiased comparison of emissions from a Stirling engine relative to Diesels and gas turbines, since the Stirling emissions data comes entirely from research engines in a laboratory environment. Table 4-6 shows that, in general, Stirling emissions of all species are lower than for Diesels and gas tubines. While it is possible to take issue with the basis of comparison, it should be clear that the combustion conditions in the Stirling engine are inherently more conducive to low pollutant formation, since the combustion products do not participate in the work-producing process, and, thus, need not be compromised in favor of engine performance. The one exception to this characteristic is that in a direct-fired Stirling engine there is an incentive to operate at high combustion chamber temperatures in order to minimize the size of the heater head, and steps to reduce NO_x by lowering combustion chamber temperatures (as by increased excess air or the use of exhaust gas recirculation) tend to increase the size of the heater head, and thus adversely affect engine power and efficiency.

Summary Engine Comparison

The preceeding discussion has cited some specific examples of Stirling, Diesel and gas turbine engines, and has attempted to characterize their salient features (size, weight, speed and efficiency) relative to their power level and intended application. While these examples permit an objective evaluation of engine characteristics, they also contain an inherent bias due to the differences in power levels and in the intended applications (e.g., automotive Stirlings versus industrial Diesels). In Table 4-7, a subjective comparison is presented which attempts to factor out biases resulting from intended application, power level, state of development, etc. This comparison leads to the following general conclusions:

1. An industrial Stirling engine should provide efficiency, specific weight and specific volume about equal to that of industrial Diesel engines, and should operate at about the same speeds. As with Diesel engines, there is a trade-off between efficiency and size; industrial Stirlings will favor higher efficiency at lower speed and higher weight and volume.

2. In comparison with industrial gas turbines equipped with gear-boxes to produce similar output speeds, Stirling engines have somewhat higher weights and volumes, and substantially better efficiency.

3. In general, Stirling engine emissions are significantly lower than those of Diesels and gas turbines, although "low emission" versions of these engines may approach Stirling engine emissions. However, it is inherently easier to obtain low emissions in a Stirling engine, and unlike its competitors, engine performance need not be compromised to achieve low emissions.

4. While it is difficult to assess relative costs due to differences in production bases, it is believed that the Stirling engine bears a cost premium in the range of 25-50% over comparable Diesel engines when directly fired with distillate or gaseous fuels. There should be an additional premium of similar magnitude for solid-fueled Stirling engines relative to liquid-fueled versions.

5.0 TECHNICAL REQUIREMENTS - 500-2000 HP STIRLING ENGINES FOR STATIONARY POWER APPLICATIONS

The following discussion will outline the technical requirements for stationary Stirling engines by:

1. Establishing the most suitable basic engine configurations.

2. Presenting the anticipated specifications and characteristics of the basic engine, (including expected performance characteristics).

3. Discussing the important characteristics of the external combustion heat source systems, including solid fuel burners.

5.1 Choice of Engine Configurations and Kinematic Designs

This subject is addressed by first discussing the configurations believed to be suitable for the subject application and then focussing on the recommended types.

5.1.1. Types Not Suitable -- Single-Acting Engines

a. One Cylinder

Single cylinder Stirling engines are suitable for a power range of the same order of magnitude as single cylinder internal combustion engines, for example, from a fraction of a horsepower to something under

30 bhp. Commercial, 1 cylinder, 4 stroke cycle I.C. engines usually cut-off at less than 15 bhp because of increasingly severe linear and torsional vibration problems, unless elaborate balancing mechanisms are provided. A single cylinder Stirling with rhombic drive is fully balanced, and the torsional vibration is less than a 1 cylinder I.C. engine because it operates in a 2 stroke double-acting mode.

As a result of scaling studies, described in Section 5.2.2, it appears technically feasible to build single cylinder Stirling engines in the power range considered in this study. However, from the larger viewpoint of economic feasibility and practical engineering considerations of manufacturing and maintenance, such designs would not be recommended. For example, from the scaling rules it is seen that engine weight increases approximately as the cube of the scaling factor, and, therefore, engine weight and volume per bhp increases as the first power of the scaling factor. Also, the cost per bhp will increase with scaling factor; but probably not linearly. Another objection to large single cylinder engines results from greater torque variation compared to multi-cylinder engines. Torsional vibrations may be present which cause serious stress problems in the coupling and electric generator, for example. Even with a large flywheel, engine accessories are adversely affected.

A single cylinder Stirling engine requires 2 pistons and a complex drive mechanism. As a consequence its cost will be higher <u>per cylinder</u> than the multi-cylinder double-acting types, described below, which require only one piston per cylinder and have simpler drive mechanisms.

b. <u>Rhombic, Multi-Cylinder</u>

A multi-cylinder rhombic drive engine can solve the torsional vibration problem of the one cylinder engine, provided at least

4 cylinders are used and the phase relation is set at 90° intervals. Each cylinder still requires 2 pistons, 2 piston rods, 2 seals, etc., just as with the one cylinder engine.

The rhombic drive itself was one of the principal objections to multi-cylinder versions in the past. It required extremely good quality control in manufacturing to avoid alignment problems, as well as requiring too many parts compared to multi-cylinder internal combustion engines. Controlling air-fuel ratio in 4 separate burners also proved difficult. Separate air-preheaters were a problem too with life expectancy generally less than 1000 hours. The authors' see nothing to change the conclusion that a rhombic drive multi-cylinder Stirling engine would be a poor choice for future designs.

c. Rocker-Arm

An experimental rocker-arm drive mechanism for a multi-cylinder Stirling engine is described in Section 4.4.2. Because of cost, bearing and balancing problems such a design would not be recommended for large Stirling engines.

d. Opposed Piston

So-called opposed piston Stirling engines have been constructed, particularly in small sizes under 10 kW. Experimental opposed piston engines have been constructed in a V form with one crankshaft; also in-line with 2 crankshafts, one on each end. They have two pistons per power unit and require sealing rings on both pistons to be loaded by gas forces compared to only one set of rings on displacer engines. This increases the mechanical losses on opposed piston engines which reduces the overall efficiency. This construction is not recommended for large Stirling engines.

e. Single Crankshaft, Displacer Engines

Experimental one cylinder displacer engines have been built utilizing a single three-throw crankshaft, as well as a more complex bell-crank attachment, for driving the displacer piston. In both cases, vibration problems were severe. The cost of a single crankshaft is certainly less than the rhombic, but the design is not recommended unless for some special application it became necessary to construct a multi-cylinder displacer engine.

f. Free Piston Engines

The free piston Stirling is an invention of Professor Beale of the University of Ohio (Ref. 64). Some of its advantages include reduced friction losses compared to a crank type machine, hermetic sealing and the ability to self-start. The largest to date is believed to produce between 2 and 3 kW for the purpose of driving a heat pump compressor. Problems have included metal-to-metal contact at end-stroke during start-up and sudden load changes, as well as vibration and mounting problems; also, so far as we are aware, no machine has operated as a complete self-contained package from fuel input to heat pump output. For electrical generation, the future of the free piston is even less certain. Reciprocating generators have proven less efficient and considerably heavier (and more expensive) than rotating types mainly because surface speeds are low. Mean piston speed for a Stirling is under 1000 feet per minute while linear speed of rotating type electric generators range from 3000 to over 30,000 feet per minute at the rotor surface. It is the opinion of the authors' that free piston Stirling machines will find their place for pumping applications at a power level under 10 kW.

5.1.2 Recommended Configurations -- Double-Acting

The Stirling engine configuration which appears to hold the greatest promise for the power range under consideration, 500 to 2000 hp, is the double-acting type now being developed for passenger cars by the Ford-Philips team and for medium duty commercial automotive service by United Stirling of Sweden. Some of its advantages over the displacer-type engines include one piston per cylinder instead of two, fewer burners and controls for a multi-cylinder design, simpler balancing methods, lower weight and volume -- and subsequently lower cost, it is believed. Double-acting Stirling engines have been configured with cylinders in-line, in a "V" and co-axial. Drive mechanisms have included a single crankshaft for the in-line and "V", double crankshafts for co-axial with 4 cylinders and both "Z" crank and swashplate for 4 or more co-axial cylinders. These are illustrated in Figures 5-1, 5-2 and 5-3.

a. Single Crankshaft Engines

In-line Stirling engines, having 4, 5 or 6 cylinders, are closer to conventional internal combustion engines in their general layout and design details, which might offer some manufacturing advantages. However, since double-acting engines require interconnections between cylinders, an in-line configuration is at a disadvantage compared to a "V" or co-axial design. This is because connecting ducts between the cold spaces of adjacent cylinders ("adjacent" from their phase relationship and not necessarily from the geometric layout) are longer for in-line designs which increases dead space and, possibly, pressure losses, depending on the duct cross sectional area and geometry. Increase in dead space at constant mean pressure reduces power but has little effect on efficiency; higher pressure losses hurt both power and efficiency. As a general rule, in-line double-acting designs of 4 or more cylinders would not be recom-

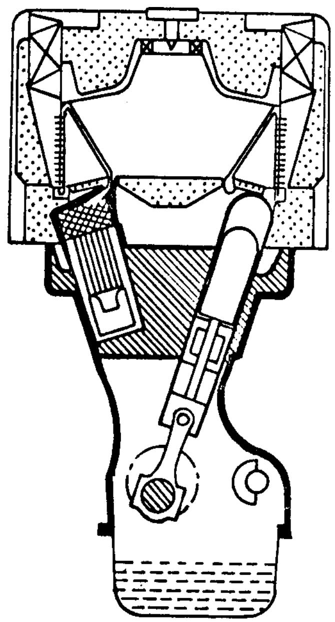


Figure 5-1

Vee Design
Single Crankshaft

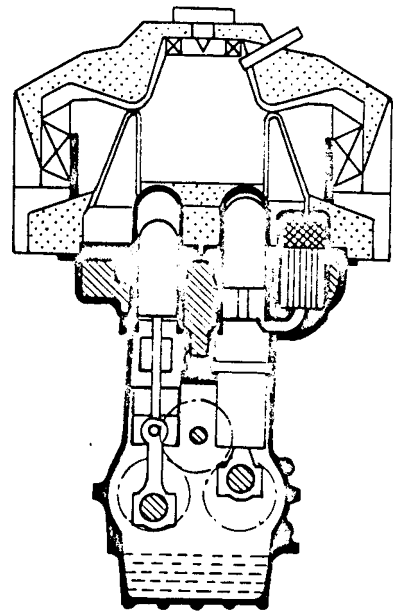


Figure 5-2

Parallel Cylinders
Twin Crankshafts

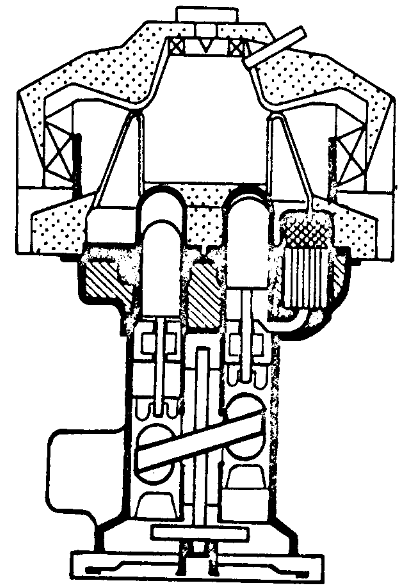


Figure 5-3

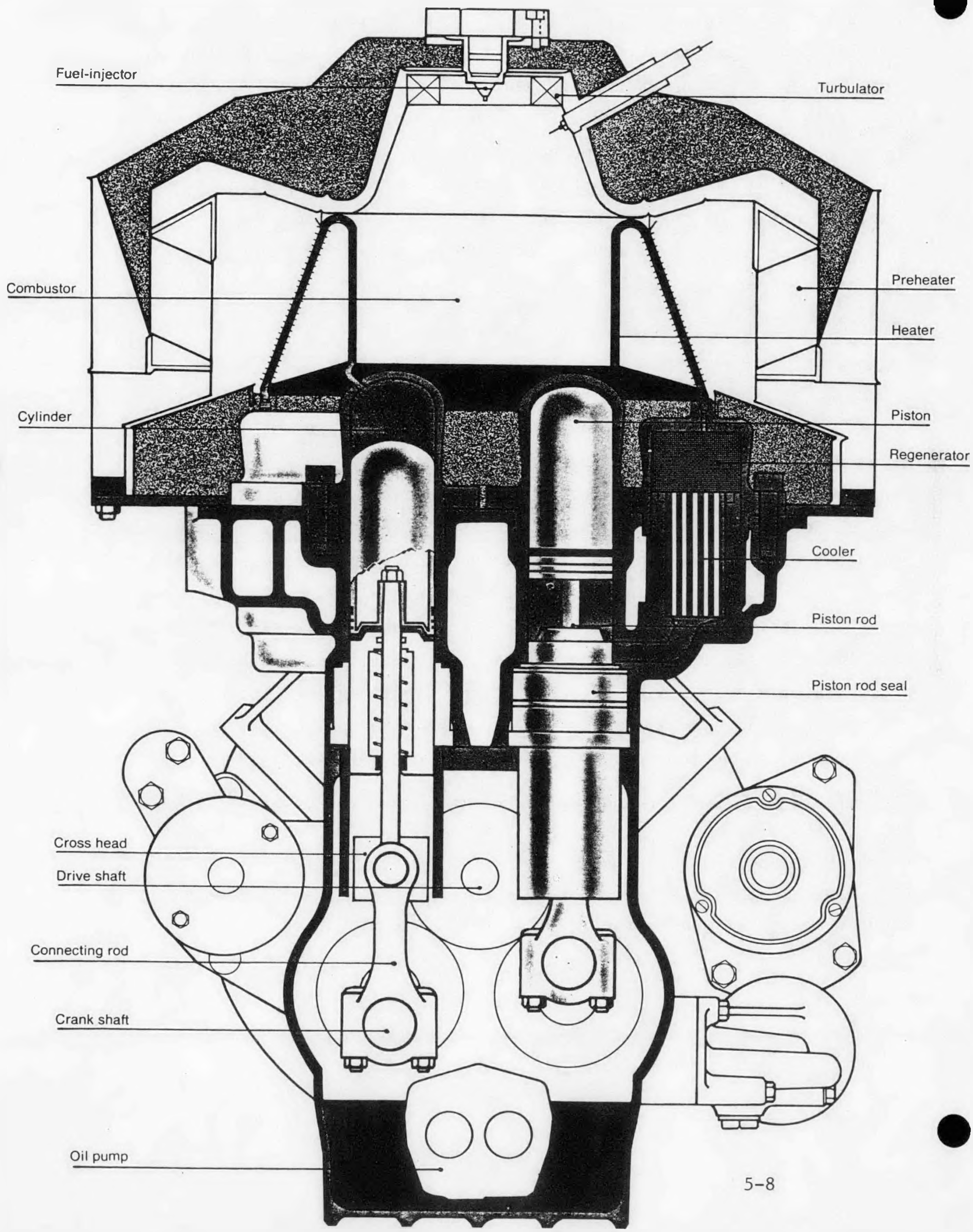
Swashplate
Configuration

mended by the authors. A 6 cylinder "V" with 3 in-line cylinders on each bank, all interconnected, might prove acceptable; also 2 parallel rows of 3 cylinders may be feasible for a 6 cylinder double-acting engine. A double-acting "V" design having 4 cylinders, first conceived and tested by United Stirling, offers the advantage of a single crankshaft as well as short interconnecting ducts. It has proven to be a useful building block from which to investigate new heater designs, control systems, combustion systems and accessories. It has been demonstrated in 2 passenger cars and a truck. This design might prove attractive for stationary power applications either as a 4 cylinder version or 8 or 12 cylinder versions.

b. Twin Crank Engines

United Stirling and others developing new Stirling engines now believe the cylinders should be parallel. United Stirling's latest design consists of 4 cylinders in a square arrangement with twin crankshafts, geared to a central output shaft (Figure 5-4). Parallel cylinders offer 4 main advantages over the V design; (1) easier balancing (and complete balancing rather than partial secondary balancing for the V designs), (2) lower cost in manufacturing with all cylinders parallel, as well as all regenerator-cooler units parallel (V designs necessitated numerous angles to be set for machining), (3) adaptability to other drives, such as the swashplate, (4) more options in number and location of regenerator-cooler units. Twin crankshafts are expected to prove no more expensive to produce than a single crankshaft because they are shorter, more rigid and involve less complex machining. In addition, the twin design lends itself to a more convenient accessory drive layout with 2 output shafts on one end and 3 on the other. Disadvantages include gear losses and possibly higher noise level.

Figure 5-4 Pictorial of United Stirling Twin-Crank Engine



c. Swashplate Engines

The swashplate drive system results in simple harmonic motion and permits more flexibility in choice of the number of cylinders, as for example, a 5 cylinder double-acting engine, which is the most optimum for maximizing efficiency. While there is much experience with swashplate drives for hydraulic and refrigeration pumps, there is much less general engine experience with swashplate bearing systems than with crankshaft bearing systems. It is not yet possible to state precisely which has the advantage in weight, volume and cost. Mechanical efficiency of both systems is believed nearly equal. It is recommended that additional studies be made of the relative merits of cross-head type swashplate drives vis-a-vis "Z" crank or wobble-plate drives using spherical bearing connecting rods.

d. Engine Configurations and Estimated Cylinder Size

What is the optimum cylinder size for the 500-2000 bhp range? This question cannot be answered definitively for a Stirling anymore than it can for a Diesel engine. From cost and maintenance considerations, it is desirable to cover the power range with the fewest possible cylinder sizes. A double-acting engine with 4 or more cylinders is probably desirable over the entire range above 500 bhp from cost considerations. Assuming only multiples of 4 cylinders and a maximum of 16, a cylinder having a nominal output of 125 bhp could cover the range. Generally, in Diesel engine practice there is some small gain as the number of cylinders increase because of improved mechanical efficiency and more efficient accessories (oil and water pumps, fans and blowers, etc.). This results in slightly more BMEP for equal IMEP; or putting it another way, the 16 cylinder engine can be more conservatively loaded and has lower specific fuel consumption than the 4 cylinder engine. The same reasoning is believed to apply to future multi-cylinder Stirling engines.

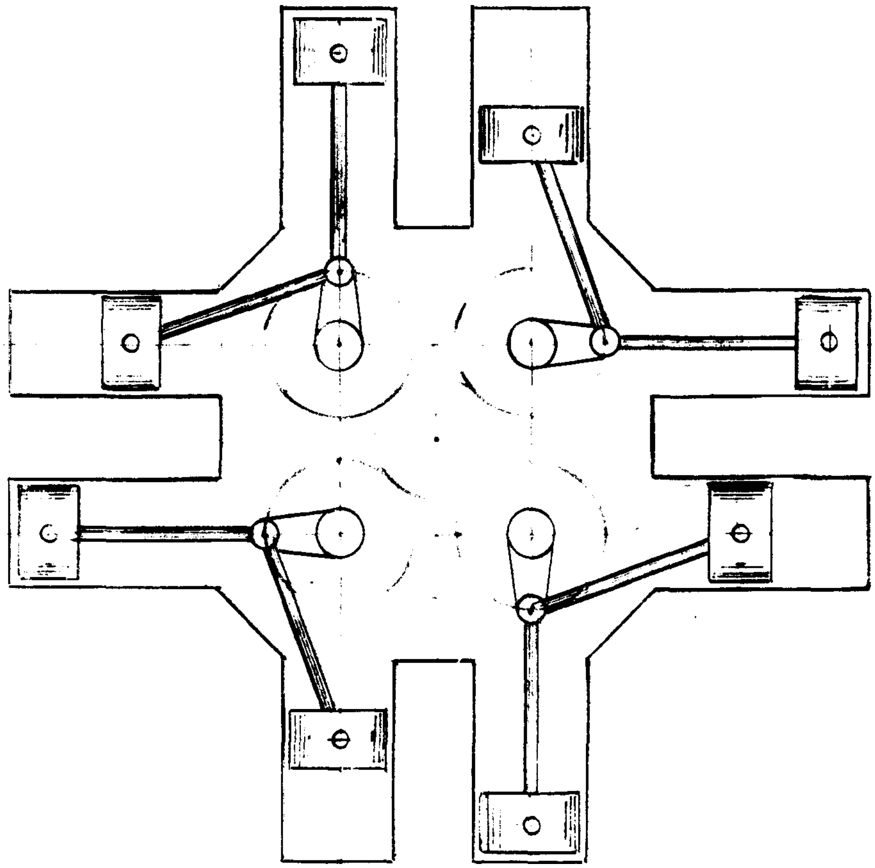
A swashplate or "Z" crank drive could incorporate 4 to 6 cylinders on one side of the drive mechanism; also, the same number of cylinders might be placed on the opposite side giving an engine with 8, 10 or 12 cylinders in a "barrel" or axial arrangement. This latter configuration might prove awkward particularly since the central output shaft can intrude into or otherwise complicate the combustion chamber arrangement.

Another possible configuration incorporates multiples of 4 cylinders in a "square" arrangement with long twin crankshafts, resulting in an in-line engine of 8, 12, 16 or even 20 cylinders. (Twenty cylinders in a "V" arrangement is considered the upper limit in Diesel engine practice because of crankshaft and crankcase design problems, in addition to maintenance problems, as the number of cylinders increase.)

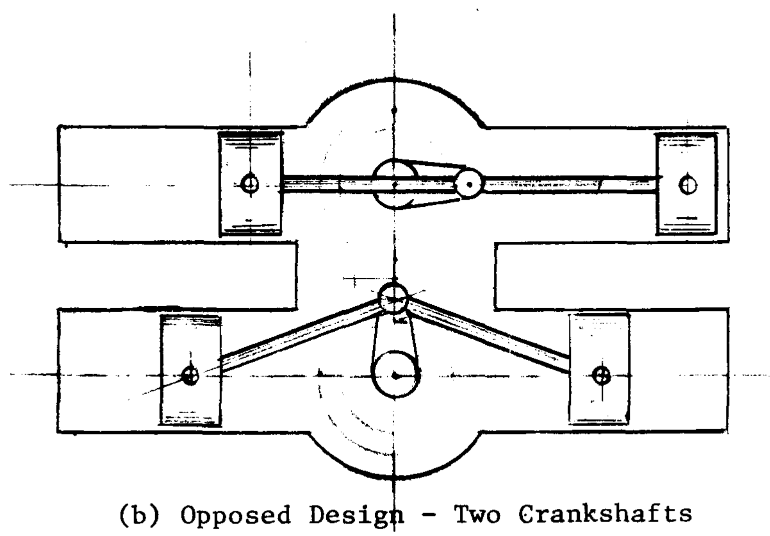
An alternative to the above is an "opposed" or perhaps "radial" design with either horizontal or vertical crankshafts (see sketch Figure 5-5). This system has the advantage of permitting 2 cylinders per single crank throw, as in the "V" type Diesel engine. It could be built with 8 cylinders opposed having 2 crankshafts either vertical or horizontal, or a 16 cylinder "X" engine with 4 vertical crankshafts. Another, but perhaps less likely, candidate is a 3 crankshaft "V" probably with the shafts horizontal.

5.1.3 Piston and Rod Seals

Perhaps the most critical mechanical components in the Stirling engine are piston rings and piston rod seals. A great deal of R&D effort has been devoted to these components over the past 20 years and significant progress has resulted. However, there is still need for further improvement to provide the performance and durability needed for stationary engines.



(a) Radial or 'X' Design - Four Crankshafts



(b) Opposed Design - Two Crankshafts

Figure 5-5 Alternative Engine Configurations Permitting Two Cylinders Per Crank Throw.

a. Piston Ring Sealing and Wear

Stirling engine piston rings must prevent excessive gas leakage by the piston or displacer while minimizing mechanical losses due to ring friction. Liquid lubricants cannot be used to improve friction and wear characteristics because any lubricant entering the hot sections would thermally decompose and contaminate the system (especially the regenerator matrix) with solid residue, and dilute the H_2 working gas with high molecular weight hydrocarbons, such as methane (Ref. 33, p 9). Therefore, solid lubricant type rings must be employed and their wear characteristics become the critical factor.

At present, the rings are made of a polymer, PTFE, or Teflon, which is impregnated with fiberglass and sold commercially as Rulon, Fluon, and other trade names. These rings have proven satisfactory for use in displacer type engines where piston ring life in excess of 4000 hours has been demonstrated and ring life as high as 17,000 hours has been projected based on measured wear rates (Ref. 33). However, double-acting engines have more severe piston ring operating conditions so further improvements may be required. The maximum pressure differential across the rings for a double-acting 4 cylinder engine is approximately 40% greater than for a displacer engine, if both are operating at the same mean pressure and assuming the displacer engine is equipped with a buffer space operating at constant mean pressure.

The wear rate for polymer type piston rings is a function of the P-V product (product of mean pressure differential and mean piston velocity). As the pressure differential across the ring increases, the wear rate goes up and there is also a tendency for this type ring to be extruded into the clearance spaces. Temperature also affects ring wear rate, but the relation is more complex and is not well understood at this time. In order to keep ring temperatures as low as possible, Stirling engines employ long hollow pistons with the sealing rings at the cold end where they are protected from exposure to the hot working gas. Also, the rings ride on a relatively cool area of the cylinder wall, frequently water cooled. As long as the ring leakage is small, the hot leakage gas is cooled nearly to ring belt temperature before passing through the ring. This is borne out by experience which shows that wear rates are nearly the same for both upper and lower rings, implying that the most important factor is the temperature of the surface on which the rings operate.

The pressure differential across the power piston sealing rings changes direction twice during each revolution of the crankshaft. This causes the sealing rings to move from one side of the ring groove to the other and leads to erratic sealing behavior. One of the problems that has resulted from this phenomenon is a tendency for double-acting pistons to pump gas between cylinders resulting in dangerously high pressure in one engine module and "starvation" of gas in others. In order to obtain more consistent ring sealing behavior and solve this "pumping" problem, recent double-acting engines have employed two sealing rings per piston as shown in Figure 5-6. A groove cut into the piston between rings is connected to the internal volume of the piston. By slots cut in the outer sides of each ring, the rings are made to pump outward which tends to produce minimum pressure in the volume between the rings. True minimum only occurs with infinite volume, however, and as a result of finite volume, there is greater leakage between hot and cold spaces across the rings. Nevertheless, this scheme has helped to maintain uniform mean pressure among cylinders while holding ring leakage to acceptable values.

Polymer type piston rings with various fillers including fiberglass are used in commercial air and gas compressor service. For reasons not yet fully explained, the wear rates of these rings are from one to two orders of magnitude greater than similar rings used in Stirling

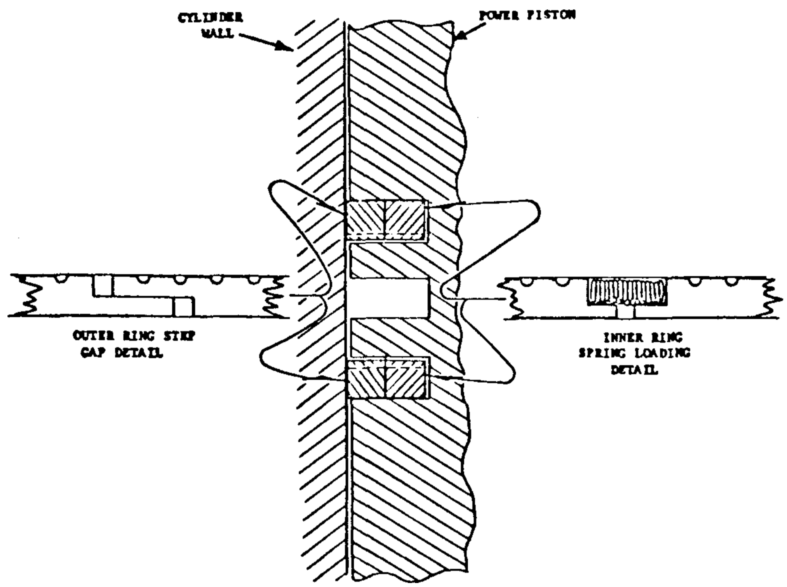


Figure 5-6 Typical Piston Sealing Ring Configuration (Ref. 33)

engines. This includes operating experience over many thousands of hours at Philips, General Motors and United Stirling. The lower wear rates in Stirling engines are at least in part, due to the clean, inert working gas environment (no oxygen present).

It is recommended that additional effort be made on piston ring design, particularly as to improving wear resistance, understanding the reasons for inconsistent pumping characteristics, and developing improved ring designs that simultaneously eliminate pumping and improve sealing characteristics.

b. Rod Seal Leakage and Wear

In the design of Stirling engines, the sealing for loss of working gas is separated from the piston or displacer sealing to minimize loss of power by gas leakage and ring friction. With this arrangement, piston ring design balances power loss by gas leakage versus ring friction and the more severe sealing to prevent loss of working gas to the environment is at the piston rod seal at the cold end of the cylinder. A tighter seal can be achieved without excessive friction, since the seal length is shorter than for a piston seal. Thus, gas leakage past piston rod seals is more critical than piston ring leakage because it represents a loss of working fluid from the system which must either be recovered, cleaned and pumped back into storage, or supplemented by make-up. Besides minimizing gas leakage, the rod seal must also prevent the crankcase lubricating oil from invading the engine working gas space, and fouling the hot heat exchange components and working gas with decomposition products.

Two basic types of rod seals have been used on Stirling engines to date: sliding seals and the "roll sock" seal invented by Philips in 1960 (Ref. 7). Although the "roll sock" seal is attractive because of its zero leakage characteristic, the developments to date, have not produced sufficient or consistent seal life (Ref. 33, pp 21-28). The rolling diaphragm, usually made from polyurethane material, has proven to be very sensitive to quality control during manufacture, small occluded impurities causing premature failure. As a result of these difficulties in development of a durable roll sock seal, United Stirling has concentrated their efforts on development of sliding seals for their double-acting engines, building upon the extensive seal work done at General Motors Research during the 1960's (Ref. 33, pp 7-20).

The duty imposed on the piston rod seal is influenced by the design of the engine. For engines with the same mean pressure, the pressure differential can be somewhat higher for double-acting designs than for single-cylinder displacer engines with a buffer space. United Stirling compensates for this by installing a partial seal consisting of a modified "O" ring as the innermost seal, to drop the pressure at the main seal to somewhere near the mean pressure level -- the equivalent of the pressure in the buffer space of a displacer engine (Figure 5-7).

Using state-of-the-art sliding rod seals, United Stirling obtains leakage of hydrogen from the 4 piston rod seals of a 100 bhp, 4 cylinder double-acting engine ranging from 0.1 to 0.2 grams/hour at full load. The life of their piston rings, as well as rod seals is currently about 4000 hours. Materials on which the rod seals operate are not as important as is their hardness, roundness, straightness and surface finish. Experience at General Motors is described in Ref. 33. Quoting from that reference, p 19:

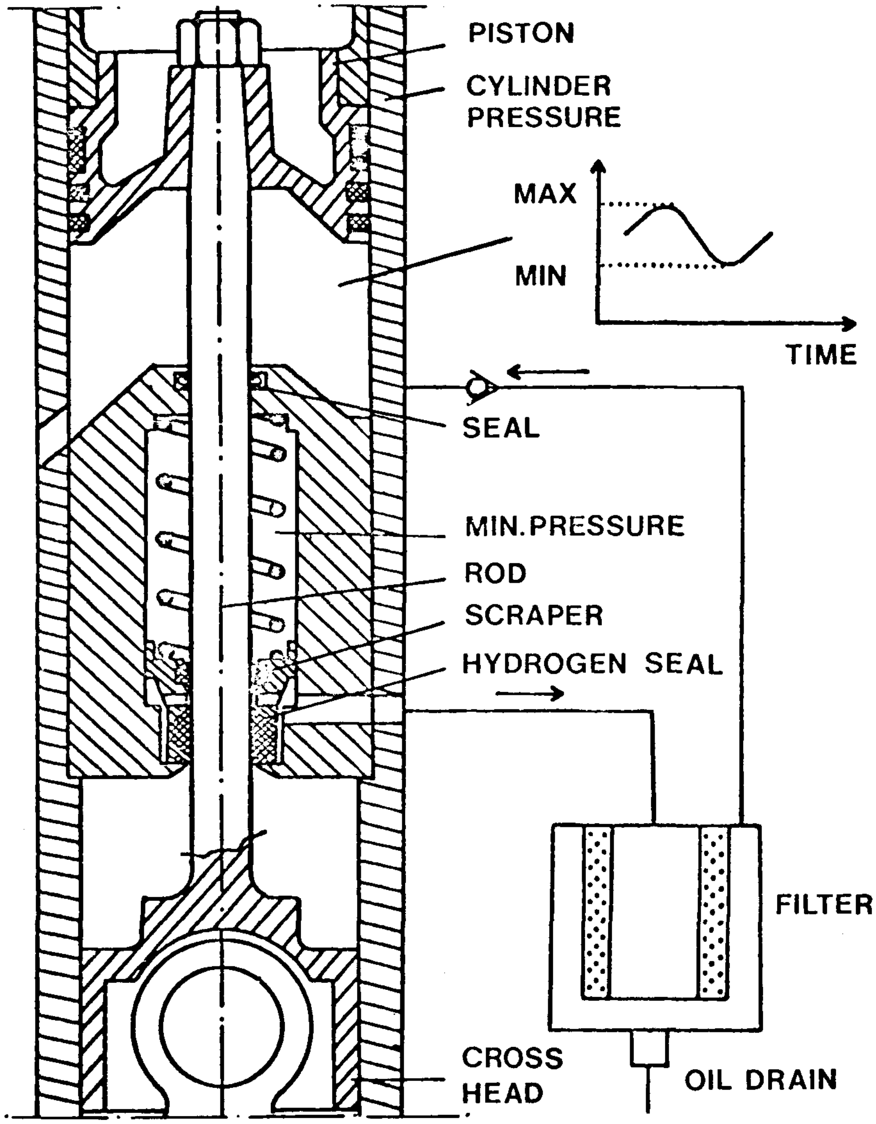


Figure 5-7 United Stirling Piston Rod Seal

"The surface finish of the piston rod or shaft was found to be important, with the optimum between 6 to 8 microinches. Initial values under 3 microinches resulted in high leakage -- an order of magnitude greater in some cases. Surface finish after running for hundreds of hours often decreased to about 4 microinches; but this appeared to have no effect on leak rate. An axial lap was preferred. Shaft out-of-roundness should not exceed 50 microinches and axial wave limit should not exceed 15 microinches in the seal operating region. Hydrostatic type grinding spindles are preferable; centerless grinding should be avoided".

At present, piston rod, sliding seal friction losses are excessive, amounting to as much as 400 watts per seal in a 100 bhp double-acting engine operating at a mean pressure of about 140 atmospheres. The seals are highly loaded both for restricting gas leakage, as well as to prevent ingress of lubricant. This necessitates cooling of the piston rod by an oil spray, which adds to the engine cooling load. Losses in the sliding seals for the GM small displacer engines were about 50 watts per seal; but the design employing a modified "O" ring was not suitable for mean pressure in excess of about 50 atmospheres and was not effective in preventing oil migration. Future sliding seal designs may incorporate dual function features. At relatively low mean pressure, one kind of seal with lower friction losses might function in combination with an oil scraping ring. As engine loading increased (higher mean pressure), another kind of seal and oil control scraper would gradually take over. As the engines become larger, the clearance between the piston rod and seal (as well as piston ring clearance) will not increase in proportion to the engine scale factor γ, but more nearly as $\sqrt{\gamma}$, or even less. In addition, the seal length is increased in proportion to γ, and internal working space volume increases γ^3. Consequently seal leakage as a percent of engine power rating will decrease and the time for a given loss of mean pressure should decrease as $\gamma^{1.5}$.

It is recommended that additional R&D be devoted to the effects of these variables on seal leakage and life; also more innovative seal designs need to be conceived and developed.

5.2 Characteristics of Basic Engine

The Stirling engine may be considered to consist of two major subsystems; the basic engine or thermodynamic cycle and the heat source system. This section deals with the basic engine which includes the components required to implement the basic cycle, e.g., cylinders and pistons, heater, regenerator and cooler, working gas and controls. The choice of mechanical arrangement for the basic engine was discussed above. This section will begin with the selection of the working gas, followed by a discussion of engine scaling rules which will permit the basic design parameters of large stationary engines to be determined by means of scaling up the size of existing automotive/truck engines. After presenting typical design specifications for a 2000 hp stationary engine, its performance characteristics will be estimated by applying the scaling rules. Then, the latter part of this section will describe the design features and problems encountered in the heat transfer components of the basic engine, i.e., heater head, regenerator, and cooler. Finally, the engine control methods are described and several miscellaneous characteristics of the basic engine are discussed.

5.2.1 Working Gases

The major considerations in the selection of the working gas for a Stirling engine are:

1. Thermodynamic, hydrodynamic and heat transfer characteristics.

2. Safety.

3. Chemical activity.

4. Diffusion through metals.

The thermodynamic, hydrodynamic and heat transfer characteristics of a given gas are all closely interrelated and are of vital importance to the satisfactory operation of the Stirling engine. Since each of these characteristics is determined by the molecular structure of the gas, the molecular weight of the gas is the most important parameter. All serious studies of the Stirling engine have concluded that performance increases rapidly with decreasing molecular weight of the gas (Ref. 16, 12, 8, 33). The optimum engine configuration is also a function of the molecular weight of the working gas with the major differences in the heat exchangers. Also the optimum piston speed of the engine decreases with increasing molecular weight. These basic effects are clearly shown in Figure 5-8 (Ref. 16), which compares the performance of optimized engines producing 225 hp/cylinder operating on hydrogen, helium and air. Although the same thermal efficiency can be maintained for the three different optimum engines, the engines with higher molecular weight gases have lower speeds and lower specific power as expressed in terms of power per unit displacement.

This scaling of performance versus hp/liter for different working gases can also be shown without a detailed design of optimum engines by means of a simplified heat transfer and flow friction loss analysis, which is presented in Appendix A. The quantitative results are within 25% of those shown on Figure 5-8. Thus, it is well established that hydrogen is the best Stirling engine working gas from the view point of obtaining high efficiency with the most compact, lightweight engine, although helium is a close second choice. All other heavier gases are not suitable for use in large Stirling engines.

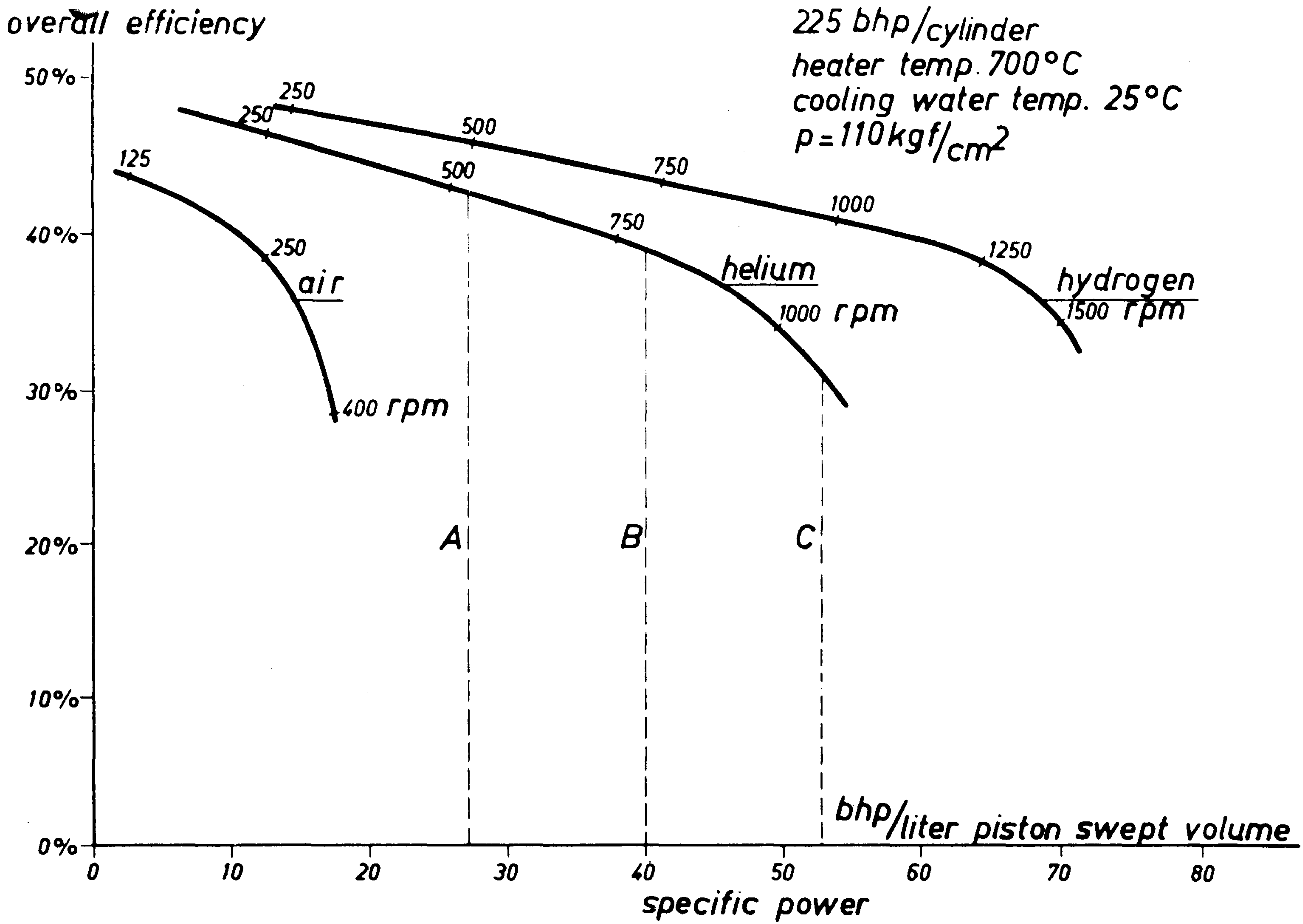


Figure 5-8 Effect of Working Gas on Efficiency in a Stirling Engine (Philips)

The second major consideration in the selection of the working gas is hydrogen safety. The issue is the improved engine power density with hydrogen versus the risk of forming explosive mixtures of air and hydrogen within enclosed spaces. The facts to be considered because of the flammability of hydrogen are:

1. Hydrogen has a wide flammability limit from the order of 6% in air to 74%.

2. Hydrogen disperses rapidly into surrounding air because of high diffusion coefficients and low density.

3. Hydrogen flames have low radiation. Small flames are almost invisible. This reduces the risk of secondary fires.

Hydrogen is a widely used industrial gas with safety problems comparable to hydrocarbon gases. Adequate ventilation of areas with potential leaks is the central rule in working with hydrogen. When possible, equipment is not enclosed and work is done in the open. The quantity of hydrogen in a large Stirling engine is small by industrial standards. Appendix B contains results of Stanford Research Institute tests performed for Ford on the hydrogen safety issue.

The third consideration in the selection of the working fluid is chemical activity. Inert helium clearly is best. The major issue with hydrogen is embrittlement of structural materials leading to loss of strength. The major mechanism is the diffusion of hydrogen into the material where the hydrogen reacts with carbon (usually a grain boundary inclusion) producing CH_4 which cannot diffuse through the metal. The resultant internal stress causes intergranular failure as though the material were brittle. Hydrogen embrittlement has not been a problem with experimental engines, since the high temperature materials (austinitic steels) which have been used, do not appear to be susceptible to embrittlement. Although water and air present corrosion problems for high temperature/high pressure engine components, these gases are not serious candidates as working gases for large engines.

The final consideration in the selection of a working gas is the problem of loss of hydrogen by diffusion through the hot structural parts of the engine. On a laboratory scale, this diffusion can be reduced to a very low level by the use of a coating with low hydrogen diffusion rates. Laboratory apparatus has been gold coated to eliminate the problem and Philips has reported success with the use of ceramic coatings. On the industrial scale of large engines, the hydrogen diffusion problem reduces to the initial cost of surface treatments to reduce loss by diffusion, versus the cost of hydrogen replenishment.

An engine with a sodium heat-pipe and hydrogen as the working fluid may have a more serious hydrogen diffusion problem. In this case, the operation of the engine may be degraded if sufficient hydrogen diffuses from the working space of the engine into the sodium space of the heat-pipe. Due to concern over this problem, all heat-pipe engines built to date, have been operated on helium, see Section 5.3.2.1.

5.2.1.1 Choice of Working Gas for Stationary Engines

From the above discussion it is clear that the selection of the working gas narrows down to a choice between hydrogen and helium. For applications where space and weight is of paramount importance, hydrogen is clearly the best choice. In stationary engines, however, volume and weight are not so important, except as they influence engine cost. High efficiency is, of course, very important. If the cost differential between high efficiency hydrogen and helium engines is not excessive, then helium would be preferred since it eliminates the safety issue and the diffusion problem. Moreover, as discussed in Section 5.3.2.1, when indirect heating via a sodium heat-pipe is employed, e.g., for systems burning solid fuels, then helium has always been used because of the concern that hydrogen diffusion would render the sodium heat-pipe ineffective* (Ref. 16). If, however, the hydrogen diffusion problem can be solved, it is possible that hydrogen engines could be run with heat-pipes.

Considering the strong emphasis on the solid fuel capability, it is the authors opinion that helium is the most likely working gas for large industrial stationary engines. However, hydrogen should not be ruled out before making a detailed examination of the probable cost differential between hydrogen and helium engines, and of the methods of utilizing low grade fuels.

5.2.1.2 Two-Phase Working Fluids

Condensing vapors, especially steam have been proposed as the operating fluid for Stirling engines (Refs. 65, 36, p 130). One of the advantages suggested for the condensing vapor is the larger ratio between mean effective pressure and cycle mean pressure, i.e., a more open indicator diagram. The second suggested advantage is more effective heat transfer processes, especially in the cooler.

The major problem with water is the higher molecular weight effect, as is shown in the foregoing discussion on the scaling of engine speed. Engine power density must be scaled down by a factor of the order of 3.5 compared with hydrogen to achieve adequate heat transfer with acceptable flow losses. This problem comes to light only when flow losses and heat transfer performance are considered in an integrated analysis.

* No tests have ever been performed with heat-pipes on hydrogen engines, so it is not clear how serious this problem really is.

The second problem with the use of a non-ideal gas is the variation of specific heat with pressure. A major feature of the ideal gas Stirling engine, is the ability to achieve near perfect thermal regeneration as limited only by the surface area of the regenerator. When the specific heat of the gas is higher at high pressure than at low pressure, the temperature difference at the hot end of the regenerator must be greater than at the cold end due to the unbalance in MC_p, as is well known for counterflow regenerative heat exchange. This effect reduces the cycle efficiency by increasing the temperature difference in the regenerator.

The third problem with the use of a condensing vapor is the non-ideal (irreversible) expansion or compression of two-phase mixtures. When a two-phase mixture is expanded or compressed, the temperature change caused by the pressure change is different in the liquid than in the vapor. A complex heat and mass transfer between phases is necessary to keep the temperature uniform as the pressure changes. These heat transfers across temperature differences cause losses and departure from isentropic compression that are not found in the same processes in single phase gases. These losses are a major factor in the non-ideal performance of reciprocating steam engines operating with wet steam. They may be expected to be even larger when a wet mixture is compressed. On compression, the vapor superheats and the liquid remains cold until heated by heat transfer from the superheated vapor. The process is further inhibited by migration of saturated vapor from the liquid surface.

Two-phase Stirling systems were considered by GM during their development program and found to have unacceptable performance based on their analysis (Ref. 33, p 56).

5.2.2 Engine Scaling Rules

Specific analytical studies of size effect for displacer engines were made by Philips and other licensees during the 1960's. The results became known as "scaling rules". They considered how scaling geometrically similar rhombic drive engine designs, covering a power range from about 10 bhp to over 3000 bhp per cylinder, would affect power, efficiency, stresses, heat transfer, sealing, etc. The studies are analagous to the Massachusetts Institute of Technology, Sloan Laboratory study made in the 1940 - 50's on geometrically similar internal combustion engines (Ref. 39). The scaling rules generally apply to other Stirling engine designs, such as the double-acting type.

Stirling engine scaling rules are like those for internal combustion engines in many respects for geometric similarity, and based on optimizing one size near the middle of the size range. Mean gas pressure, cooling water temperature and average heater tube temperature are assumed constant. When the major linear dimensions of the engine are scaled by the scaling factor, γ, the results are:

1. Bhp is proportional to the scaling factor (γ) squared, to a first approximation.

2. Engine speed is inversely proportional to the scaling factor; i.e., piston speed is held constant.

3. Stresses due to gas pressure and inertia forces remain constant; thermal stresses will increase with γ, but not linearly.

4. Efficiency remains nearly constant over a power range of about 50 to 1.

5. Heat transfer components, optimized for maximum engine performance do not scale according to exact geometric similarity, but overall thermal performance remains nearly constant.

For example, heater tube spacing does not scale as rapidly as γ, but more as $\gamma^{1/3}$*. Heater tube diameter tends to scale as γ, but the hydraulic diameter in the regenerator and cooler do not scale as rapidly as γ.

Applying these scaling rules to existing engine designs (e.g., Table 4-1), the piston diameter for 125 bhp cylinder will be about 7", and the piston will weigh approximately 35 lbs. From consideration of manufacturing, assembly and maintenance, the size and weight are within the limitations for handling by one man. Piston stroke is 4", and engine speed at maximum power approximately 1200 rpm. This speed is suitable for direct driving a 6 pole generator.

5.2.3 Typical Specifications for 500 - 2000 HP Stationary Stirling Engines

Although stationary Stirling engines may take on different forms and have different characteristics, depending on specific application, this section will describe the specifications and characteristics of a "representative" stationary engine. It is assumed that the primary function of the engine is electric power generation, so it will operate at constant speed with load varying according to demand. The utilization factor is expected to be high so the engine design should favor high efficiency and long life at some sacrifice of specific power (hp/ft^3 or hp/lb).

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* See Appendix D for detailed explanation.

Some derating should also be done in choosing the full load operating point to favor high efficiency and durability.

Since the primary advantage of the Stirling engine is its ability to burn solid fuels and other low grade fuels which would probably require indirect heating means (see Section 5.3.2), the "representative" stationary engine will be based on a sodium heat-pipe system to transfer combustion heat to the heater tubes (section 5.3.2.1). When the heat-pipe system is employed, current technology permits operation with heater tube temperatures of 800°C (1470°F) (1% creep in 10,000 hours), so this temperature is assumed for the "representative" engines. Cooling water temperature is assumed to be 30°C (86°F) which is typical for systems employing a wet cooling tower or equivalent.

Section 5.1.2 set forth the logic for choosing a series of multi-cylinder double-acting engines with 125 bhp per cylinder to cover the 500 - 2000 hp range. Thus, engines of 4 cylinder, 6 cylinder, 8 cylinder, 12 cylinder and 16 cylinder (all employing the same basic cylinder size and internal components) would have outputs of 500, 750, 1000, 1500, and 2000 hp respectively. The 4, 8, 12 and 16 cylinder engines would be built up from multiples of 4-cylinder double-acting engine power units similar to the United Stirling P-75 engine (Figures 5-4, 4-21, and Ref. 37). The 6 cylinder version could be implemented with a single 6 cylinder double-acting power unit.

The preliminary design of the "representative" engine series was accomplished by scaling from the most recent United Stirling heavy-duty truck engine, the P-75 unit (Ref. 37). This is a 4-cylinder double-acting "U" type twin crank engine (Figure 5-4) with a 86 mm bore, 48 mm stroke and 100 hp (75 kW) output, when operating direct-fired with hydrogen working gas, 720°C heater temperature and 70°C cooling water. The actual engine scaling was done by starting with a proprietary United Stirling engine map for the P-75 engine optimized for a heat-pipe heater head with helium working gas, and 800°C heater tube temperature and 30°C cooling water temperature (approximately 27°C in and 33°C out). At the derated speed of 1800 rpm and a mean working gas pressure of 14 MPa (2030 psi) the power output is 111 hp (83 kW) or 27.75 hp/cylinder. Since the power output varies as the scale factor γ squared, then to scale up to 125 hp/cylinder the scale factor becomes:

$$\gamma = \sqrt{125/27.75} = 2.12$$

Then the bore, stroke and rpm scale as follows:

Bore = 2.12 x 86 mm = 182 mm (7.18 in)

Stroke = 2.12 x 48 mm = 102 mm (4.0 in)

RPM = 1800 ÷ 2.12 = 850 RPM

Table 5-1 summarizes these foregoing specifications for a 2000 hp, 16 cylinder engine. The size is estimated by scaling up from the P-75 engine as the basic 4-cylinder power unit using the scaling factor of 2.12. Overall dimensions are estimated to be 143 in (3632 mm) long, 50 in (1270 mm) wide and 68 in (1727 mm) high. Weight is estimated at 32,000 lb (14,550 Kg), which represents a specific weight of 16 lb/hp. This is almost identical to the specific size and weight of equivalent power Diesel engines used for similar service, e.g., Alco Model 8V251G, (Table 4-2).

The 16 cylinder engine consists of four, 4-cylinder double-acting power units arranged in-line with twin crankshafts carrying 8 cylinders each. A single combustion system operating on either liquid or

TABLE 5-1

SPECIFICATIONS FOR REPRESENTATIVE 2000 HP
STATIONARY STIRLING ENGINE

Power Rating - 2000 bhp* - Continuous @ 850 rpm
- 2600 bhp - Intermittent @ 1130 rpm

No. Cylinders - 16 (four, 4-cylinder double-acting units)
125 bhp per cylinder (continuous)

Cylinder Arrangement - Twin crank - 8 cylinders in-line per crank

Bore - 7.18 inches (182 mm)

Stroke - 4.0 inches (102 mm)

Heater Temperature - 800°C (1470°F)

Cooling Water Temperature - 30°C (86°F)

Displacement - 162 in^3 (2.65 liter)/cylinder

Working Gas - Helium

Mean Operating Pressure - 2030 psi (14 MPa)

Combustion System - Indirect firing with sodium heat-pipe

Fuel - Can be furnished with either liquid fuel or solid fuel combustion system.

Brake Thermal Efficiency ** @ Full Load Operating Point - 39.6%

Brake Thermal Efficiency @ 1/2 Load - 35.6%

Performance Map - See Figure 5-10

Dimensions ***

Length	143 inches	(3632 mm)
Width	50 inches	(1270 mm)
Height	68 inches	(1727 mm)
Weight	32,000 pounds	(14,500 kg)

* Net power rating after allowance for all engine auxiliaries except cooling water system fan.

** Thermal efficiency is based on net shaft power output ÷ fuel LHV.

*** Without indirect combustion system.

solid fuel is mounted on one side of the engine where it supplies heat to the heater head assemblies of the four power units through a sodium heat-pipe system. Since in the twin crankshaft arrangement, the output shaft is geared from the two cranks, any desired output shaft speed could be employed, e.g., 1200 rpm to direct drive a 6 pole generator.

5.2.4 Estimated Performance Characteristics

An engine power and efficiency map for the United Stirling P-75 truck engine is shown in Figure 5-9, (Ref. 66). This is a direct-fired hydrogen engine. Power increases as the mean working gas pressure and operating speed are increased. The maximum efficiency of 37% occurs at about 1/3 speed and at maximum gas pressure.

The "representative" stationary engine discussed in the previous section was scaled from a United Stirling proprietary engine map for the P-75 engine optimized for a heat-pipe heater head and helium gas as discussed above. The engine performance map for this representative engine is shown in Figure 5-10. It was constructed by scaling from this proprietary map and making appropriate allowances for power consumed by the various engine auxiliaries such as combustion blower, oil and water pumps, fuel pump and atomizing air pump. Among these components the combustion air blower is the most significant, consuming an average of 3-1/2% of engine power. All other auxiliaries combined only consume about 1-1/2% of engine output. The performance map does not include any allowance for a cooling water radiator or cooling tower fan or a helium compressor. If an installation employing air cooling is contemplated, then an appropriate allowance would need to be made for the fan. Although the helium compressor power was not included, it will be very small because it is only running a small fraction of the time.

Note that the maximum thermal efficiency of this engine is about 40%, and also occurs at about 1/2 speed and maximum pressure. The operating point for the stationary power generation application has been selected at 850 rpm (about 70% speed) and 14 MPa cycle pressure to provide

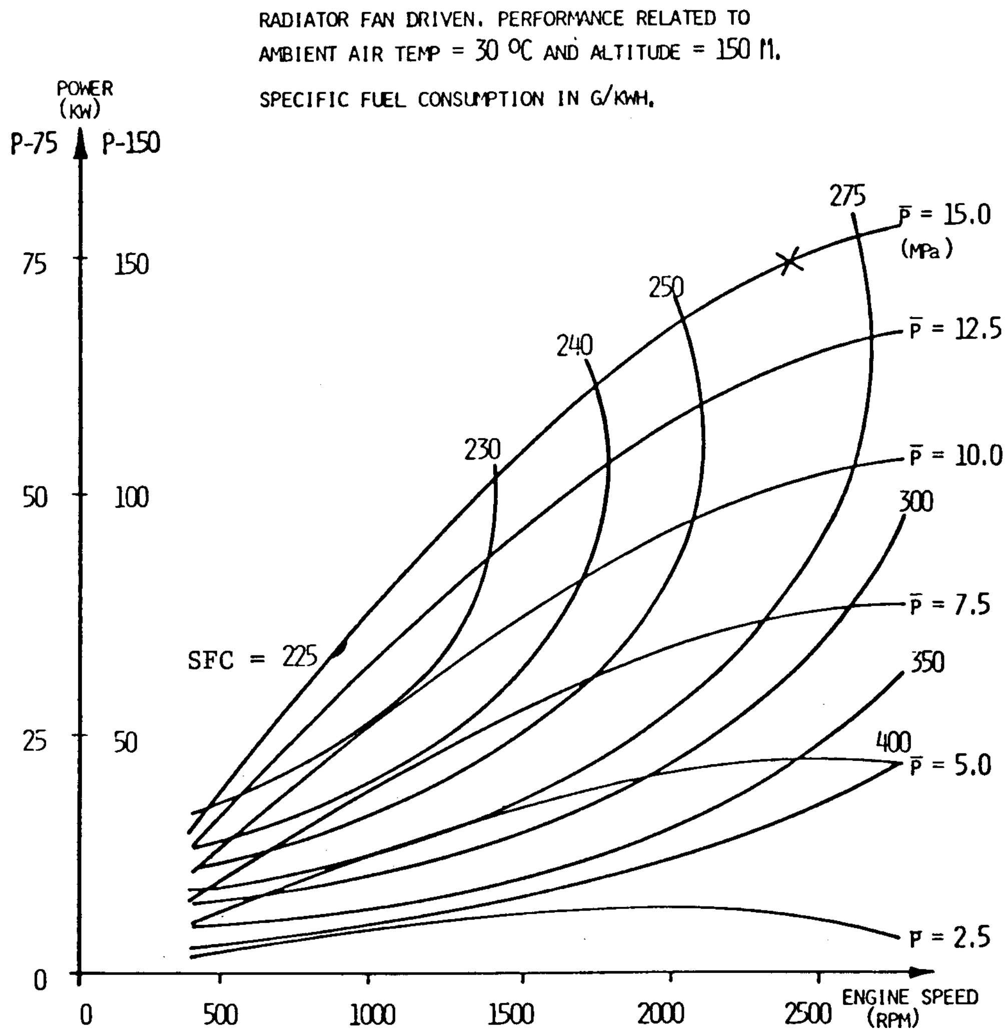


Figure 5-9 Performance Map Stirling Engine P-75/P-150 ; Mean Pressure Power Control. (courtesy United Stirling)

- Optimized for Heat Pipe
- Working Gas - Helium
- Heater Tube Temperature - 800°C
- Cooling Water Temperature - 30°C

$\bar{p}$ = mean working pressure (MPa)

η = Brake Thermal Efficiency (LHV)

* Net Power After All Engine Auxiliaries
Except Cooling Water Fan

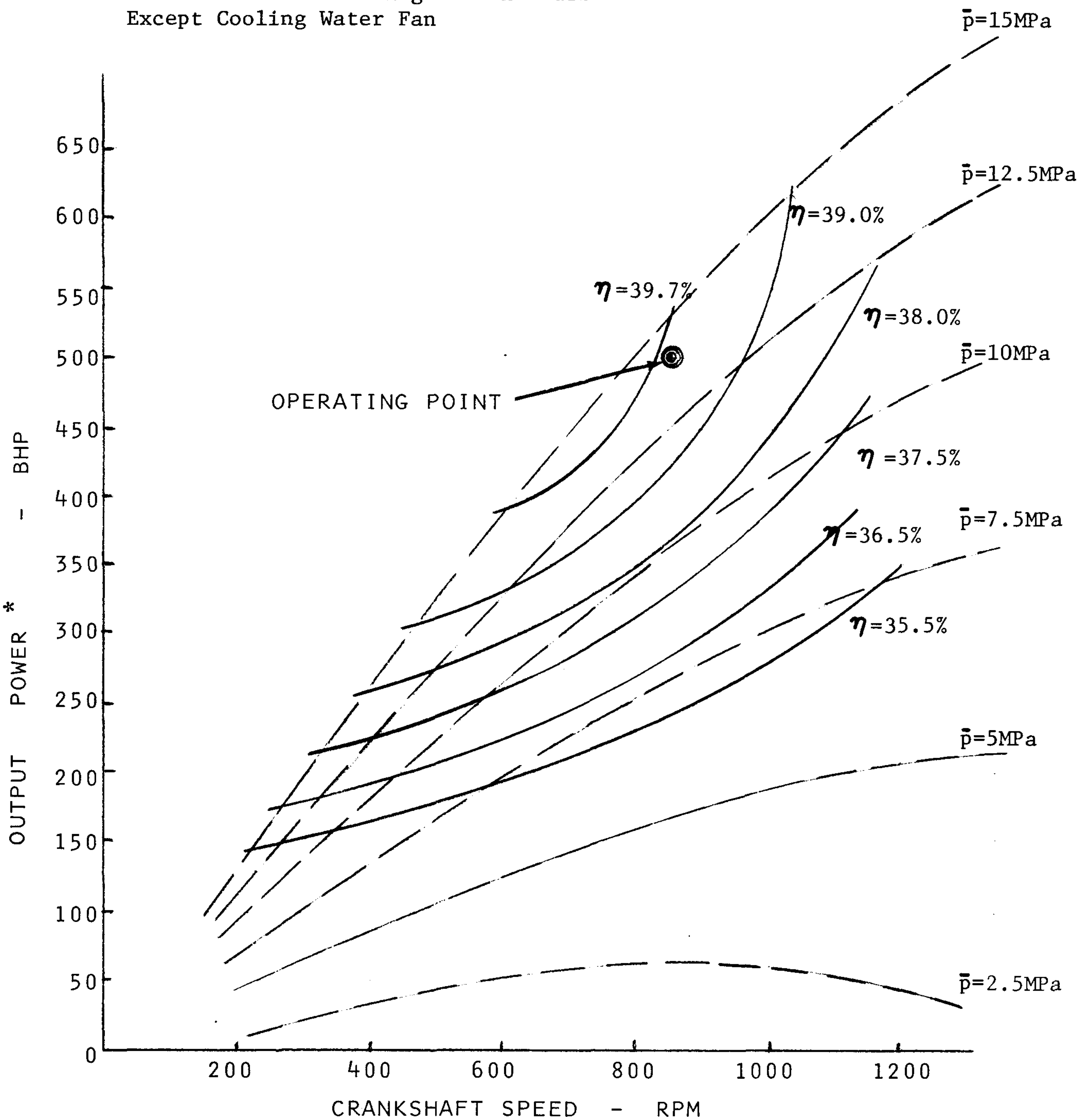


Figure 5-10 <u>Estimated Performance Map for Representative Stationary Stirling Engine - Basic Four Cylinder Power Unit</u>

improved efficiency and durability. Operating point efficiency is 39.6%. The effect of this selection on efficiency is clearly shown in Figure 5-10.

As stated in Section 5.2.3, the representative engine design is based on operation with a heater tube temperature of 800°C and an average cooling water temperature of 30°C. Both power and efficiency are affected by changes in these temperatures. Figure 5-11 illustrates the effect of heater tube temperature and Figure 5-12 shows the effect of cooling water temperature. These sensitivity curves clearly indicate the importance of maintaining high heater temperature and low cooling water temperature. For example, increasing heater tube temperature from 720°C to 800°C, increases power by 12% and increases efficiency by 10%; decreasing cooling water temperature from 70°C to 30°C increases power by 18% and increases efficiency by 13%.

Cooling fan power has not been included because of its dependence on a specific cooling system and installation, as for example, dry cooling with radiators, wet cooling tower, or direct cooling from streams, lakes, irrigation sources, etc. Generally, only the dry cooling system will require fan power. Fan power for dry type cooling systems depends on first, the initial temperature difference between the incoming water from the engine and the incoming ambient air temperature. It is also dependent on heat exchanger (radiator) frontal area, core depth, core design and fan efficiency. Curves of fan power shown in Figure 5-13 are typical for a well-designed radiator which might be installed exterior to a Stirling-electric power station.

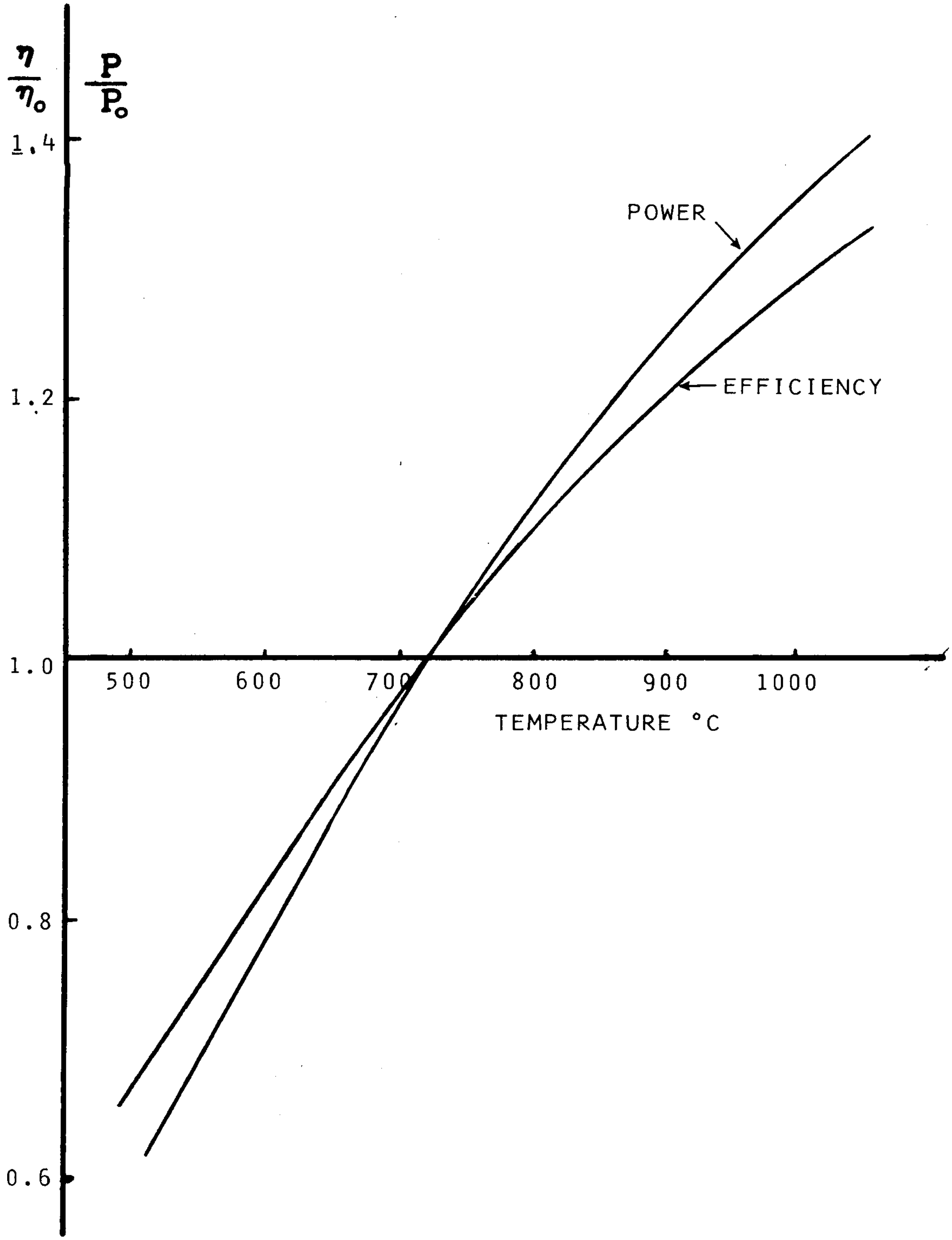


Figure 5-11 <u>Effect of Heater Head Temperature on Power and Efficiency in Stirling Engines</u>. (courtesy United Stirling)

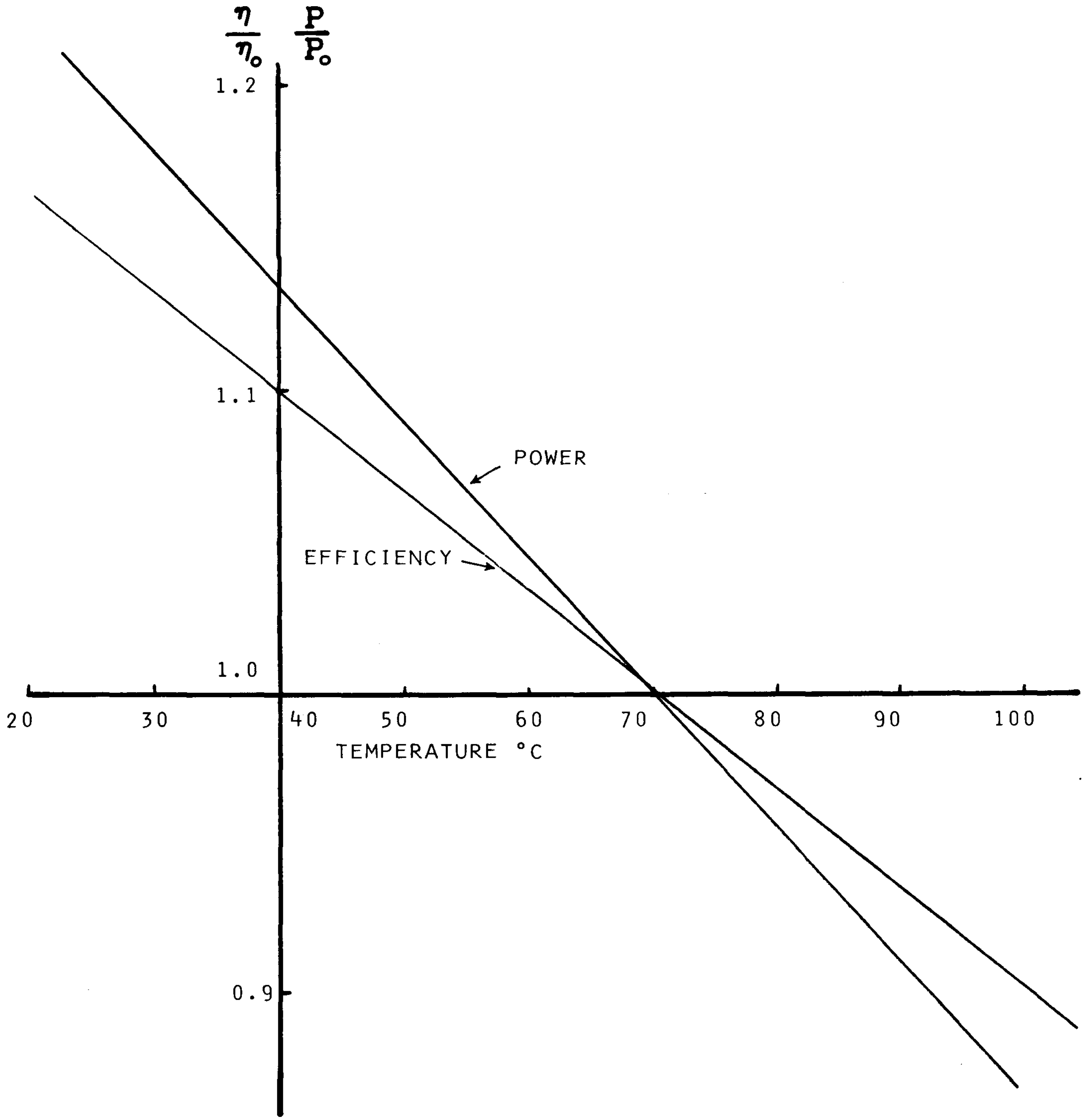


Figure 5-12 Effect of Cooling Water Temperature on Power and Efficiency in a Stirling Engine. (courtesy United Stirling)

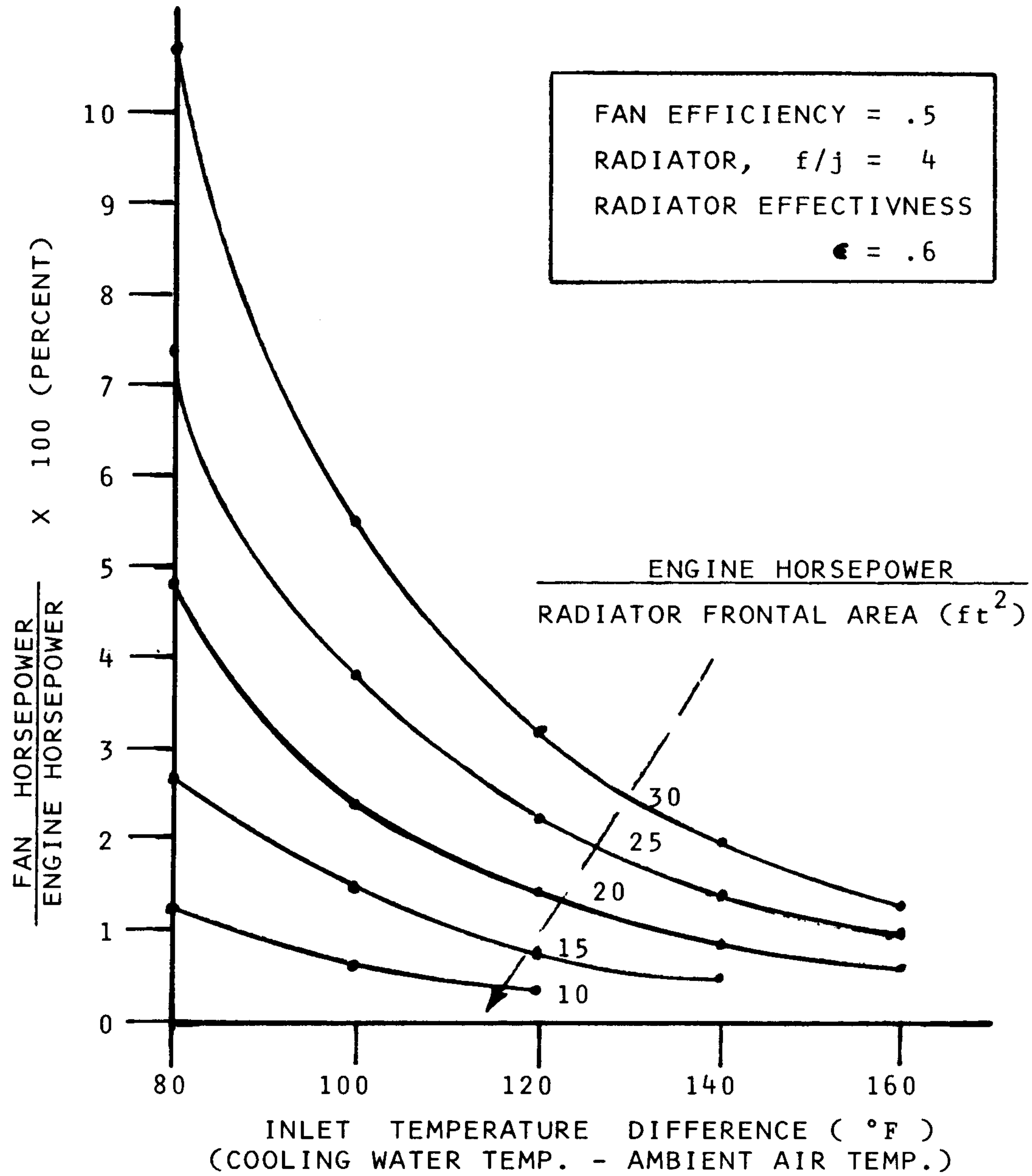


Figure 5-13 Cooling Fan Power For Stirling Engines Utilizing Radiator or Dry Cooling Tower.

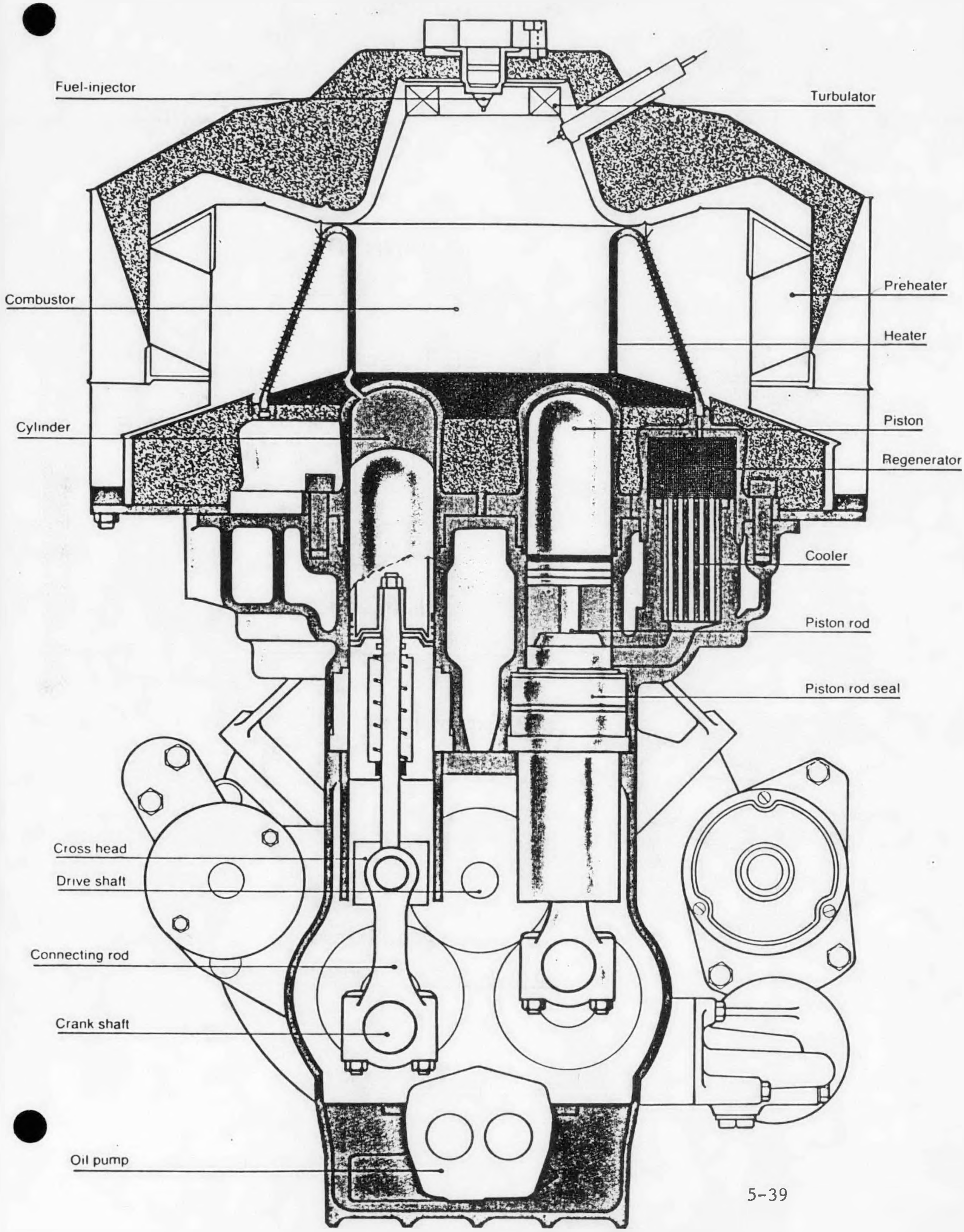
5.2.5 Heater Head Designs

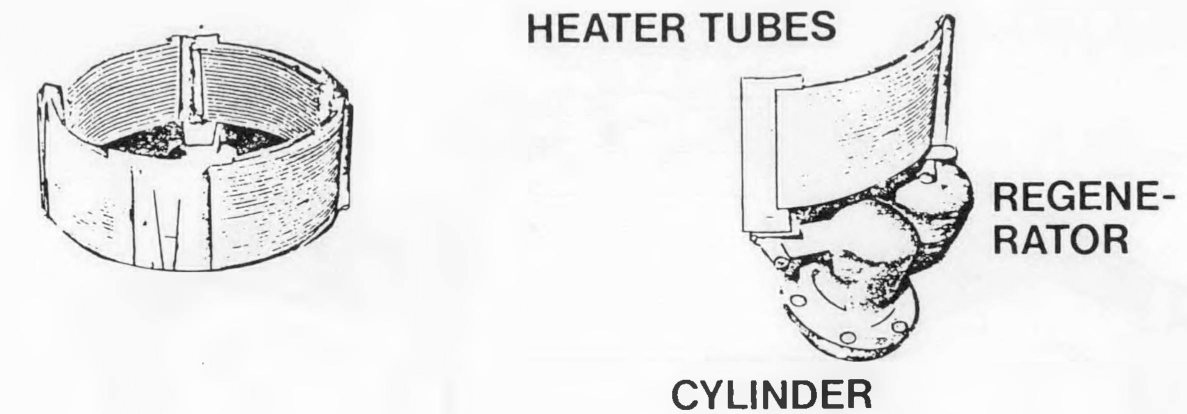
Presently, 4-cylinder double-acting Stirling engine designs incorporate a circular high temperature heat exchanger having two concentric rows of closely spaced tubes. Internal flow of working gas is sequential from one row to the next rather than in parallel (Figure 5-14). Generally the inner row, adjacent to the central combustion chamber is of the plain tube design whereas the outer row is finned. This is done in order to obtain minimum heater tube void volume for a given allowable heater tube wall temperature.

The circular array is further divided into quarter-sections, with the end of each tube in one section being connected by a short header to one cylinder head (hot space), while the opposite end of each tube is connected by short headers to two regenerator housings (regenerators per cylinder may vary from 1 to 3). A cooler tube housing is attached to the base of each regenerator, as seen in Figure 5-14. At the base of each of the two cooler housings a duct leads to an adjacent cylinder, entering the cold space under the piston. Thus, are the 4 cylinders and 8 regenerator-cooler units inter-connected for the double-acting functions.

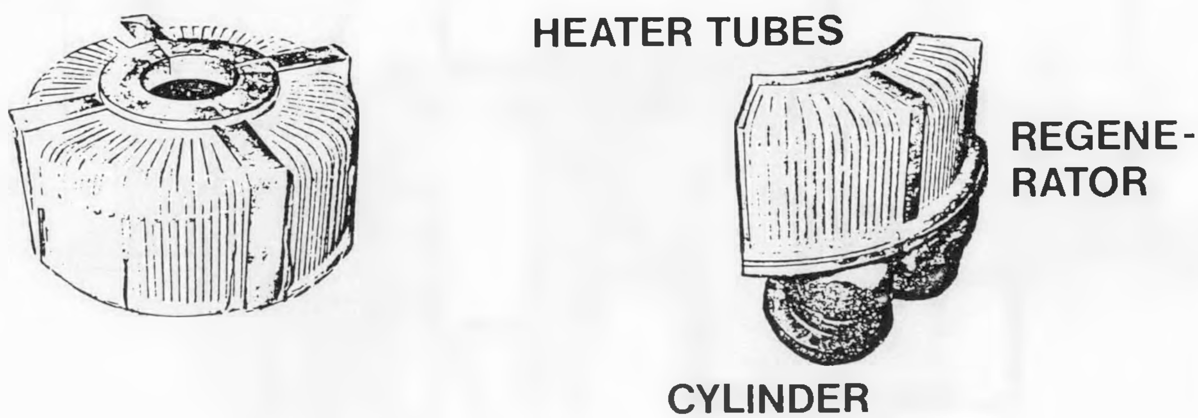
Major problems encountered by Philips and United Stirling in developing heater heads include minimizing dead volume and pressure losses in the headers, leakage of combustion gases between the quarter-sections and around the combustion chamber where it contacts the heater tube array, non-uniform temperature distribution between quarter-sections, as well as general fabrication and brazing difficulties. Figure 5-15 shows 3 of United Stirling's progressive heater designs, the tower, the temple and the most recent envolute design. The latter (shown more clearly by the photograph in Figure 5-16), has substantially eliminated the leakage and thermal expansion problems, simplified and reduced the dead volume in the headers

Figure 5-14 Pictorial of United Stirling Twin-Crank Engine

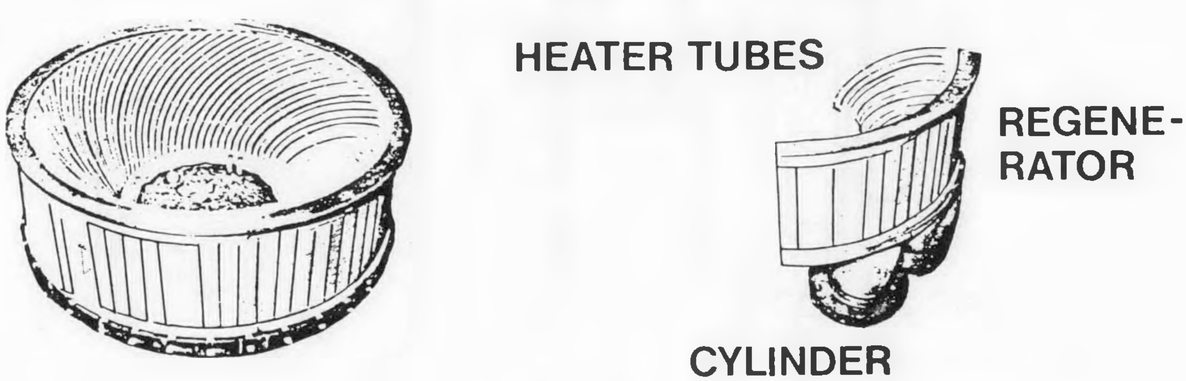




Tower type heater head.



Temple type heater head.



Envolute type heater head.

Figure 5-15 Three United Stirling Heater Head Designs

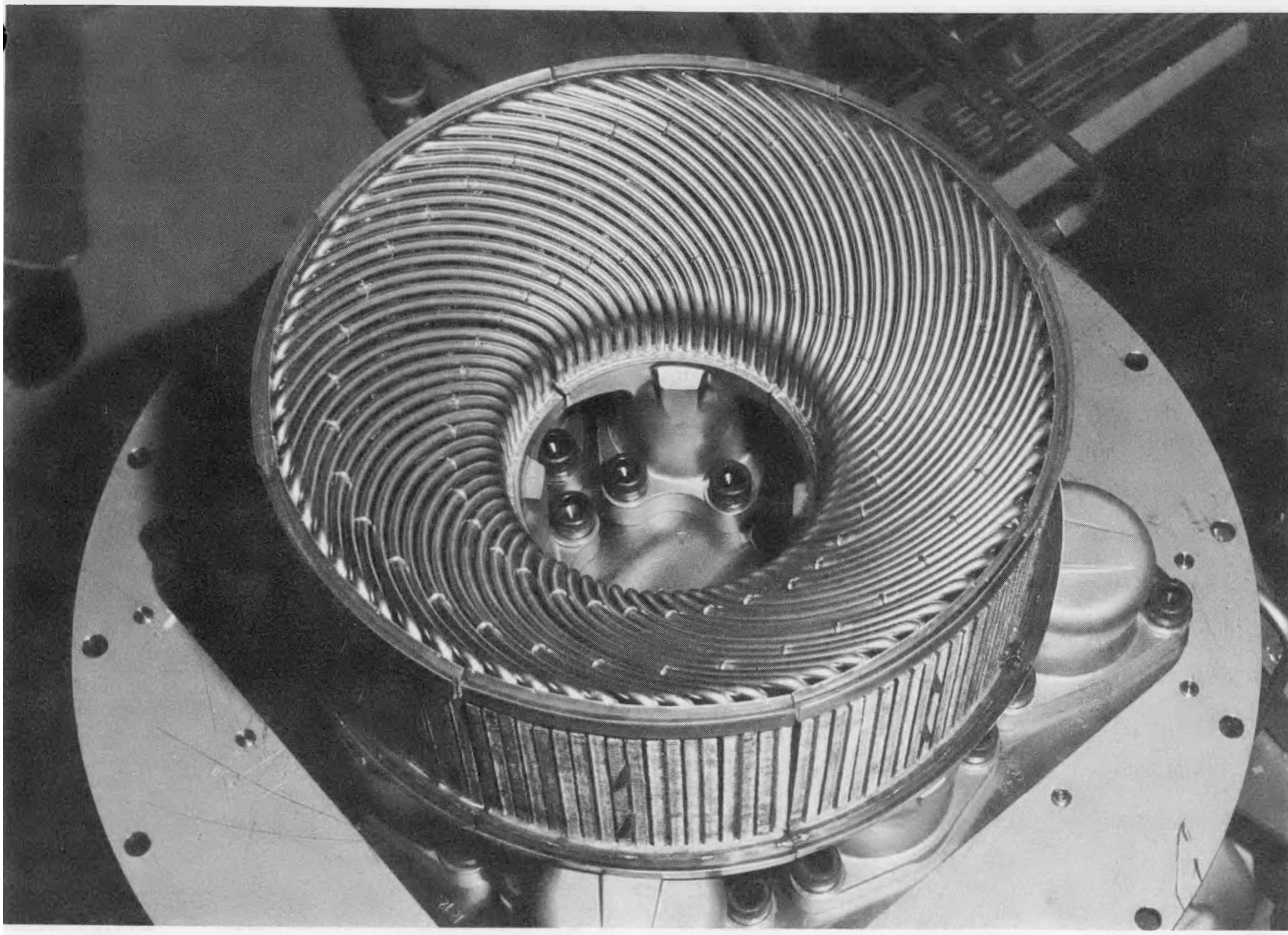


Figure 5-16 Envolute Heater Head for United Stirling P-40 Engine (courtesy United Stirling)

and made fabrication much easier. In production, it is believed automatic electron-beam welding can replace the slower and more expensive batch brazing process for attaching tubes to the headers. Each tube in the heater is alike, and headers are cast integral with the cylinders and regenerator housings. Overall thermal efficiency was increased from about 32% with tower heater design to over 36% with the envolute design, while the power has increased some 25% for equal mean working pressure.

It is the opinion of the authors that large double-acting Stirling engines will also use heater construction much like the smaller engines. All dimensions can be proportioned according to the scaling factor Gamma except for the spaces between tubes. These are scaled at a lesser "rate" than the actual factor, i.e., approximately $\gamma^{1/3}$. Since the temperature drop through the tube wall will increase with γ, for a given outer wall temperature limit the internal gas temperature will decrease slowly with increasing scale, but it is not a serious consideration in the range 500 - 2000 hp being considered. As the cylinder size increases, it is believed that techniques for joining the tubes to the cylinder and regenerator headers can be improved because of the greater space available.

At present, most heater cylinders are made of 310 stainless and the tubes of Multimet (N-155). Tube operating temperatures are about 720°C. The temperature limit for metallic heaters using more advanced wrought alloys, such as Incoloy 800 and Inconel 617, is considered to be about 815°C for a 1% creep in 10,000 hours. Creep in tubes and cylinders has not been a limiting problem for past experimental Stirling engines, and no rupture of a hot cylinder has ever occurred so far as the authors are aware. The question of stress rupture at elevated temperature, thermally induced cylinder ratcheting and cylinder stress distribution is further discussed in Ref. 33 p 58. The temperature of the cylinder and regenerator housing is 40°-50°C lower than the tubular part of the heater assembly since it is the practice to protect them from direct flame impingement by a molded ceramic cover.

For metal temperatures above 1500°F, superalloys used in gas turbine nozzles and rotating wheels are also possibilities for Stirling heaters. They involve higher costs both for the materials and fabrication. Some are only available in cast forms which practically excludes their use for small diameter tubes. There are also availability risks for some of the alloying elements, particularly chromium and cobalt.

The heater head may be the most critical component from a development, design and material point of view for improved efficiency Stirling engines. As has already been pointed out, all prototype Stirling engines use iron-base alloys with major alloy additions of nickel and chromium, while in some cases, alloys also containing cobalt and refractory metals such as columbium, molybdenum and tungsten are used. This results in two major concerns, that of high costs and the fact that most of the alloy additions are considered to be strategic materials (Ref. 53).

Prototype Stirling engines presently operate at gas temperatures to 1500°F and pressures to 3000 psi, while the conditions for future engines are proposed at hydrogen temperatures of 2000°F and pressures to 3900 psi. The heater tubes are exposed to a reactive hydrogen environment on the inside and to corrosive combustion gases on the outside. Under these conditions, the superalloys have the following problems:

- Degradation of materials properties such as high temperature strength and creep.
- Hydrogen loss due to permeation.

- Failure due to hydrogen embrittlement.

To some extent, some of these effects can be minimized by the use of barrier coatings which reduce the permeation of hydrogen. Philips has claimed a certain degree of success with this technique.

Ceramics appear to offer an attractive potential for both reduced cost and improved efficiency. However, it is the authors' opinion that the perfection of ceramics for heater heads at the higher temperatures being suggested will require extensive development and large expenditures. A great deal of funding has already been made available for gas turbine ceramic work and all of the technology being developed will be available for use in future Stirling engines. Two of the candidate materials, silicon carbide (SiC), and silicon nitride (Si_3N_4) have very good high temperature properties; however, it would not be productive to concentrate on their use in large Stirling engines at this time. These materials are being evaluated in less demanding applications and even in these situations, there would appear to be a long road to practical use of hot ceramic components.

Of particular interest to future heater head design, is development work now underway at Garrett in testing of a 10-U-tube silicon carbide heat exchanger fabricated by Norton (Figure 5-17). Material temperatures to 2200°F and helium gas pressure of 500 psi were used with no gas leakage problems or thermal cracking. Although the Garrett program (Ref. 54) is focused on ceramic heat exchangers for closed cycle gas turbines, the results should also be applicable to Stirling engines.

5.2.6 Regenerator and Cooler Designs

In the design of the regenerator and cooler for a Stirling engine, the major consideration is best thermal effectiveness with a minimum flow loss and a minimum dead volume in the working gas space. A combined

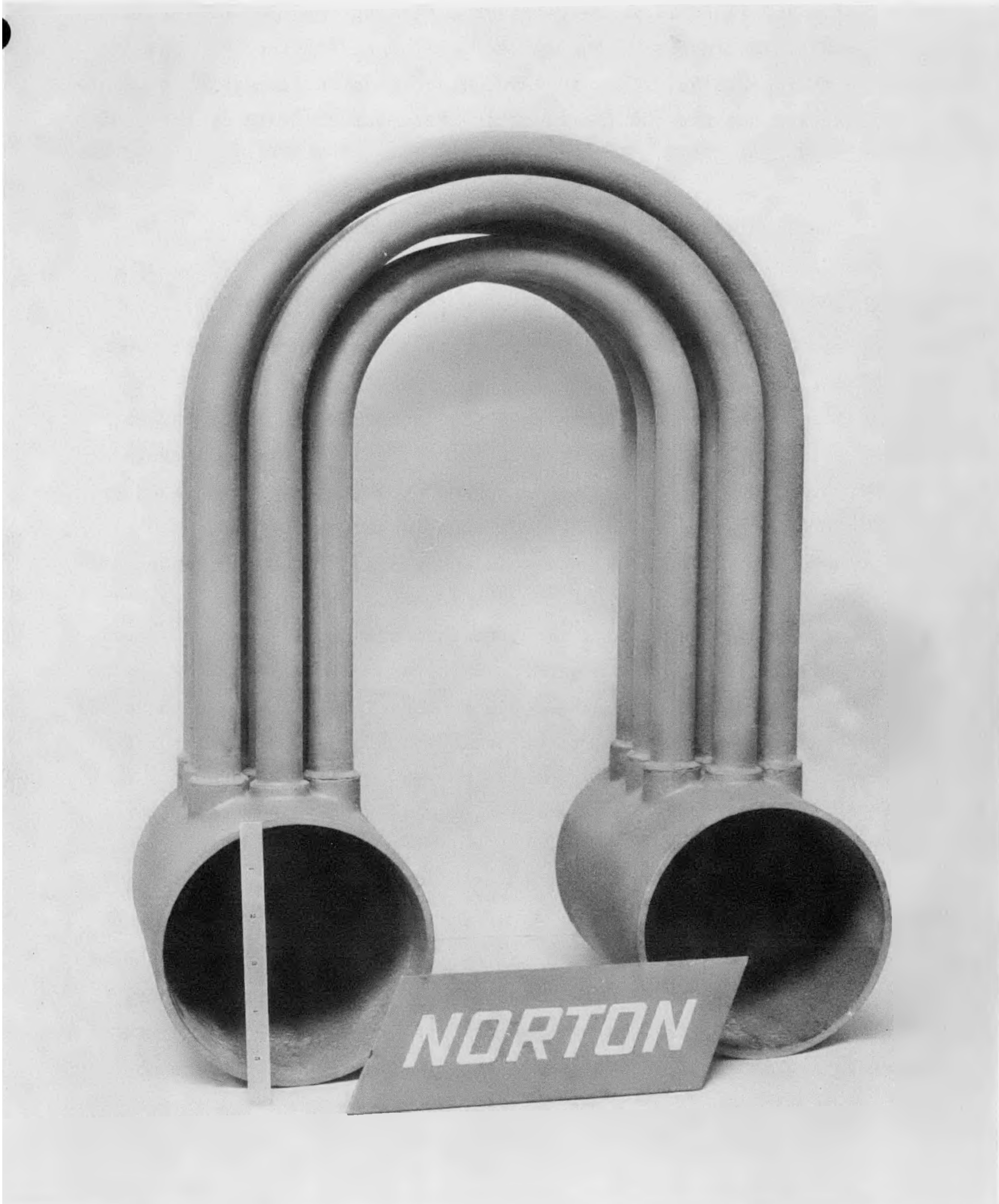


Figure 5-17 U - Tube Silicon Carbide Ceramic Heat Exchanger (courtesy Norton)

heat transfer and fluid flow analysis shows that for a given performance, the void volume decreases as the hydraulic diameter of the flow passages is reduced (Ref. 10, 15). This indicates that the finest mesh screens are best for the regenerator packing and that the smallest diameter tubes are best for the cooler.

5.2.6.1 Regenerator

In the regenerator, the lower limit on the hydraulic diameter is fixed by axial conduction heat leak, by non-uniform flow distribution and by packing material costs. As the hydraulic diameter becomes smaller, the conduction leak in the regenerator container becomes more significant because the regenerator becomes shorter with a larger frontal area. This gives a higher temperature gradient over a larger regenerator wall cross section. A good regenerator packing should be inexpensive to produce with a fine mesh. It should have a low effective axial thermal conductivity to reduce heat leak and a high transverse conductivity to help compensate for non-uniformity of flow over the frontal area. The packing must have a very uniform mesh, so that all passages and all surface is uniformally utilized. (In a packing with non-uniform mesh, the smaller passages have very low flow so their volume contributes to the dead volume without adequately contributing to the heat transfer.) In addition, the packing must have adequate heat capacity thermally close to the gas surface so that adequate heat storage can be achieved without excessive change in the local temperature of the matrix.

Since the regenerator is subjected to continuously varying mass flows, pressures and inlet temperatures,a cycle simulation is required to find the average duty imposed on the regenerator. Although some (Ref. 34, 35, 40, 52), have calculated the regenerator action as an integral part of the cycle simulation, satisfactory results can be obtained by first calculating the average duty on the regenerator and then using less detail models for calculating regenerator effectiveness (Ref. 10, 15). In any case, a good regenerator design has the proper balance between the losses and the influence of the void volume.

To date, most Stirling engine regenerators have consisted of multiple layers of fine wire screen, typically 200 mesh woven with 0.0015" diameter stainless steel wire. As far as the authors are aware, both Philips and United Stirling still use this type of regenerator matrix.

Ideally, when scaling up to larger engine cylinder sizes, the regenerator matrix material would remain the same, i.e., same screen mesh and stack height, but the regenerator frontal area would increase in proportion to cylinder horsepower (or as γ^2). In some cases, it may be desirable to accommodate this increase in frontal area by increasing the number of parallel regenerators per cylinder. In practice, it may be desirable to allow screen wire size to increase slightly, but not in proportion to engine scale factor γ (Philips recommends $\gamma^{2/3}$).

The principal problem with the Stirling engine regenerator is its high cost. In the early 1960's, General Motors studied the cost of a variety of possible Stirling regenerator matrix materials, including various weave screens, felt metal, foam metal, expanded metal, photoetched plate, electroplated nylon mesh, etc., (Ref. 33, pp 29-33). They found that costs varied from a high of \$11.70 per engine horsepower for square weave screens (@ 200 mesh) to about \$0.30 per hp for a GM produced foam metal called "Met Net". Unfortunately, the cheaper materials exhibit inferior performance due to greater voids, larger axial heat conductivity and greater flow losses. However, GM was able to obtain satisfactory engine performance on the Met Net regenerator material with engine efficiency about 90% of that obtained with wire mesh regenerators (Ref. 33). They believed that further improvements were possible with this type of material.

The authors believe that when the other critical problems in Stirling engine development are resolved and production of Stirlings is contemplated, the regenerator cost problem will need to be addressed and solved.

5.2.6.2 Cooler

The Stirling engine cooler must provide adequate heat transfer from working gas to cooling water while minimizing its contribution to flow losses and void volume on the working gas side. The construction of a typical Stirling engine cooler follows closely that of a shell and tube heat exchanger. The cooler consists of many small diameter tubes carrying the high pressure working gas with cooling water circulating outside the tubes, usually with a single pass crossflow arrangement. As with the regenerator, it is important to use the smallest possible hydraulic diameter (tube diameter) because the cooler void volume decreases approximately linearly with the tube diameter when the cooler flow losses and heat transfer performance are held constant.

As the diameter of the tubes is decreased, the number of tubes in parallel must increase according to the relation $n \propto 1/d^2$; thus, the final choice of tube diameter becomes an engineering compromise between performance and manufacturing cost. A typical tube size currently employed in automotive size engines is about 1/16" diameter.

Operating experience with cooler assemblies has been good; the only problems have been with leaks caused by faulty brazing or by corrosion from the cooling water side. A combination of careful manufacturing and properly inhibited coolant fluids should solve this problem (Ref. 33).

If cooler tube diameter is scaled $\propto \gamma$, then the internal heat transfer coefficient $\propto \gamma^{-0.2}$ and performance is only slightly degraded.

5.2.7 Controls

Torque

The modern Stirling is controlled by what is termed "mean pressure variation". The engine torque is proportional to the amount of working gas in the internal thermodynamic circuit of the engine. Since this internal circuit is normally a constant volume space, then torque is proportional to mean gas pressure, to a first approximation. There are at least three other, less effective, ways to control torque: (1) varying the temperature of the heater and/or the cooler changes torque; but the response is slow and thermal efficiency is drastically affected; (2) varying the dead space by means of added discreet volumes affects torque in a step function; but the system has proven difficult to achieve in practice and is no longer recommended; (3) by-pass control which opens a circuit between the top and bottom of the piston affects torque almost instantly; but the efficiency is cut to near zero. By-pass control is presently used with mean pressure control to prevent overspeeding upon loss of engine load. Its use is only for brief periods so that overall efficiency is only slightly reduced. The control system as used by United Stirling is shown schematically in Figure 5-18 , (Ref. 37). It is envisioned that regardless of the manner of heat addition, the torque control system would remain the same as outlined above.

Temperature

Under normal operation the temperature of the metal heater tubes is maintained constant by a thermostat, independently of torque control. A schematic of the system employed by United Stirling for use with gasoline and distillate fuel combustion systems is shown in Figure 5-19 . It consists of a thermocouple attached to the heater and connected to an electronic controler which positions an air throttle valve in

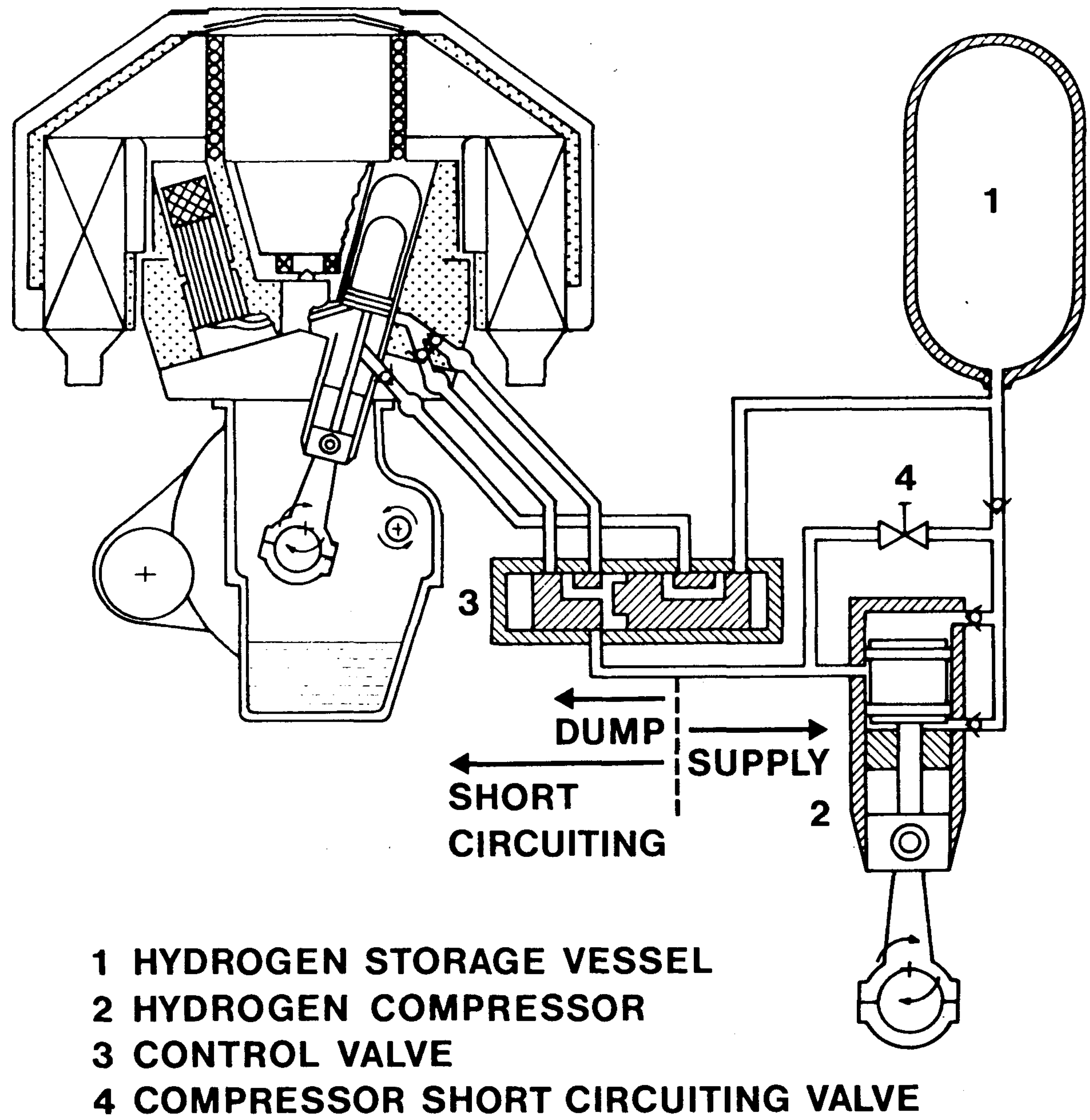


Figure 5-18 Simplified Diagram of the Power (Torque) Control System
(courtesy United Stirling)
(ref 38)

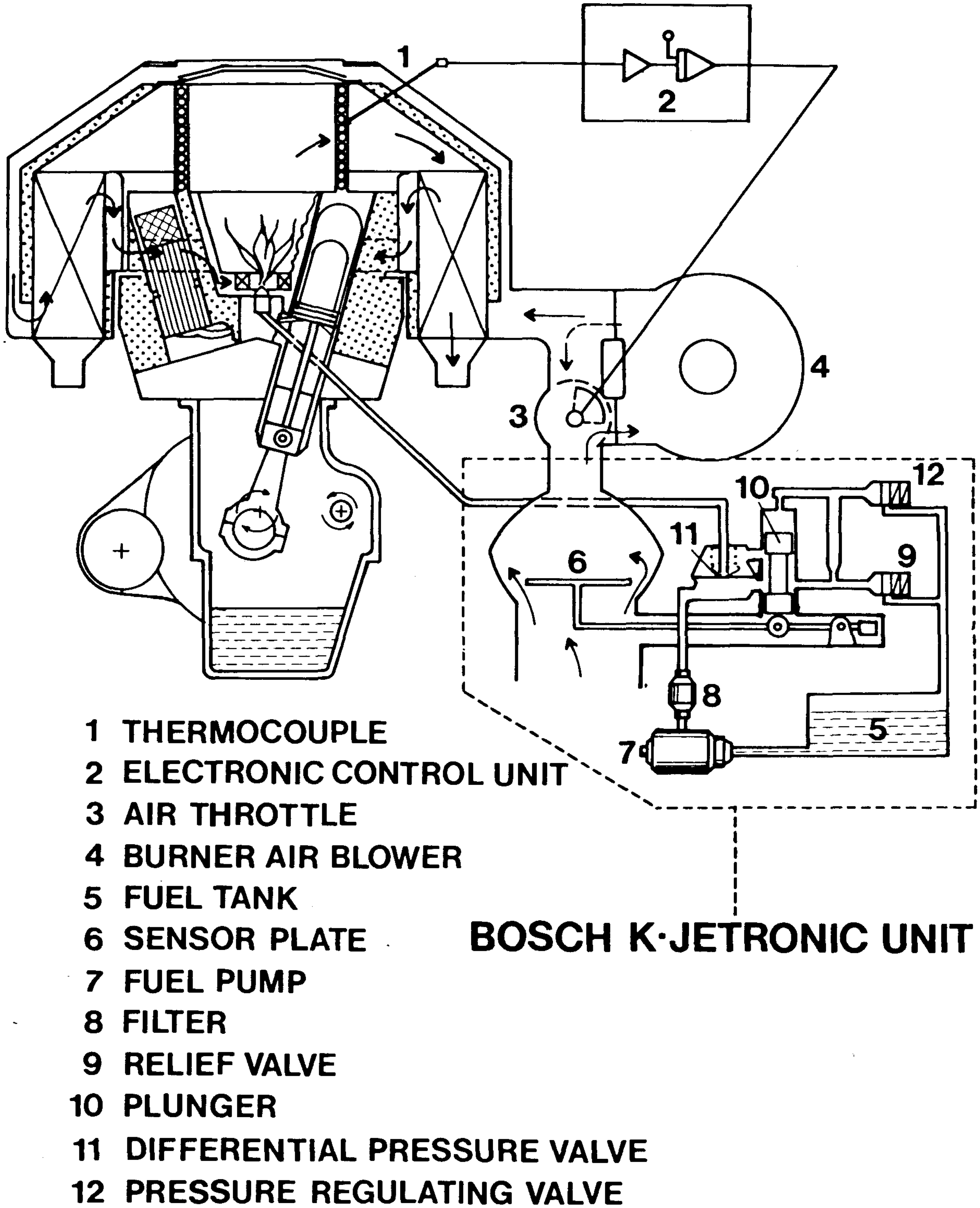


Figure 5-19 Schematic of the Temperature and Air-Fuel Ratio Controls
(courtesy United Stirling)
(ref 38)

response to the signal. An air mass flow meter (Bosch Jetronic unit shown) positions the fuel flow valve in proportion to air flow, thus modulating the burner heat flux and keeping the air-fuel ratio constant over the load range. It is envisioned that similar control concepts would be employed to maintain constant heater head temperature for other types of heat sources, such as solid fuel combustion, etc. These are discussed briefly under "External Heat Sources" in Section 5.3.

5.2.8 Miscellaneous Effects

5.2.8.1 Acoustic and Vibrational Characteristics of the Engine

The noise level of the bare engine, including radiator fan, is not expected to exceed 85 dB(A) at 1 meter distance. Linear vibration level should be at least as low as a thoroughly balanced six cylinder in-line internal combustion engine of the same power range. Torsional vibration is minimal because torque variations are relatively small. For example, the torque "ripple" of a 4 cylinder Stirling engine is approximately ±5% of the mean torque. It is difficult to compare a Stirling to an internal combustion engine without a complete Fourier analysis; but as a rough approximation a 4 cylinder Stirling is comparable in "smoothness" to a 16 cylinder spark ignition engine, both without flywheels. Reasons for the favorable acoustic and vibration levels include a closed working cycle with a nearly sinusoidal pressure curve, continuous combustion, the absence of valve gear, relatively low linear piston speed and complete balancing of the reciprocating and rotating parts.

5.2.8.2 Potential Hazards

Stirling engine cylinders and regenerator housings are pressure vessels operating up to 200 atmospheres. There have been no ruptures of a cylinder or housing at Philips, General Motors, United

Stirling, MAN-MWM or Ford since the inception of developments in 1954, so far as the authors are aware. Neither has there been any evidence of hydrogen embrittlement in any part of the engines.

The danger of hydrogen is small because the quantity which might escape could not form a combustible mixture in a typical room where a Stirling engine might be installed. The amount present within the engine working spaces and the storage bottle of a 100 bhp engine is about 35 grams. It requires a 9% by volume mixture with air to propagate flame in all directions, while a 4% mixture will propagate in a vertical direction only. Even if the entire 35 grams escaped quickly (a very unlikely event) it would require a room volume of less than 185 cubic feet in order to achieve an explosive mixture. The rapid diffusion of hydrogen into air compared to other gases reduces the chances of localized combustion in large spaces. The results of tests performed by Stanford Research Institute for Ford on the danger of hydrogen are included in the appendix. Future Stirling engines may store part of their hydrogen as a metal hydride rather than in a high pressure bottle. Two companies now market canisters of metal hydrides, typically for laboratory applications. One model weighs 52 lbs and holds 1/2 lb of hydrogen, sufficient for a 1000 hp Stirling engine. The weight is less than an empty compressed gas cylinder of equal capacity. If a canister is accidently punctured the leakage is self-limiting as the hydride cools to a point at which decomposition stops.

There has been some apprehension about the danger of sodium heat-pipes. These devices normally operate under vacuum conditions. A leak in the pipe would allow air to pass into the pipe rather than sodium spilling out. In addition, the quantity of sodium contained in a heat-pipe compared to sodium in a circulating or "pumped" piping system (for example in a nuclear reactor) is about 5%. Also, a Stirling engine running on a heat-pipe operates with helium as the working gas rather than hydrogen.

5.2.8.3 Engine Cooling Requirements for Various Climates

Engines are expected to be cooled with a standard inhibited 50-50 water-glycol mixture, and rated for 30°C ambient and 500 ft. altitude. Under emergency conditions power could be kept constant above 30°C for short periods by increasing the engine mean pressure, providing cooling capacity is sufficient. Altitude has a negligible effect on performance below approximately 5000 feet. At 10,000 feet maximum power is reduced to about 85% of sea level performance. It may be possible to install a turbo-supercharger on a Stirling engine, just as for an internal combustion engine, in which case the sea-level power can be maintained. It would require considerable redesign of the external combustion circuit and air pre-heater ducting (this is discussed in more detail in Section 5.3.1.4.)

5.2.8.4 Air Supply and Exhaust Provisions Required

Air consumption per bhp hour is expected to be approximately mid-way between a spark ignition engine and a Diesel engine, at full load. Air cleaning requirements are less severe than for internal combustion engines. Exhaust temperature is between 150° and 200°C, which simplfies exhaust ducting and allows the use of aluminum or some elastomers and plastics. No muffler is required.

5.2.8.5 Durability and Life

The following are the operation time goals for United Stirling on major components for commercial automotive service:

B_{20} life for major components before failure shall not be less than 4500 hours of operation.

Engine life shall not be less than the following hours of operation:

P-75	7500 hours (B_{50} life)
P-150	15000 hours (B_{50} life)

One major overhaul for the P-150 shall be foreseen at 7500 hours. A major overhaul is defined by United Stirling (for a vehicle) as an overhaul requiring removal of the engine from the chassis, dismantling for the purpose of exchange or repair of worn or deformed parts and reassembly.

For the critical hot components (heater head, pre-heater, burner) of the engine the following hours of engine operation are required:

P-75	8500 hours (B_{50} life)
P-150	17500 hours (B_{50} life)

The number of cold starts shall be assumed to be:

P-75	20000
P-150	40000

The commitment by United Stirling for production in 1982 was based partly on the demonstrated reliability of present development engines on dynamometers, and in experimental installations in passenger cars and trucks. For production units they expect reliability to be equal or superior to the modern Diesel engine. The amount of maintenance work required at 500 hour intervals is prescribed not to exceed 2 man hours; and intervals between refilling with hydrogen should not be less than 500 hours or 3 months, whichever comes first. This is for automotive service. For stationary engines the interval will be greater.

5.2.8.6 Projected Availability of Stirling Engines

United Stirling of Sweden (USS) has committed itself to production in Sweden of a limited quantity of Stirling engines in the 100 to 200 bhp range beginning in 1982. The decision has the backing of the Swedish government and includes the long range plans of USS. The three major areas of application in Scandinavia and northern Europe include underground mines, buses and smaller trucks, and heat pumps for cluster housing, greenhouses, etc.

5.2.8.7 Cost Per Unit After They Are In Production

The estimated list price per engine for the period of initial production is approximately 50% above a comparable Diesel engine. Estimated manufacturing costs (including material, components, salaries and wages) in 1974 dollars was $10.50 per bhp, based on 50,000 units per year.

For an engine overhaul, the cost of components and material is not expected to exceed 10% of the engine manufacturing cost; and the average cost to the customer should not exceed 30% of the customer's purchase price for a new engine.

5.2.8.8 Qualifications (Training, If Required) Of Maintenance And Repair Personnel

Training of maintenance people will be the responsibility of the engine manufacturer. The qualifications and background for technicians should be the same as for Diesel engine technicians.

5.3 Characteristics of External Combustion or Heat Source System

5.3.1 Direct Heating, Combustion with Air

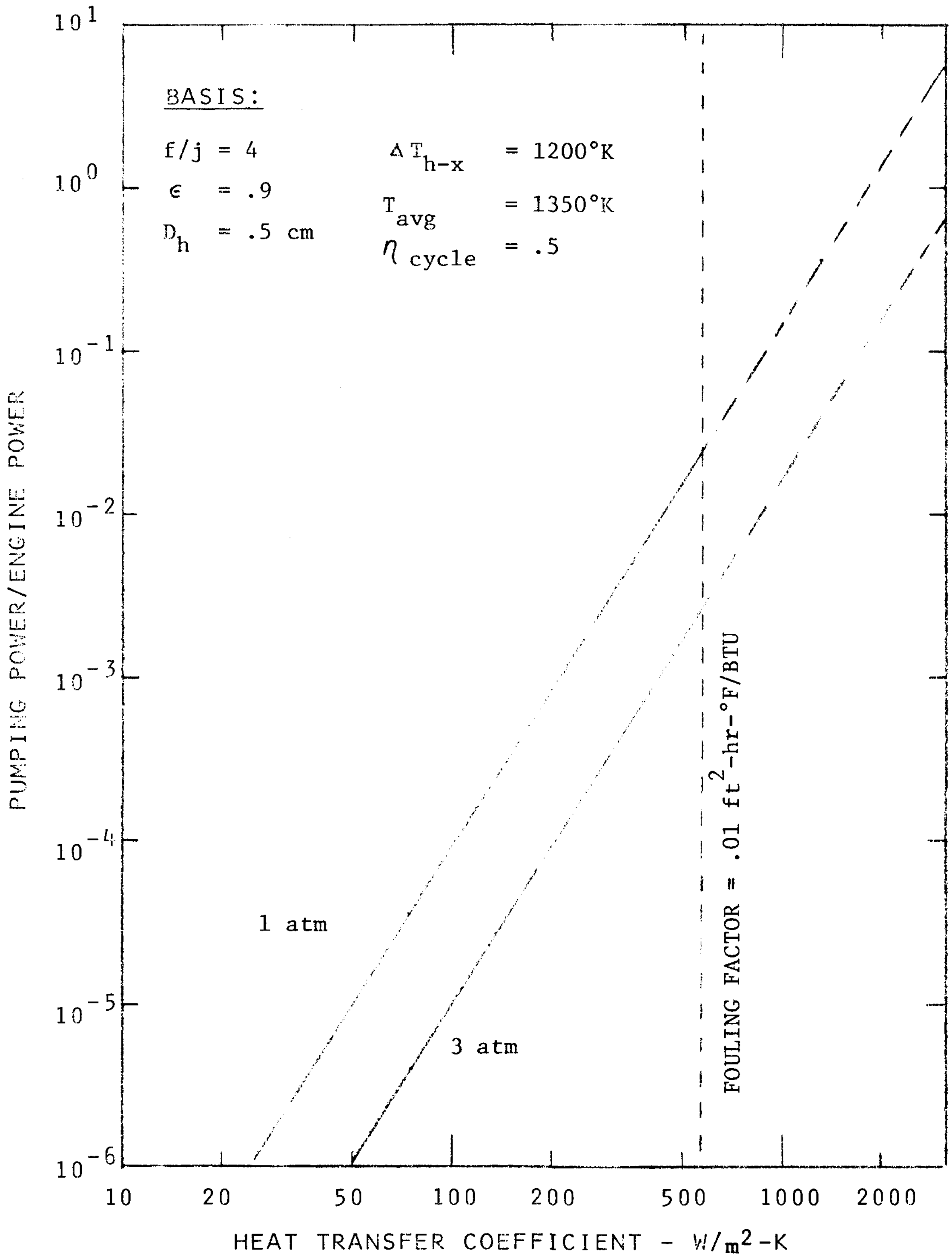
Most Stirling engines built and operated to date have employed combustion of liquid fuels with atmospheric air as the heat source. The simplest and most direct way to apply this heat source is to pass the hot products of combustion directly over the heater head tubes containing the working cycle gas. Thus, heat passes directly from combustion gases, through the heater tubes and into the working fluid. Before examining the details of the combustion system, it is useful to review the characteristics of heat transfer from combustion gases to the working fluid because it has an important influence on both the efficiency and size of the engine.

5.3.1.1 Heat Transfer

The optimum design of a heater head for combustion gas heating is quite involved and must simultaneously consider both the heat transfer design and the effect on the working gas cycle performance. Comprehensive analytical methods are required to produce an optimum design. However, significant insight to the problem can be gained by simple reasoning and by examining the results of some simplified analyses of the principal elements, as discussed below. In the design of the heater head assembly it is important to minimize the internal heater tube void volume and flow losses, because the engine specific power and efficiency are strong functions of these parameters. The heater tube void volume is proportional to tube diameter and total tube surface area, whereas the flow losses are proportional to surface area. Hence, both tube diameter and total surface area should be minimized for best engine performance.

Direct heating with products of combustion is characterized by fairly low coefficients of heat transfer outside the heater tubes due to the relatively poor thermal transport properties of gases, and by fairly substantial expenditures of combustion air blower power due to the relatively low density of the products of combustion. If the combustion products have a high particulate concentration, as with combustion of coal or heavy oil, then heat transfer is further inhibited by fouling of the heat exchanger surfaces. Because the internal (working gas side) coefficient and tube wall conductance are both quite high, the combustion gas-side coefficient controls the heat transfer. Thus, for a certain desired thermal performance or heat exchanger efficiency, the magnitude of the combustion gas heat transfer coefficient determines the amount of heater head surface area needed. With low coefficients of heat transfer, a greater heater surface area is required. To a certain extent, the system designer can provide increased heat transfer coefficients by providing higher velocities, although at the expense of increased combustion blower power.

The approximate relationship between the combustion-side heat transfer coefficient in a Stirling engine and the required combustion air blower power, expressed as theoretical air pumping power per unit of engine output, is shown in Figure 5-20. Also shown is the effective limit to combustion-side conductance due to fouling of the heat exchanger surfaces by combustion products. The value shown is typical for coal and heavy oil with periodic cleaning or soot-blowing. It is seen that in order to approach the fouling limit, it is necessary to expend combustion air blower power amounting to several percent of gross output. If the combustion gases are pressurized, as with a fluidized bed combustion system or in a turbo-charged combustion system, the necessary combustion air pumping power can be reduced significantly, but the maximum realizable conductance is still of the order of 500 W/m^2-°K (100 Btu/hr-ft^2-°F).



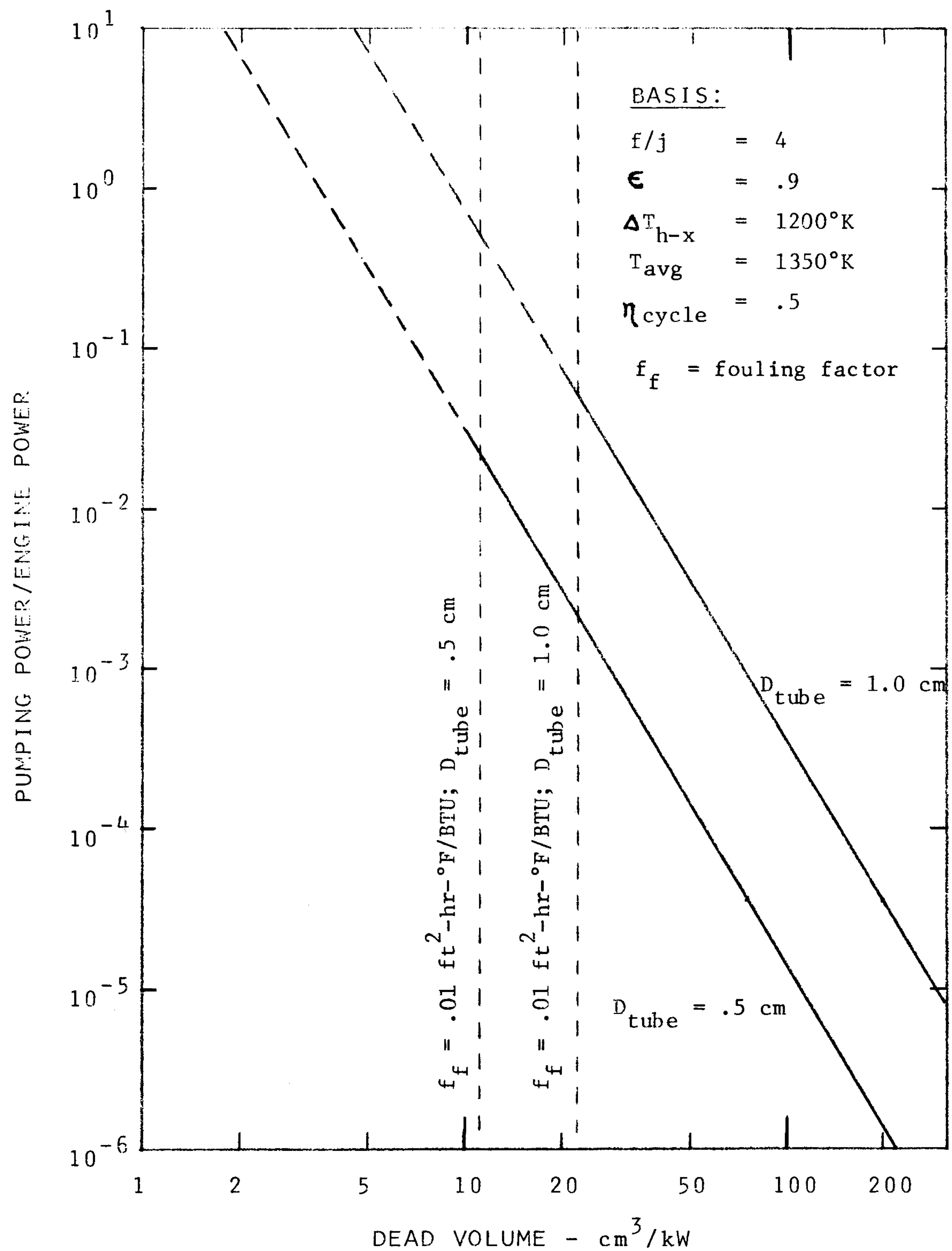
HEATER AIR PUMPING POWER VERSUS COMBUSTION-SIDE HEAT TRANSFER COEFFICIENT

Figure 5-20

Figure 5-21 shows the heater head dead volume, as determined by the external heat transfer coefficient, as a function of air pumping power. In this example, it is assumed that the heat exchanger consists of bare tubes, but if extended surfaces were employed, then the dead volume would be reduced in proportion to the effective extended surface ratio. Here it is seen that even for fairly small diameter tubing, the dead volume is of the order of 10-20 cm^3/kW even if a fairly high expenditure of air pumping power is made. If the air pumping power is to be less than the order of 1%, then the dead volume is of the order of 20-50 cm^3/kW.

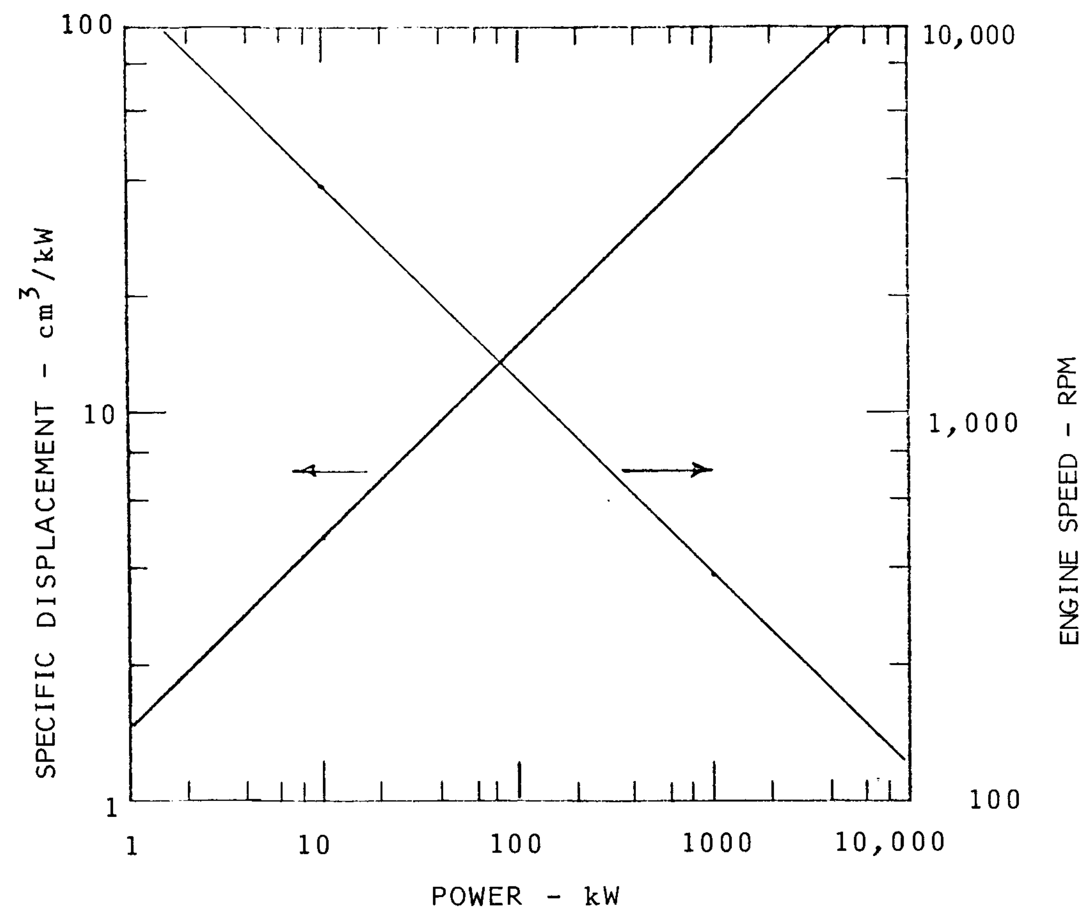
In order to estimate the signficance of dead volume as a function of engine size, an elementary analysis was performed to estimate the necessary expander displacement as a function of power level. Here, it was assumed that the mean piston speed would be limited to 5 m/s, and that a hydrogen Stirling engine operating at 200 atmospheres and 725°C would require an expander displacement of approximately .3 cm^3/J. The resulting relationship between displacement, engine speed and power level is shown in Figure 5-22. This indicates that at a per cylinder power level of 100 kW, the necessary displacement is approximately 15 cm^3/kW at an engine speed of 1200 rpm. This figure also illustrates the strong effect of power level on specific displacement and engine speed. Here, specific displacement varies as the square root of engine power, while engine speed varies inversely as the square root of power.

The importance of dead volume is illustrated in Figure 5-23. Here the ideal power ratio, expressed as the work per cycle divided by the product of peak pressure and total swept volume, is seen to decrease roughly linearly with the ratio of dead volume to swept volume. For reasonable specific power, the total dead volume must not be significantly larger than the order of the swept volume. For example, current Philips and United Stirling engines have a dead volume about 1.9 times the swept volume. Since the dead volume in the heater head is but a portion



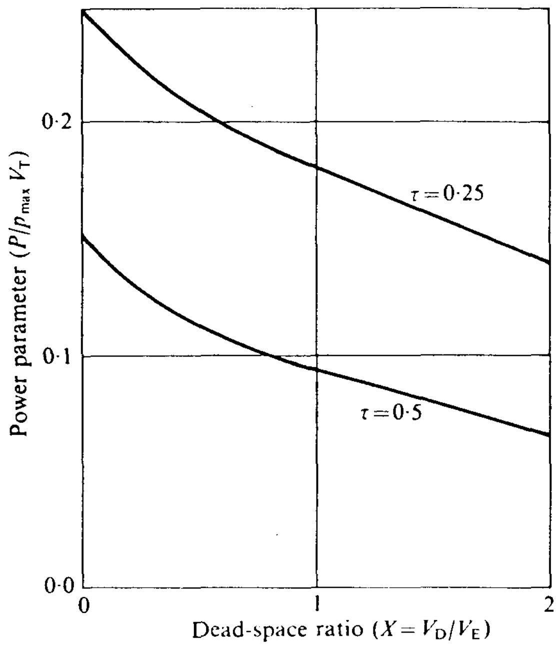
HEATER AIR PUMPING POWER VERSUS HEATER DEAD VOLUME

Figure 5-21



EXPANDER DISPLACEMENT AND ENGINE SPEED VERSUS ENGINE POWER. ASSUMES MEAN PISTON SPEED = 5 M/S; V_E/P = 3×10^{-7} m^3/J.

Figure 5-22



Effect of the dead-space ratio X on cycle power. The figure shows the effect on the non-dimensional power parameter $P/(p_{max} V_T)$ of variation in the dead-space ratio X, with constant values of τ, κ, and α. The dead space is the porous volume of the regenerator and other heat-exchangers, connecting ducts, and the clearance volumes in the expansion and compression spaces. Increase in the dead volume decreases the ratio of maximum volume to minimum volume which decreases the range of the pressure excursion, thus causing a decrease in the cycle power. The dead space *must* be minimized for high cycle-power.

(Walker, Ref. 36)

Figure 5-23

of the total system dead volume, it is clear that the heater volume must be kept of the same order as the expander swept volume. Since the cylinder displacement at the power range of interest will be of the order of 15 cm^3/kW, it is clear that heater volumes of the order of 20-50 cm^3/kW impose an undue penalty on engine performance.

Referring again to Figure 5-21, for a given heat exchanger surface, dead volume can be reduced at the expense of increased air pumping power. However, even with relatively small diameter tubing, both tube fouling and practical limitations to the allowable fan power would not permit the dead volume to be reduced below about 10 cm^3/kW. However, through the use of extended surfaces on the outside of the tubing, the dead volume can be reduced significantly below this figure without the expenditure of additional fan power.

In conclusion, when the heater head is heated directly with combustion gases, it is necessary to allow the combustion blower to expend a few percent of engine output (e.g., 3-5%), in order to restrict the heater dead volume to an acceptable value. When burning dirty fuels, the fouling of heater tube surfaces can impose further limits on the external heat transfer making it necessary to increase dead volume beyond the values used with clean fuels, thereby reducing engine power and efficiency. The use of indirect heating schemes can circumvent this problem for "fouling" fuels (see Section 5.3.2).

5.3.1.2 Combustion Chamber

The important technical requirements for combustion systems to be used for direct heating of a Stirling engine are as follows:

1. The system must be capable of handling combustion air pre-heated to a temperature of the order of 700°C.

2. The combustion system must be capable of producing low emissions of oxides of nitrogen, whose production is unfortunately enhanced by the high pre-heat temperature.

3. The products of combustion should not cause excessive fouling of the heater surfaces.

4. The combustion system must be capable of interfacing with a compact heater in close proximity to the engine cylinders.

5. The combustion system should provide relatively uniform discharge temperature and should be relatively controllable in order to maintain constancy of heater temperature.

Since the approaches for meeting the above requirements differ markedly for liquid and gaseous fuels versus solid fuels, these will be discussed separately.

Liquid and Gaseous Fueled Combustors

The combustion of gaseous or distilate fuels in a Stirling engine is not unlike that in regenerative gas turbines. Because the combustion air is pre-heated, pre-combustion temperatures are of the order of 500-700°C. Although this high pre-heat temperature enables rapid fuel evaporation and combustion, it also greatly accelerates the rate of formation of oxides of nitrogen. Therefore, measures must be taken to otherwise limit NO_x formation.

A liquid fueled combustor generally utilizes mechanical or pneumatic atomization of the fuel, which is usually mixed with 100%

primary air. Intense mixing and high pre-heat air temperatures make it possible to reach combustion intensities of the order of 5 x 10^6 Btu/ft^3-hr (5 x 10^4 kW/m^3). The high combustion intensity enables the use of a relatively small primary combustion chamber in which the walls are air-cooled by incoming air, and a larger secondary combustion chamber which utilizes the heater tubes to form part of the combustion chamber "wall" (see Figure 5-24). This arrangement not only minimizes the need for refractory combustion chamber lining, but also serves to limit the formation of oxides of nitrogen. The high pre-heat temperature creates combustion temperatures of the order of 2000-2100°C, versus about 1700°C which would occur if there were no pre-heat. At these temperatures, and for a short time at temperature, NO_x forms at a rate roughly linear with time, and theoretically as an exponential function of temperature. However, since the residence time in the combustion chamber is inversely proportional to combustion intensity, the high combustion intensity helps to limit NO_x formation. Measurements show that air pre-heat increases NO_x by a factor of about 5 to 6 (see Figure 5-25).

Even with the short combustion chamber residence times (of the order of 5-10 milliseconds), additional measures must be taken to reduce NO_x. Both exhaust gas recirculation (EGR) and operation with high excess air have been used to reduce NO_x. EGR reduces NO_x both by lowering the adiabatic flame temperature and by reducing the oxygen concentration. High excess air also works by lowering the adiabatic flame temperature, but since it increases the oxygen concentration, it is not as effective as EGR. Both high excess air and EGR tend to reduce the thermal performance somewhat, and increase the necessary combustion air blower power. However, unlike operation at high excess air, EGR does not reduce the performance of the air pre-heater, and so does not contribute as much to the burner exhaust loss.

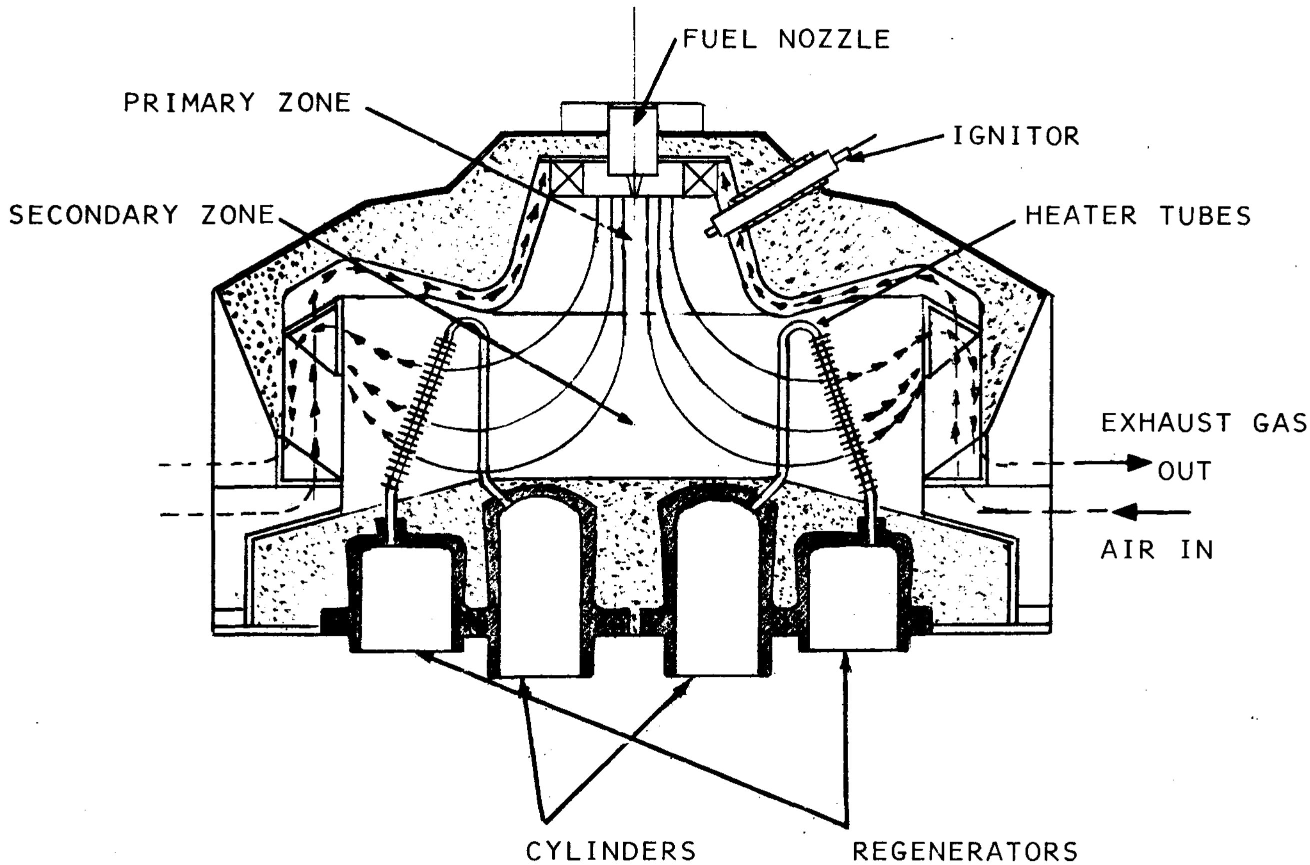


Figure 5-24 Liquid Fuel Combustion Chamber (courtesy United Stirling)

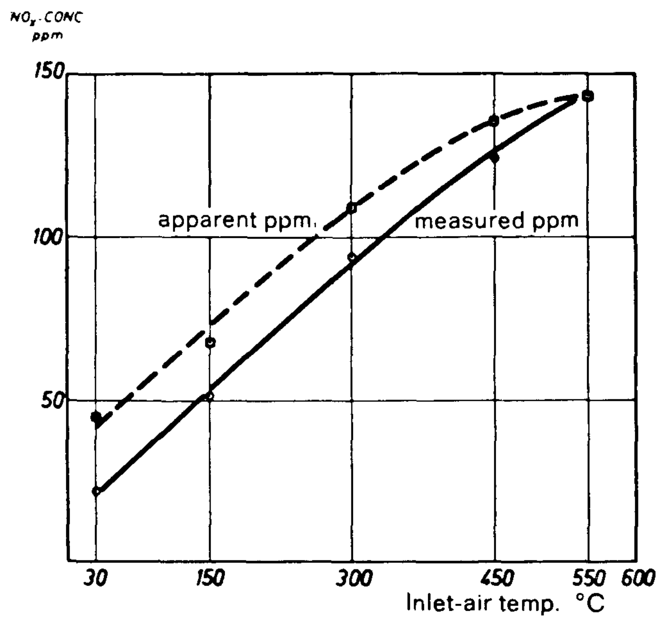


Figure 5-25 Experimentally Found Values for the Nitrogen Oxide Content in the Exhaust Gases as a Function of the Burner-Air Temp. (Ref. 12)

The emissions of other species (unburned hydrocarbons and carbon monoxide) are extremely low in well-designed combustors, and do not represent a problem. Typical emissions for combustors fired with distillate fuel are shown in Table 4-2.

A possible alternative to heterogeneous (droplet-type, non-premixed) combustion is to operate the burner with a homogeneous (premixed air and fuel) mixture, as is being done in low NO_x gas turbine combustors (Refs. 41, 42, 44). This type of combustion has the advantage of lower emissions, especially NO_x, and it permits higher combustion intensities. The main difficulty with this approach is the high pre-heat temperature of the air, which is above the ignition temperature of hydrocarbon fuels. However, even at temperatures of the order of 800°C, the ignition delay is greater than 10 milliseconds, which is more than enough time to enable fuel mixing and vaporization prior to combustion without pre-ignition. A further advantage of homogeneous combustion is that since combustion occurs as soon as the mixture reaches ignition temperature, there is no opportunity for the fuel to be exposed to high temperatures in the absence of oxygen, which would lead to polymerization and cracking, and result in the formation of soot. In this regard, homogeneous combustion of oil may be compared to the burning of natural gas. This quality can be quite important where compact heat exchangers rely on clean heat transfer surfaces to maintain their high performance.

Combustor controls must maintain proper air/fuel ratio for stable combustion and low emissions, and adjust the rate of combustion to maintain constant heater-head temperature. Unlike the internal combustion engine, power is not controlled by varying fuel input, but by modifying the mean working gas pressure within the engine (see Section 5.2.7). Generally, the best approach is to meter the combustion air flow rate through an accurate, variable-area metering device. The temperature of the heater-head is then used in a feedback control loop to govern the quantity of combustion air. Fuel flow is slaved to air flow according to a fixed schedule to maintain proper air/fuel ratio. The control technology is quite similar to and has benefited from automotive fuel injection systems.

Corrosion of heat transfer surfaces can be experienced with gaseous or distillate fuels, but this problem is not nearly as serious as with solid fuels or residual fuel oil. The only problem with distillate fuels is occasional SO_2 corrosion in the pre-heater when exhaust temperatures are allowed to get too low (see Section 5.3.1.5). Corrosion from solid fuel exhaust products is covered in more detail in Section 5.3.2, Indirect Heating.

An attractive alternative fuel for use in Stirling engines is low Btu gas. Here, the high air pre-heat temperature can be used to advantage to obtain stable combustion with the low enthalpy rise afforded by low Btu gas. This factor should become of increasing advantage to Stirling engines as the natural gas and petroleum supply becomes more critical.

In summary, liquid or gaseous fuel combustion systems for direct heating of Stirling engines permit the use of compact, clean, easily controlled, and relatively inexpensive combustors, which can be readily integrated with a Stirling engine to provide a compact package capable of low emissions. Of course, these very qualities are responsible for the strong demand of these fuels for many other purposes, and were it not for the critical supply of these fuels, they would be the obvious choice for use in the Stirling engine.

5.3.1.3 Combustion Air Supply

The combustion air supply can have an important influence on overall engine efficiency since the power necessary to drive the combustion air blower can be up to several percent of the engine gross output (see, for example, Figure 5-21). In automotive Stirling engines, combustion air is supplied at a pressure of approximately 20" H_2O (5000 Pa), which results in shaft power at the blower being approximately 3 % of engine output. The blower typically consists of a radial vane or backward leaning centrifugal blower, driven from the engine output shaft through an overrunning clutch. An electric motor is used to drive the blower for 15-30 seconds during start-up.

The power to the blower must be efficiently controlled if excessive drive losses are to be avoided, especially during part load operation. Modulation of combustion air can be accomplished by several methods: variable-speed blowers; multi-speed; constant-speed with variable guide-vanes; and constant-speed with dampers. Parasitic power loss is shown in Figure 5-26 for several control methods. For reference, we have assumed that at full load the combustion air blower power amounts to 1% of the engine gross output. With a variable-speed drive, the blower power will vary between the square and cube of engine power, with the result that the parasitic loss as a percentage of gross output decreases with diminishing load. If a fixed blower speed is used with air flow being modulated by throttling, then the combustion air blower power remains approximately constant, and becomes an increasing fraction of engine output at part load. This characteristic would obviously impose a severe penalty in an engine that must operate over a wide power range. A multi-speed blower with throttling can reduce this penalty substantially, and may be quite acceptable for engines which operate at moderate to high loads most of the time. The power associated with a

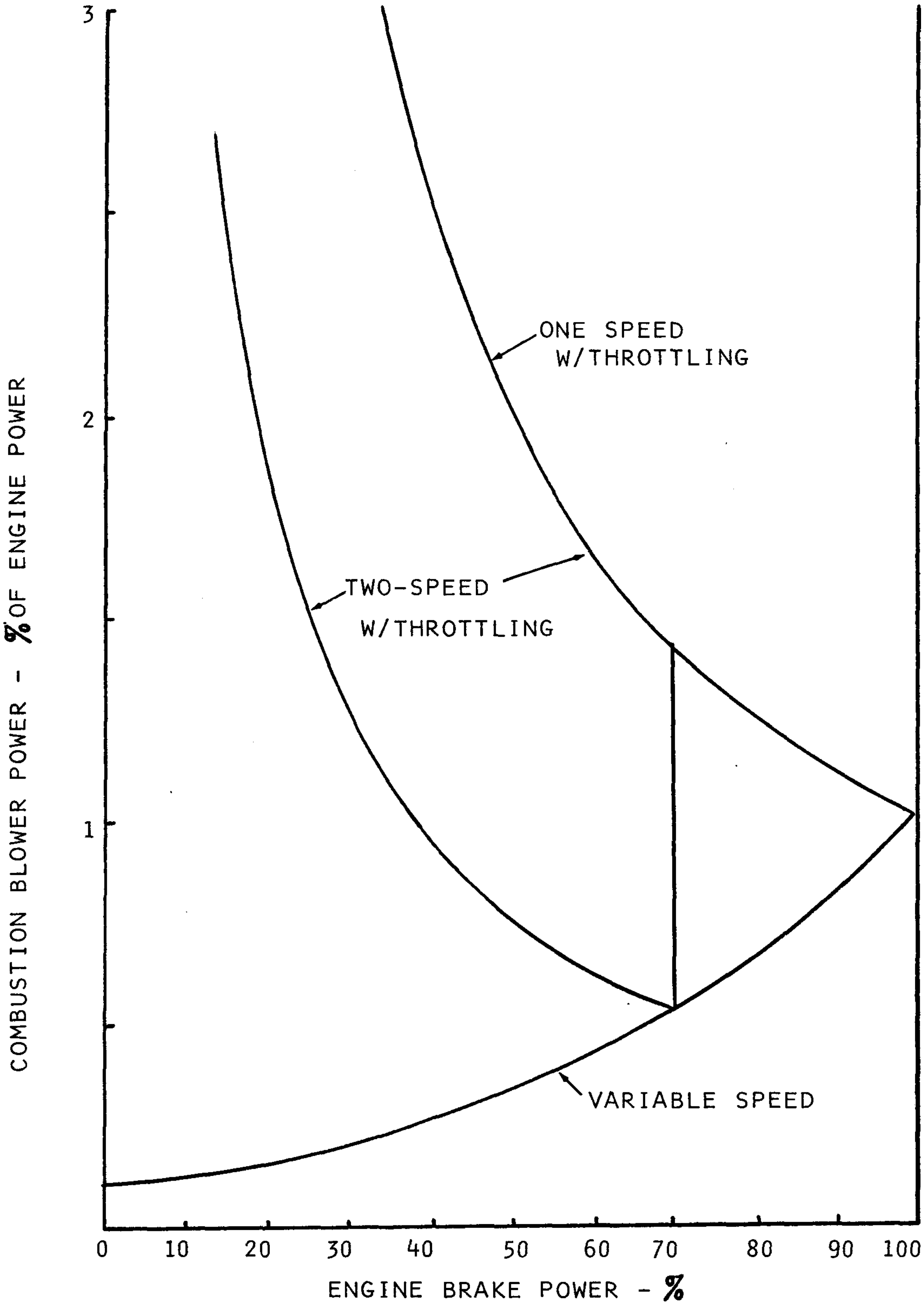


Figure 5-26 Combustion Air Blower Power Requirements

constant speed drive with variable inlet guide-vanes is not shown in Figure 5-26, but would fall in between the fixed and multi-speed drives with throttling at low to moderate power levels, and below them at high power levels.

In summary, in applications where the engine is to be base-loaded, modulation of the combustion air supply can be relatively simple without incurring a significant parasitic power penalty. In cases where the engine must operate for a considerable portion of its duty cycle at low power levels, then a more elaborate method of modulating the air supply must be employed in order to avoid significant efficiency penalties. Since specific combustion air blower power appears to increase as engine size increases, this may be an important concern in large industrial engines (see Appendix D).

5.3.1.4 Supercharged Combustion

A large Stirling engine can be fitted with a turbo supercharger system which pressurizes the combustor, while providing circulation of combustion air. This makes a combined cycle plant with the Stirling engine in effect topping a Brayton cycle as shown schematically in Figure 5-27. In this arrangement, the cold combustion air is compressed in the turbo compressor and then heated in the regenerative/recuperative air pre-heater. Combustion and heat transfer to the heater tubes is at the elevated pressure. Hot exhaust gases from the heater are expanded in the turbo expander and then further cooled in the air pre-heater.

There are two reasons for considering this turbo charged system. First, the overall efficiency may be improved since the stack loss of the Stirling engine burner is converted into heat rejection for the Brayton cycle. Second, the pressurized combustion reduces the volume of the combustor and increases the heat transfer coefficient between the combustion gases and the heater tubes. This improved heat transfer allows improvements in the Stirling engine through reduction in the dead volume, as will be discussed in a later section on heat-pipes.

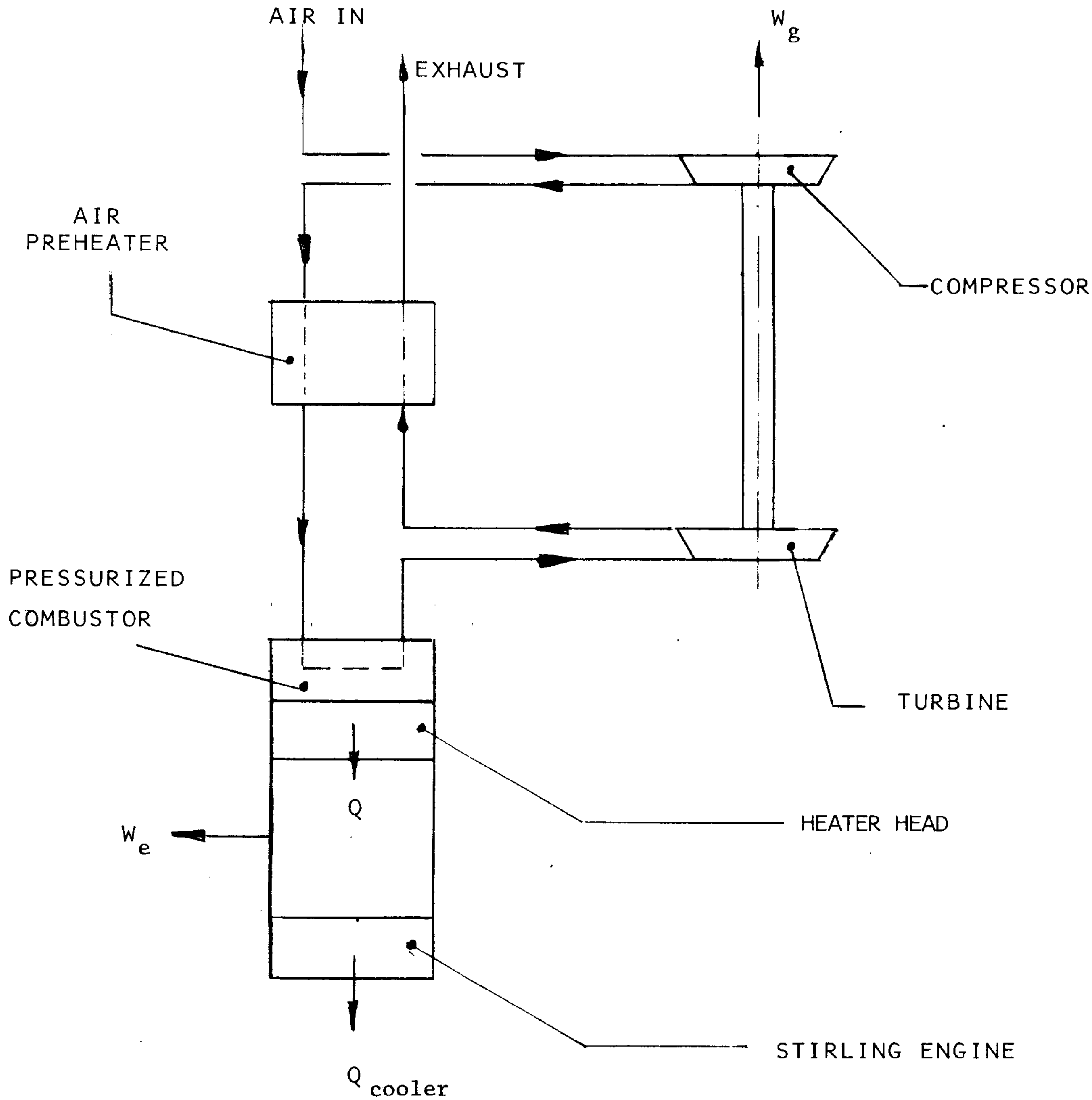


Figure 5-27 Schematic of a Stirling Engine with Turbo-Charged Combustor

The potential for an efficiency increase with turbo charging exists because there is a lower limit on stack gas temperature at the exit from the air pre-heater. If the temperature falls below the dew point temperature, the resulting condensate corrodes and fouls the cold end of the pre-heater. Acceptable exhaust temperatures vary with the fuel and with the level and nature of fuel impurities. For example, 200°F is acceptable for clean natural gas, while 350°F may be required for high-sulfur heavy oil. In the simple pre-heater burner system, stack temperature is determined by air pre-heater effectiveness, heater temperature and ambient air temperature. The minimum stack temperature then imposes a maximum air pre-heater effectiveness. The minimum stack energy loss, L_s, is thus fixed by the dew point of the combustion gases. The degradation of the Stirling engine efficiency η_E to overal efficiency - η , is thus:

$$\eta = \frac{\eta_E Q_E}{Q_E + L_s} = \frac{\eta_E}{1 + L_s / Q_E}$$

Where Q_E is the heat from the burner to the engine.

With the turbo charged combustor, the compressor should compress the air sufficiently to reach minimum stack temperature less the air heater temperature difference. In this case, the overall efficiency becomes:

$$\eta = \frac{\eta_E Q_E + W_g}{Q_E + L_s + W_g}$$

Where W_g is the net shaft work produced by the turbo charger. The turbo charger can be expected to increase the net shaft power by about 4%, while increasing the fuel requirement only about 1.3%. This could increase the overall efficiency about 1 percentage point, i.e., from 33% to 34%. The cost of this improvement is first, the turbo charger; and second, a larger air pre-heater. The turbo charged system requires a heat exchanger effectiveness of 0.90, whereas the atmospheric pressure system requires an effectiveness of only 0.80.

5.3.1.5 Pre-Heater

The air pre-heater is the heat exchange component which regeneratively heats the combustion air with the exhaust gases leaving the heater head. Without effective regenerative air pre-heating, the furnace efficiency for the external combustion system would be at best on the order of 55-60%, since the engine cools the combustion gases to only about 1500°F (815°C). With effective heat exchange in the air pre-heater, stack losses are reduced and furnace efficiencies of 80-90% can be achieved. As previously discussed, the improvement in furnace efficiency is limited by the dew point of the exhaust gases.

Effective air pre-heating can be accomplished in either a steady-flow heat exchanger functioning as a recuperator, or in a periodic-flow heat exchanger functioning as a heat storage regenerator. The recuperative heat exchanger requires separate headering for the air passages and the exhaust gas passages. A large number (order of several hundred) of small hydraulic diameter (order of 1/8") passages must be headered in parallel in order to achieve effective and compact heat exchange. The exchanger must be arranged for counter-flow or cross-counter-flow (5 or more cross-passes) in order to achieve the required effectiveness in the range of 0.8 to 0.9. The design of recuperative air heaters is dominated by considerations of fabrication costs, especially the headering for the two separate flows. Typically the exchanger and headers are formed from metal sheet stock and joined by high temperature brazing. The low pressure difference between the two flows (less than 0.1 atmosphere) allows the use of heat transfer surfaces made from flat sheet stock in contrast to high-pressure-difference exchangers which require more expensive tubular surfaces (for example, the heater and cooler where pressure differences are the order of 200 atmospheres).

The volume required for the air pre-heater core is fixed largely by the hydraulic diameter of the flow channels, once the overall thermal and pressure loss performance has been specified. As the channel diameter is decreased to achieve a smaller core, the number of channels must be increased and the length of the channels must be decreased. The increased number of channels increases fabrication costs. The smallest practical channel dimension may also be limited by fouling and the minimum channel length may be limited by axial conduction along the channel wall in the flow direction. Also, making the core of the pre-heater smaller does not decrease the required volume of the headers, but in fact, increases the volume since the mass flow per unit of frontal area is decreased.

Early Stirling engines used pre-heaters fabricated from tubular surfaces, but these were found to be too expensive and subject to failure by cyclic thermal stresses. A study of lower cost pre-heater designs (Ref. 33, pp 34-42), led to the selection of the "accordion" type surface as the preferred arrangement for recuperative type pre-heaters. This type pre-heater is illustrated in Figure 5-28. The heat transfer surface consists of thin stainless steel sheet metal folded in "accordion" fashion and arranged so that the incoming combustion air flows between corrugations on one side of the sheet, while the exhaust gases flow in counter-current between corrugations on the opposite side of the sheet (Figure 5-28). Headering of the two streams is accomplished in a relatively simple manner as shown. With this configuration, all of the heat transfer surface is prime surface (no fins) thus, minimizing the total surface area required. Cost studies at General Motors in 1968 (Ref. 33, p 38) showed that the basic core material cost for this type pre-heater was $0.38/engine hp, versus $18.30/hp for a pre-heater using straight round tubes. These figures illustrate the importance of using pre-heater cores fabricated from sheet stock rather than tubes.

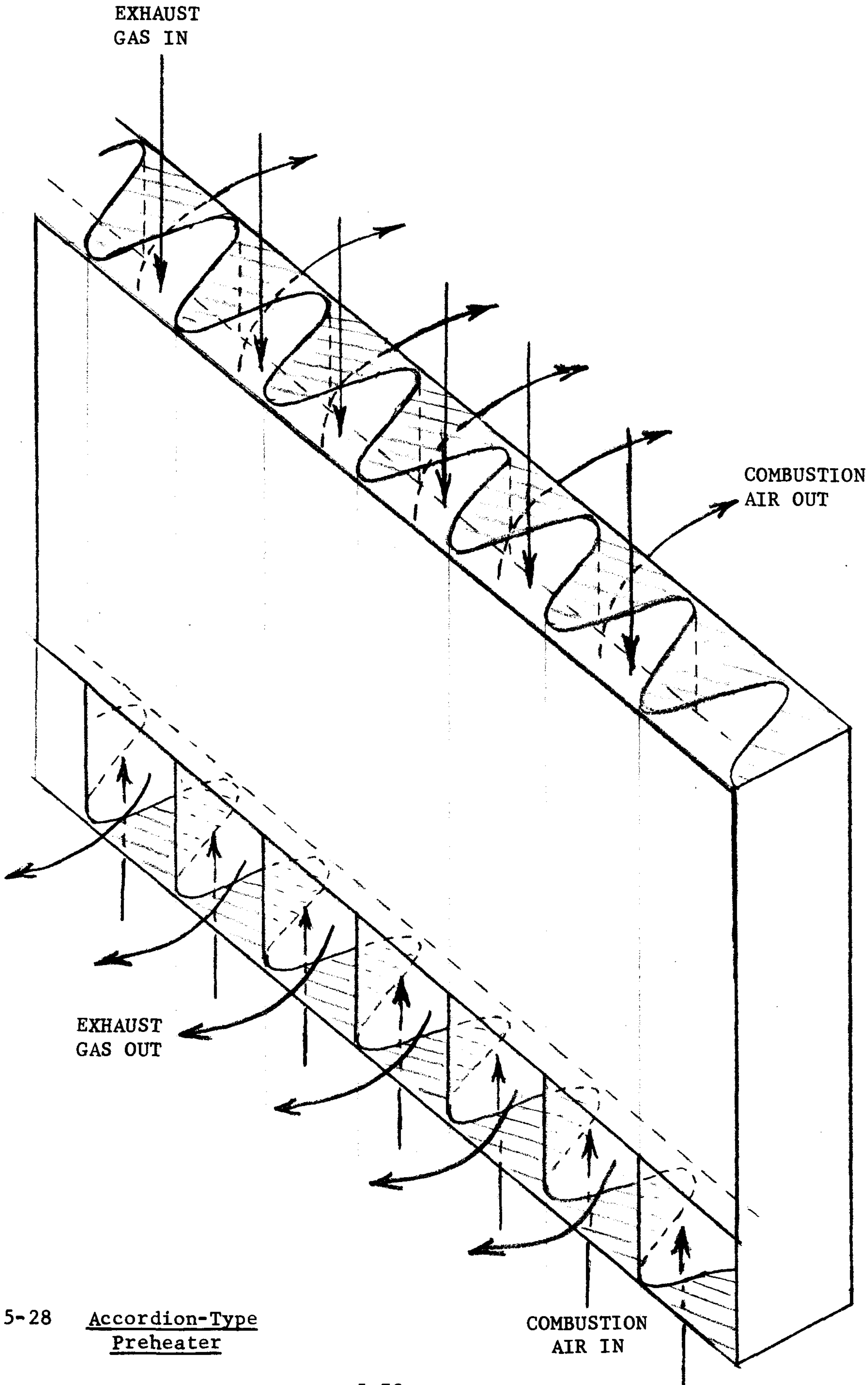


Figure 5-28 Accordion-Type Preheater

United Stirling and MAN-MWM both use this accordion type recuperative pre-heater in their current engine designs. Figure 5-4 illustrates a United Stirling design wherein the accordion sheet is arranged as a cylindrical element forming part of the exterior casing enclosing the combustion chamber and heater head. Note the short flow length for the pre-heater gas passages. This suggests that they employ rather small hydraulic diameter channels in the pre-heater core.

Effective air pre-heating can also be accomplished with a periodic flow thermal regenerator. The rotary regenerator used in the Ford-Philips engine (Figure 4-19) is similar to that developed for regenerative gas turbines. In this system, the regenerator packing or core is continuously rotated, passing periodically in turn through the hot exhaust gas stream and the cold combustion air stream. The heat from the hot gas is stored by increasing the temperature of the core and then carried mechanically to the cold air stream. The main advantage of the rotary regenerator is that the small flow passages of the core do not require individual headering, so that any porous solid can be used for the core. With this simplification in construction, pre-heater cores with flow passage diameters down to 0.020" are practical. However, to retain reasonable flow losses the passages become short with steep temperature gradients in the streamwise direction so core materials must have low thermal conductivity. The most successful regenerator cores are of glass-ceramic material, such as Cer-Vit by Owens-Illinois and Cercor by Corning. These materials have conductivities on the order of 0.4 Btu/hr-ft-°F, whereas stainless steel has a conductivity of 15 Btu/hr-ft-°F.

With the rotary regenerator for air pre-heating, there are two losses that are not present with a recuperator. The first is gas leakage between the exhaust gases and the combustion air as the result of the gas volume carried in the core and the leakage past the seals where the core passes between the streams. This loss is not important for the air pre-heater which operates with both streams nearly at atmospheric pressure. The second loss results from the temperature change of the core required to store the heat absorbed from the hot gas stream. The magnitude of this loss is limited by increasing the rate of rotation of the rotary core of the regenerator. The MC_p of the gas passing through the regenerator (sum of mass of both streams) in one revolution should be about 10% of the MC of the regenerator core.

The rotary regenerator also requires a slow speed drive to rotate the regenerator core at a few rpm, and sliding or close clearance seals where the core passes between gas streams.

The rotary regenerator has not been popular as the air pre-heater for current state-of-the-art Stirling engines, in spite of the potential for more compact cores and lower base material costs. Only Philips has favored this type regenerator for use in the automotive engine for Ford (Figure 4-19). Problems with rotary pre-heaters have included cracking or degradation of the cores, high seal wear and excessive leakage.

5.3.2 Indirect Heating

When the Stirling engine heat source is obtained by combustion of solid fuels (coal), residual oil, or other types of heavy fouling fuels, the low allowable combustion gas heat transfer coefficient (see Section 5.3.1.1), makes it desirable to utilize an indirect heating scheme whereby the heater tube heat transfer process is decoupled from the combustion gases. Before discussing solid fuel combustion systems, it is appropriate to examine the methods available for transporting heat from the combustion gases to the heater tubes.

5.3.2.1 Heat Transport Systems

Two methods of achieving indirect heating are through the use of a liquid metal heat-pipe and through forced convection of a single phase medium, such as a high pressure, low molecular weight gas.

Heat Pipes

As applied to Stirling engines, the heat-pipe can be advantageously employed in two main areas: (1) in coal-fired engines, the heat-pipe may be the only practical method of obtaining the high heat flux densities needed at the heater-head while absorbing heat from the coal at the relatively low flux densities necessitated by the fouling tendency of the coal; and (2) in engines fired with "clean" fuels, the interposition of a heat-pipe between the direct-fired heat exchanger and the heater head removes certain geometric constraints and provides higher heat fluxes to the heater tubes than if the heater were fired directly, both of which enable greater power and efficiency to be achieved within a given engine (see following section, "Bonus" Effect).

At the operating temperatures of interest, the most suitable working fluids for a heat-pipe are the alkali metals (principally sodium), due to their low vapor pressure, high thermal stability, high surface tension, and excellent thermal transport properties. Sodium has excellent compatability with most of the metals used at high temperatures: stainless steels, Inconel, nickel, niobium-zirconium alloys, and most of the refractory metals. Properly designed liquid metal heat-pipes have operated for over 20,000 hours (Ref. 55). Philips (Ref. 24) has designed and tested heat-pipe systems for Stirling engines with capacities of several hundred kilowatts. United Stirling has also run engines with electrically heated sodium heat-pipes.

Due to its high wettability, high surface tension, and its very low absolute change of saturation pressure with temperature, pool boiling of sodium results in high wall superheat and explosive bubble formation, which makes this mechanism of heat transfer quite unsuitable for a heat-pipe. Instead, the evaporation in a sodium heat-pipe must be caused to occur at the liquid vapor interface (film evaporation), rather than at nucleation sites on a solid surface. This is achieved by covering the inside surfaces of the evaporator with a thin porous matrix, generally made up of fine wire gauze, so that liquid will be distributed uniformly over the evaporator surface, and so that the thickness of the liquid will not be so great as to permit nucleation of vapor within the liquid. Because of the necessity of preventing bubble nucleation, wicks must be used even if the heat-pipe is of the thermal siphon-type, in which condensate can be returned to the evaporator by gravity.

Because capillary forces must be relied upon for proper liquid distribution, surface cleanliness and purity of the working fluid is of the utmost importance. Not only must stringent precautions be observed during the manufacture of the heat-pipe, but if the heat-pipe is to have a long useful life, provisions must be made to minimize in-leakage and to trap contamination products. The importance of maintaining system purity suggests that the best approach for an industrial engine would be a hermetically sealed heat-pipe, as this would minimize the potential for contamination. On the other hand, a hermetically sealed system may place some limitations on the serviceability of the engine.

Typical heat fluxes to and from sodium heat-pipes are in the range of 100-400 kW/m^2 (Ref. 29, 31), the lower figure being typical of the heat flux to the evaporator in a fluidized bed coal combustor, and the higher figure being typical of heat-pipes fired with high temperature combustion products or of heat fluxes in the heat-pipe condenser. For comparison, assuming a reasonably high gas convection heat transfer coefficient of 500 w/m^2-°C, a gas-to-wall temperature difference of 800°C would be required to achieve a heat flux of 400 kW/m^2.

Among the problems associated with the heat-pipe are: cost, serviceability, life, safety, start-up and control. Cost can be a problem initially due to the additional heat transfer surface involved, but is compounded by the exacting manufacturing requirements and high material cost, both for the heat resistant alloys and for the wick materials.

Serviceability may be a problem either of the engine proper if a hermetically sealed heat-pipe is used, or of the heat-pipe itself if the heat-pipe is designed as a non-hermetic system. The potential safety hazard of a liquid metal heat-pipe is associated with the high affinity of alkali metals for water and the resulting potential for explosion and fire. It is estimated that about 10 g of sodium per kilowatt are required in a small engine (Ref. 31), so a 1000 kW engine should contain about 10 kg of sodium, if such an extrapolation is valid. With regard to safety, a hermetically sealed heat-pipe might be safer than one designed for serviceability. If compatible materials of construction are used, the limiting factor determining service life is contamination, which is aggravated by the sub-atmospheric pressure of sodium which allows in-leakage, and which may be difficult to control in a hermetically sealed heat-pipe.

Because sodium freezes at a temperature of 97.5°C, care must be taken during start-up to assure that all parts of the heat-pipe are above the melting point so that the liquid return path cannot be blocked. Furthermore, at temperatures below about 400°C, the sodium vapor is in the free molecule flow regime, which both limits the allowable heat flux and causes substantial axial variations in vapor temperature (Ref. 25). Since high start-up temperature gradients can create excessive thermal stresses, the heat input during start-up must be limited. Starting times in experimental engines are as long as 1 hour (Ref. 25), but these long times should not be taken as typical.

The dynamic response of the heat-pipe is not very fast due to its heat storage capacity, with characteristic times of the order of several seconds. However, it is felt that with the proper type of load-following control, this need not be a problem.

Single-Phase Heat Transport System

An alternative to liquid metal heat-pipes is to use forced circulation of a high pressure, low molecular weight gas such as helium or hydrogen. Both helium and hydrogen possess high specific heat and high thermal transport properties, which enable a high heat transport rate with reasonably low pumping power. Although hydrogen possesses better thermal transport properties, we consider helium to be the more attractive medium because of the lower safety hazard, especially considering the relatively large amount of high pressure gas that would be involved.

A single-phase indirect heating system would consist of a direct-fired high pressure helium heat exchanger in which helium at a pressure of approximately 100 atmospheres would be circulated by a low pressure-ratio circulator. High helium pressure is necessary since the pumping power varies inversely as the square of the pressure. The engine heater head would be a shell-and-tube type heat exchanger with the engine working fluid inside the tubes, and the helium as the shell-side fluid. The major new technology component would be the high-pressure circulator, as this would have to operate at a temperature of approximately 750°C. However, because of the low pressure ratio (of the order of 2×10^{-4}), the circulator would be a rather simple, but high speed, single-stage compressor, and should present few of the problems associated with similar rotating machinery being developed for high-temperature closed-cycle energy conversion systems.

A preliminary calculation has been carried out for a 100 atmospheres helium system suitable for a 1000 hp engine. Some of the pertinent parameters are shown in Table 5-2. Although this design is by no means optimized, it does indicate that the design parameters are reasonable, and that the pumping power would be acceptable. The analysis shows that the pumping power would be of the order of 1% of the gross engine output, which may be more than compensated for by the bonus effect (see following section).

In summary, either a conventional or a single-phase heat-pipe can be employed as an indirect heating medium for the Stirling engine. Both methods enable the design of the engine to be effectively decoupled from its heat source, both in a thermodynamic and geometric sense. Thus, they would permit conventional solid fuel combustion systems to provide heat for the engine, and could enable an industrial Stirling engine to be developed which could utilize a wide range of heat input sources without requiring the engine to be custom-designed for each specific fuel.

TABLE 5-2

HELIUM HEAT TRANSPORT DESIGN PARAMETERS

Engine Power Output	750 kW
Helium Working Pressure	100 atm
Stirling Working Fluid Temperature	700°C
Helium Temperature	750°C
Helium Temperature Range	37.5°C
Helium Flow Rate	34,600 kg/h
Heater Head Pressure Drop	2700 Pa
Heater Head Flow Cross-Section	0.14 m^2
Heat Transfer Coefficient	3500 W/m^2-°K
Theoretical Pumping Power in Heater Head	5.5 kW

Heat-Pipe "Bonus Effect"

Ordinarily, one would expect that when an indirect heating medium is used instead of direct heating, thermal performance will suffer due to the additional resistance to heat transfer. Paradoxically, the heat-pipe can be employed in a Stirling engine in such a way as to improve both the specific power and efficiency of the engine (Ref. 16). This "bonus effect" is due to the fact that the design of the heat input sub-system, whether the heat source is from combustion gases or some other source, can be largely decoupled from the design of the Stirling heater head. This separation of design requirements enables several benefits to accrue to the advantage of engine performance:

1. The design of direct-fired heater head tubes must contend with a widely varying gas temperature from the combustion gas inlet to the outlet, while at the same time assuring that the safe working temperature of the tubes is not exceeded. As a result, the average heater tube temperature is substantially below the "hot spot" temperature which is governed by material creep properties. By contrast, the heat-pipe system operates with nearly constant heater tube wall temperature (no hot spots) due to the uniform sodium vapor temperature and its extremely high condensing heat transfer coefficient, relative to the wall conductance and the inside coefficient. Hence, with the heat-pipe the average heater tube wall temperature can be boosted to the creep temperature limit, an increase of about 50-75°C above that for direct-fired systems, thereby improving engine output and efficiency substantially.

2. Due to its high condensing heat transfer coefficient, the sodium heat-pipe allows the heater tubes to operate at substantially higher heat flux densities than are possible with direct combustion gas heating. Therefore, less heater tube surface area is required, resulting in lower dead volume and hence, higher engine specific power and efficiency.

3. Since the heat-pipe decouples the surface area at the burner from the surface area at the heater head, additional heat transfer surface can be provided at the burner so that the burner efficiency can be increased without increasing heater dead volume.

4. Because the heat transfer surface at the burner is no longer limited by dead volume considerations, it is possible to lower the flame temperature while still retaining a high burner efficiency; thus by reducing the flame temperature through operation with increased excess air or by exhaust gas recirculation, lower NO_x formation is achieved.

An example of the magnitude of the "bonus effect" is shown in Figure 5-29. This figure relates to high specific power double-acting engines optimally designed for direct heating with helium and hydrogen working gases. Also shown are characteristics of a helium engine optimized for a heat-pipe heater. The calculated performance for the heat-pipe engine is substantially better than for direct heating. Because of the high diffusivity of hydrogen, there has been a great reluctance to use hydrogen as the working fluid of an engine equipped with a sodium heat-pipe, as diffusion of hydrogen into the heat-pipe would adversely affect the

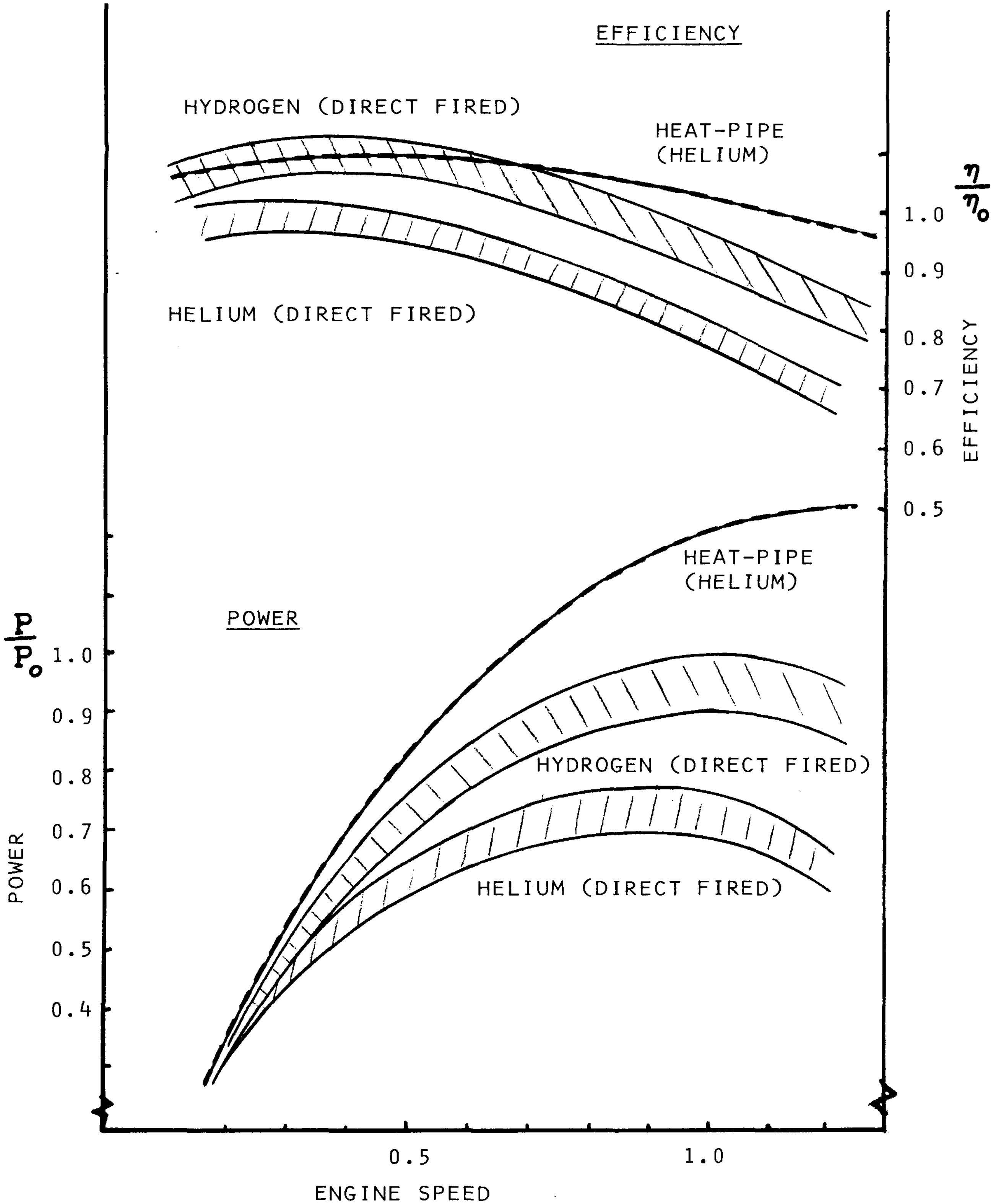


Figure 5-29 Power and Efficiency Comparisons; Heat-Pipe vs Direct Fired Engines. (courtesy United Stirling)

condensing heat transfer coefficient. For this reason, all of the heat-pipe equipped engines have used helium, and this trend is likely to continue unless effective hydrogen diffusion barriers are developed. Thus; a realistic appraisal of the bonus effect should compare direct-firing with hydrogen to indirect firing with helium. This comparison (Figure 5-29) shows that the helium heat-pipe engine specific power will be about 20% higher than the direct-fired hydrogen version, while the efficiency will also be slightly greater. Thus, there is still a worthwhile performance benefit to the heat-pipe heater system even though the working gas is changed to helium, and the relatively minor safety hazard associated with hydrogen working gas is exchanged for a similar risk with sodium.

A similar type of bonus effect, but differing in magnitude, is achieved by other methods of heat transfer enhancement, such as through the use of pressurized combustion or through indirect heating by forced circulation of a high-pressure, low-molecular weight gas, see Sections 5.3.1.4 and 5.3.2.1.

5.3.2.2 Solid Fuel Combustion Systems

One of the primary attributes of the Stirling engine is its ability to burn non-petroleum and non-gaseous fuels. Of special interest is its potential to utilize a solid fuel heat source, in particular coal. Practical experience in operating Stirling engines on coal or other solid fuels is very limited. Only Philips of Holland has ever experimented in this area, operating a 1/2 hp engine on charcoal for demonstration purposes and a 10 hp engine on powdered coal in a fluidized bed. In both cases heat was transported to the engines by a sodium heat-pipe. Running time for each test was less than 8 hours.

Since the Stirling engine is an external combustion engine, and the combustion process is separate from the work producing processes, it might appear that the development of a coal-fired Stirling engine would be a reasonably straightforward exercise of existing coal combustion technology. This is not the case, however, as a coal combustion system must satisfy several requirements before it can be a practical heat source for a Stirling engine. The primary requirements are as follows:

- Low emissions
- Low operating cost
- High heat flux capability
- High turndown ratio
- Fast response
- Readily integrable with heater head

It is highly desirable that the gaseous and particulate emissions be low, since in the size range of interest, it may not be practical to employ complex and expensive stack clean-up techniques. Operating costs, including costs of auxiliary power, operating man-power, and maintenance, will also have to be low if the engine is to be cost-competitive with more conventional power sources. Of course, it should be taken for granted that the emissions and operating cost of a coal-fired Stirling engine will be higher than for an equivalent oil or gas-fired engine, but these disadvantages must not be so great as to offset the advantage of being able to use the lower grade fuel.

The coal combustion system must be suitable for maintaining Stirling heater head temperatures in the neighborhood of 700°C. If the heater head is to be direct-fired, this means that furnace exit temperatures must be of the order of 1500°C or greater in order to provide sufficiently high heat transfer rates to require small heater head surface area and volume. This requirement may be relaxed if an intermediate heat transfer medium such as a heat-pipe is to be utilized. In either case, since the heat utilization temperature is so high, the combustion system must either have the ability to operate with high air pre-heat temperatures, or the exhaust products must be used in some other worthwhile manner, such as the generation of process steam. Finally, since in most cases, the engine will be used in other than base-load applications, it is necessary that the combustion system have a high turndown ratio and be capable of relatively fast response.

Current coal combustion technology is applied mainly to industrial and utility steam generation. Coal combustion systems may be categorized into four main types according to chronological order of development: stoker, pulverized, cyclone, and fluidized bed. While the state-of-the-art of the former three is well-developed, the technology of fluidized bed combustion is still developing. The principal attributes of these four types are summarized in Table 5-3. In the discussion which follows, it will be seen that while one type of combustion system may be preferred over another, no one system satisfies all of the requirements.

Stoker-Fired Burners

Stoker firing is a method of uniformly feeding coal onto a grate within the furnace and removing the ash residue after the fuel has been burned. The size and complexity of stokers varies considerably from small units suitable for residential boilers to large automatic utility and industrial stokers capable of refined control and high efficiency. Generally, however, stokers are used with small and moderate sized

TABLE 5-3

COMPARISON OF COAL COMBUSTION SYSTEMS

TYPE

	Stoker	Pulverized	Cyclone	Fluidized Bed
Typical Capacity	$<100\text{x}10^6$ Btu/hr ($<$30 MW_{th})	$>100\text{x}10^6$ Btu/hr ($>$30 MW_{th})	$>100\text{x}10^6$ Btu/hr ($>$30 MW_{th})	$3\text{x}10^6$ to $300\text{x}10^6$ Btu/hr (1-100 MW_{th})
Combustion Intensity Volume	---	20,000-40,000 $\frac{\text{Btu}}{\text{ft}^3\text{-Hr}}$ (Furnace)	500,000-900,000 $\frac{\text{Btu}}{\text{ft}^3\text{-Hr}}$ (Cyclone)	20,000-100,000 Btu/ft^3-Hr
Area	350,000 to 750,000 Btu/Hr per ft^2 of grate	50,000-100,000 Btu/hr per ft^2 of furnace area	---	100,000-1,000,000 Btu/Hr per ft^2 of cross-section
Turndown	5:1-10:1 Can be banked	3:1	2:1	Up to 5:1
Fly Ash	.25-.75 lb/1000 lb gas	50%-80% of all ash	5%-20% of all ash	Close to 100% of ash plus bed material
Unburned Combustible	Low to high, depending on type	.2-2%	$<$.1%	5%-15% without recycle or carbon burn-up; 1% to 3% after recycle
Excess Air	High	15%-22%	10%-15%	5%-15%
Air Preheat	250-450°F max.	Up to 600°F	Typically 600°F	400°F-600°F
Response	Slow start-up, reasonably fast load change	Fast	Fast	Slow start-up, fast load change
Control	Simple	Complex	Complex	---
Operation	Low Skill High manpower	High skill Low manpower	High skill Low manpower	High skill Low manpower
Fuel	All coals, wood, bark, waste, etc; coarse size	High fusion coal best; fine (pulverized) size	Low fusion, high-volatile coal, coal/waste mix; coarse size	All types of coal, char, waste; coarse size

TABLE 5-3 (Con't)

COMPARISON OF COAL COMBUSTION SYSTEMS

TYPE

	Stoker	Pulverized	Cyclone	Fluidized Bed
Furnace Exit Temperature	1800°F to 3000°F, generally limited by ash fusion temperature			1400°F-1600°F
Auxiliary Power Requirement	Low	High	High	High
Other	• Most suitable for smaller applications • Low erosion • May not require precipitators • Practical for single engine • Low air preheat must be used to prevent caking and grate damage • May be impractical for rapidly swinging loads	• High erosion, high maintenance costs • Large furnace volume • Generally must use precipitators • Probably unsuitable as heat source for single engine	• Can burn wide range of coals, waste • Lower maintenance cost than pulverized • May not require precipitators • Probably unsuitable as heat source for single engine	• Low SO_x, NO_x emissions • High particulate emissions • High heat transfer coefficients • Solids disposal about 3 times higher than conventional • Size suitable for single engine • Low gas temperature suggests use with heat-pipe • Economics not demonstrated, not yet commercial
(References)	(46)	(46)	(48)	(48)

boilers, especially where highly skilled manpower for operation is unavailable. Because of its applicability to small systems, its reasonably simple operation, and its ability to burn a wide range of solid fuels, the stoker is well suited for firing Stirling engines in the 500 - 2000 hp range.

The stoker can be applied either to a direct-fired or indirect-fired Stirling engine. In the direct-fired mode, the low fly ash content of the combustion gases is an important advantage. However, considering the importance of low fouling factors on the heater head of the Stirling engine, it is likely that some type of hot gas particulate removal system would be required. Hot gas clean-up would not be needed if a heat-pipe or other indirect heating method were used.

Another advantage of the stoker is that its low fly ash loading reduces the severity of erosion on heat transfer surfaces. This may be more important at the higher temperatures involved in the Stirling engine relative to steam boilers.

One limitation of stoker firing is that high air preheat temperatures cannot be used since they cause caking of the coal bed and warping of the moving grate. Depending on the type of coal, the preheat temperature can be no higher than about 250 to 450°F (Ref. 46, p 16-5). This means that some other method of recovering the heat of the high temperature gases exiting from the heater must be utilized. One attractive possibility is to utilize the high temperature exit gases to generate process steam in applications where this can be put to use. It is likely to be impractical to use the high temperature gases in a bottoming cycle, as this would both compromise the efficiency of the Stirling engine and is likely to be economically unattractive.

Other potential problems include the need for suitable methods of reducing gaseous emissions. Particulate emissions are quite low with stoker fired burners, and are amenable to conventional particulate removal techniques. Another limitation is that the start-up of a stoker fired burner is quite slow, and although its heat output can be changed quite rapidly, it is not practical to follow rapidly and continuously swinging loads.

Pulverized Coal Burners

Pulverized coal burners were developed to provide an economical and efficient method of firing coal in large utility boilers and large industrial boilers and furnaces. Although the pulverized coal burner requires a higher capital investment than for stoker fired burners, by virtue of their economy of scale they are less labor-intensive, and because they can operate with less excess air and higher pre-heat, they are more efficient. However, they become economically attractive only in sizes above about 100 x 10^6 Btu/hr (30 MW_{th}) (Ref. 46, p 17-1).

Pulverized coal may either be pre-processed and dried and stored in bins ready for firing, the so-called bin system; or it may be processed, dried and conveyed to the burners in a continuous process, the so-called direct-firing system. The bin system tends to be more expensive, and introduces problems of disposal of processing air if it is remote from the burner, and the increased hazard of bin explosions. Direct-firing systems tend to be less expensive, and the process is simpler and more efficient. In either case, pulverized coal firing requires an on-site or nearby coal processing plant which may be thought of as the mechanical analog of a chemical coal refining plant. The high capital investment required by this processing system generally rules out its application to small plants, and in particular, would rule out its application to the Stirling engine.

Cyclone Furnace

The cyclone furnace was developed as an alternative to pulverized coal firing to reduce the erosion and excessive particulate discharge caused by the high fly ash content of pulverized coal flue gases, to reduce the relatively large furnace volumes required by pulverized coal burners, and to provide an effective method of burning lower ranks and lower grades of coal which have high ash contents, low ash fusion temperatures or poor grindability.

A cyclone furnace is a water cooled horizontal cylinder lined with a refractory chrome ore. Crushed coal is fed into the furnace and combustion air is introduced tangentially to impart a swirling motion to the coal. Combustion occurs at temperatures exceeding 3000°F (1700°C) at high heat release rates of 500,000 to 900,000 Btu/ft^3-hr (5 to 9 MW/m^3). (Ref. 46, p 28-2). The ash in the coals melts to form a slag which coats and protects the walls of the cyclone. The incoming coal is thrown to the walls of the cyclone by centrifugal force, floats on the surface of the slag, and is scrubbed by the high velocity combustion air. This process enables combustion to be completed relatively quickly and completely. Most of the ash is retained as a liquid slag and is tapped into a tank below the furnace. This results in a very low quantity of fly ash of small particle size which produces very little erosion. Because of the high cyclone furnace combustion intensity and the lower dust loadings, the secondary radiant furnace can be smaller than in a pulverized coal furnace. Coal preparation power is less than that required for pulverized coal firing; but on the other hand, forced draft fan power is higher, with the net result being that the auxiliary power requirements for a cyclone furnace are comparable to those for a pulverized coal furnace (Ref. 46 p 28-7).

Generally, cyclone furnaces are built in capacities ranging from 100 x 10^6 to 500 x 10^6 Btu/hr (30 to 150 MW_{th}). While there does not appear to be a physical limitation to the size of a cyclone furnace, the cross sectional area of the furnace scales with capacity, so a small furnace would have a relatively large length-to-diameter ratio. For example, a 3 MW_{th} furnace would have a diameter of approximately 1.5 ft (.5m) and a length of approximately 10 ft (3m). However, due to its limited turndown capabilities and precise control requirements, it is generally conceded that the cyclone furnace is not suitable for such a small application.

Fluidized Bed Combustion

Fluidized bed combustion systems are a relatively recent development which has been spurred by the need to reduce gaseous emissions from coal combustion. The fluidized bed is a mixture of inert bed materials, coal particles, and an SO_2 sorbent, typically limestone or dolomite. The bed is fluidized by introducing pre-heated combustion air at the bottom of the bed at a superficial gas velocity of between 1 and 15 ft/sec (.3 to 5 m/sec) (Ref. 48, p 3-3). Heat transfer surfaces are located within the bed and above the bed to maintain the bed temperature in a range of 1400 to 1600°F (750 to 875°C) (Ref. 48, p 3-7). The SO_2 sorbent reduces SO_x emissions, and the low bed temperature lowers NO_x emissions. Because of the high entrainment of ash, unburned carbon and other bed material in the stack gases, cyclone-type separators must be used to recycle or otherwise contain particulate emissions. Because of unfavorable chemical conditions, it is believed that electrostatic precipitators may not be effective with fluidized bed burners. Fluidized bed combustors can be either atmospheric (AFBC) or pressurized (PFBC). PFBC systems offer the potential for increased sulfur removal and greater compactness. However, the pressurized system necessitates the efficient retrieval of compressor power through a hot gas expander, thus forming a combined cycle power plant. The authors believe that while the PFBC system may have merit for large power plants, the additional capital investment is unlikely to be justified for engines in the size range of interest.

The practical size range of fluidized bed combustors appears to be quite broad, but since the technology is still under development, it is not yet known whether the system will be commercially practical in small sizes. The turndown range is typically 3 to 1, but some units under development claim a range of 5 to 1 (Ref. 48, p 3-5). A principal advantage of the fluidized bed system is the high heat transfer coefficient available within the bed. Overall coefficients of heat transfer in fluidized bed boilers range from 40 to 80 Btu/hr-ft^2-°F (225 to 450 W/m^2-°C) (Ref. 48, p 5-17). Thus, one might envision direct heating of the Stirling engine within the fluidized bed without paying an appreciable penalty in specific power or efficiency due to increased void volume. However, because of the low gas temperature within the bed, heat transfer rates are too low for direct heating. Also, the geometrical problem of locating the heater head within the bed makes this impractical. If the engine cannot be heated directly within the bed, then the other alternative is to utilize an indirect heat transfer medium, such as a heat-pipe.

Philips (Ref. 16) has operated a 10 hp Stirling engine from a small experimental fluidized bed. The engine was coupled to the bed via a sodium heat-pipe. The bed operated at temperatures between 800 to 950°C and was designed to produce a minimum sodium temperature of 750°C. Aside from several problems owing to the experimental nature of the apparatus, especially associated with start-up, good results were achieved. It was found that the system was extremely responsive to load changes without the need for extremely complex controls. Bed temperature was controlled by the coal injection rate. Large step changes in load, of the order of about 5 kW, caused an instantaneous deviation of only 20°C at a bed temperature of 850°C. This good load following ability was believed to be due to the large heat capacity of the bed in conjunction with the rapid burning rate of the coal.

Philips' future approach is to utilize a regenerative air pre-heater to enable air pre-heat temperatures to the bed of 900°C or higher. This would enable full utilization of the high flue gas temperature, and would permit more rapid start-up. Furthermore, their approach would be to keep the bed hot at all times by a combination of banking and highly effective insulation.

Their experience suggests that the fluidized bed coupled to a Stirling engine via an indirect heating medium may be the most practical method of achieving a commercially acceptable coal-fired Stirling engine.

In summary, the fluidized bed combustor offers a very attractive method of firing a Stirling engine due to its compactness, low gaseous emissions, high heat transfer coefficients, and its ability to use high sulfur coals. Conversely, the technology is still under development, and some important questions regarding particulate emissions and waste disposal remain to be resolved.

Costs

Obviously, the costs of a solid fuel combustion system are significantly higher than for gaseous or liquid fuel firing. A commercially acceptable coal-fired Stirling engine will require a highly automated fuel handling and waste removal system, as well as an automatic and simply operated combustion system. It is not feasible to quantitatively estimate the costs of a solid fuel combustion system for an engine in the 500 - 2000 hp range, since most of the economic factors associated with coal combustion apply to much larger systems. Coal combustion tends to be highly capital intensive, and one can expect that the percentage of plant costs attributable to the combustion system will tend to be larger in a small plant. Roughly, it is estimated that the coal-fired Stirling engine would be 30% to 50% more expensive than an equivalent oil or gas-fired version. This premium is not unlike that encountered in large electric power plants, and may be an acceptable cost penalty as the cost differential increases between coal and the more precious fuels or if the burning of coal becomes more widely mandated.

5.3.2.3 Pre-Heater

The air pre-heater for combustion systems which employ indirect heating to the heater tubes is essentially the same as for systems with direct heating. Some of the problems are, however, more severe in the case of indirect heating. First, the hot end temperatures are somewhat higher because heat-pipe systems typically operate at higher average heater temperatures due to the elimination of hot spot problems. The second problem, corrosion and particulate fouling, is more serious and arises from the use of dirty fuels. Although the heat-pipe isolates the heater tubes from the contamination and uncertain temperatures encountered when burning dirty fuels, it does not, however, protect the air pre-heater from contamination, corrosion and particulate fouling, nor from the more non-uniform and variable temperature of the combustion gases. These factors tend to suggest the need for air pre-heaters of high temperature ceramic which is chemically stable and constructed with flow passages of sufficient diameter to avoid particulate plugging.

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APPENDIX A

ANALYSIS OF ENGINE SPEED SCALING FOR DIFFERENT WORKING GASES

The scaling of performance with working gas properties can also be shown without a detailed design of optimum engines. This is done by scaling the heat transfer components of an engine of constant displacement, V, at constant values of NTU, $\Delta p/p$, T, p, A_H, $D_{hydraulic}$ and $P_r = C_p \mu / k$. Since the losses which are related to the properties of the gas are fixed by the number of transfer units, NTU, and the pressure drop due to gas flow, Δp in the heat exchangers, this is equivalent to scaling the engine at constant efficiency and constant operating conditions, but variable operating speed. Constant T and p imply scaling with the same heater and cooler temperature and the same mean gas pressure. Constant heat transfer surface area A_H and constant hydraulic diameter, D, imply scaling at constant total heat transfer surface and constant dead volume in the heat exchange components.

The scaling is based on simple Nusselt number and friction factor correlations of the form:

$$\frac{hD}{k} \propto \left(\frac{GD}{\mu}\right)^{a}$$

$$f \propto \left(\frac{GD}{\mu}\right)^{-b}$$

where a and b are constants. The basic equations are:

$$NTU = \frac{h A_H}{\dot{m} c_p}$$

$$\frac{L}{D} = \frac{A_H}{4A_c}$$

$$\Delta p = 4f \frac{L}{D} \frac{G^2}{\rho}$$

$$\rho = \frac{p}{RT} = \frac{p}{T} \frac{M}{\overline{R}}$$

$$\dot{m} = GA_c \propto \frac{p\dot{V}}{RT} = \frac{p}{T} \frac{M}{\overline{R}} \dot{V}$$

where M is the molecular weight of the gas, $\dot{V}$ is the displacement rate of the engine , and $\overline{R}$ is the universal gas constant.

If L, G, A_c,h and f are eliminated by substitution,

$$\Delta p \propto \frac{1}{\rho} (NTU)^{\left[\frac{3-b}{a}\right]} (\mu)^{\left[-\frac{3-b}{a}+3\right]} (\dot{m})^{\left[\frac{3-b}{a}-1\right]}$$

where the constant factors except for Δp and NTU are not shown. The elimination of the flow passage length, L, and the frontal area of the flow passages, A_c, is equivalent to arranging the heat transfer components to have constant performance as expressed by NTU and Δp. When ρ and $\dot{m}$ are expressed in terms of M and $\dot{V}$.

$$\Delta p \propto (M)^{\left[\frac{3-b}{a}-2\right]} (\dot{V})^{\left[\frac{3-b}{a}-1\right]} (\mu)^{\left[-\frac{3-b}{a}+3\right]}$$

where M is the molecular weight of the gas and $\dot{V}$ is the displacement rate of the engine which is proportional to N•V where N is engine rpm. For constant flow losses Δp is proportional to p and is constant. Thus, the scaling of the engine speed with working gas molecular weight and viscosity is:

$$N \propto M^x \cdot \mu^y$$

$$x = -\frac{3-b-2a}{3-b-a}$$

$$y = \frac{3-b-3a}{3-b-a}$$

For flow in straight tubes the exponents a and b of the correlations are a = .8, b = .2 then:

$$N \propto M^{-0.6} \cdot \mu^{0.2}$$

The scaling of engine speed with working gas for the same thermal performance, displacement, and operating temperatures and pressure is shown in Table 1.

TABLE 1

Gas	M	μ at 200°F lbm/hr-ft	N_{H_2}/N Scaled	N_{H_2}/N Philips Optimized Engines
Air	29	.052	4.30	5.75
Steam	18	.031	3.58	--
He	4	.065	1.25	1.59
H_2	2	.025	1.0	1.0

The engine speed scaling shown in Table 1, is within about 25% of the fully optimized values taken from Figure 5-10 for an efficiency of 40%.

This simple analysis clearly shows that engine power density must be decreased as the molecular weight of the working gas is increased. The influence of variations of the thermal conductivity and specific heat of the gas with working gas do not appear in the scaling since these effects are combined in the Prandtl number $C_p \mu/k$ which is essentially the same for all of the gases.

The influence of the specific heat ratio $\gamma = C_p/C_v$ is not in this simple scaling, because only heat exchange components are considered. The differences in γ influence the nearly adiabatic compression and expansion processes in the hot space and in the cold space respectively.

APPENDIX B

SRI TEST REPORT ON HYDROGEN SAFETY

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HYDROGEN SAFETY OF THE STIRLING ENGINE

The Ford Motor Company has retained Stanford Research Institute to assess the safety of a Stirling automobile engine containing 30 grams of hydrogen as the working fluid. In a preliminary study started in October 1972, two conditions for which the hazard could not be established from existing information were identified: leakage of hydrogen from a parked vehicle in a closed garage, and leakage in collisions.

Subsequent tests performed by Stanford Research Institute in a closed garage showed that flammable hydrogen-air mixtures of a significant volume occur only for very high hydrogen release rates such as would result from catastrophic rupture of the connection to the hydrogen storage bottle. Moreover, for a catastrophic release of 30 grams of hydrogen in 9.5 seconds, it was found that for a spark ignition source under the hood, detonation occurred at an ignition delay of 5 seconds after release, but only fire resulted for all other ignition delays up to 19 seconds, after which no ignition occurred. For a spark ignition source above the hood, detonation occurred at an ignition delay of 5 seconds, but no ignition was obtained for a delay of 10 seconds or longer. In the two detonations that occurred, one fender skirt was slightly deformed. Thus, given the unlikely event of catastrophic rupture of the connection to the hydrogen bottle of a vehicle parked in a closed garage, an ignition source must be present within narrow limits of time and space if fire or detonation is to result; and if detonation does occur, damage to the vehicle is slight.

The maximum credible explosive hazard that could result from release of hydrogen from the Stirling engine in a collision was investigated using twenty-two grams of hydrogen mixed with 10 standard cubic feet of air in thin-walled rubber balloons within and under the engine compartment of a 1960 Ford automobile. Spark ignition of the mixture produced a detonation that buckled one strut under the hood and bowed the hood upward without breaking the hood latch. No other significant damage occurred and no effect of the explosion was visible in the passenger compartment.

Additional testing will be undertaken in which the shock overpressure in and near the vehicle will be measured for conditions that produce maximum explosive effects. The boundary to which flammable hydrogen-air mixtures extend will be determined, and comparisons will be made between the hazards of hydrogen and gasoline under similar conditions. Parallel tests will be conducted for hood designs with and without ventilating louvers.

APPENDIX C

THERMAL EFFICIENCY

The overall thermal efficiency of the Stirling cycle engine can be expressed as:

$$\eta_o = \frac{P\ net}{Ef}\ \text{(constant)}$$

where P net = net shaft power with all auxiliaries driven

Ef = fuel energy flow rate

= fuel mass flow rate x heating value per unit mass (lower heating value is assumed)

To arrive at η_o it is convenient to start with the so-called indicated or "bare" cycle efficiency, η_I, and then multiply it by ratios representing, (1) the mechanical efficiency η_M, (2) the external heating system efficiency η_H, and (3) the auxiliaries power factor η_A which is the ratio of P net to P gross, the latter being the power of the bare engine when all accessories are driven from an outside source. In summary, overall efficiency $\eta_o = \eta_I \times \eta_M \times \eta_H \times \eta_A$. Typical values for the three ratios are: η_M between .85 and .90; η_H between .85 - .90 for a fossil heat source; η_A between .85 - .95. The latter values cover a wider range mainly due to differences in engine application and the need of a radiator cooling fan -- a marine engine might achieve .95, whereas a truck in the summer only .85. For a heat source such as nuclear, isotopic, thermal storage, etc., η_H can be very close to 1.0.

Even before considering η_I we can look at the Carnot efficiency. This is the basic thermodynamic upper limit for all heat engines, regardless of the cycle, and is sometimes termed the Carnot law. It is equal to 100 × $\frac{T_1 - T_2}{T_1}$ where T_1 is the absolute temperature of heat addition (averaged if necessary where heat input is at a varying temperature) and T_2, the temperature of heat rejection. For "state-of-the-art" Stirling engines, T_1 ranges from 600°C to 800°C (+273°) while T_2 might vary from 20°C to 80°C, depending mostly on ambient conditions. The Carnot efficiency can therefore range from about 65 to 73%. In order to arrive at η_I, the Carnot efficiency must be multiplied by a factor which represents the "real" engine's capacity to achieve the Carnot "limits". This factor might be called the Carnot coefficient and varies between .65 and .75, depending on the general engine configuration, working gas, operating speed, heat exchanger design, dead space, and how well the engine has been optimized -- both by computer and construction details. For the future the Carnot coefficient is expected to increase gradually as the Stirling technology advances, possibly to .80 or higher.

If we assume a Carnot efficiency of 70% (T_1 = 720, T_2 = 30) and a coefficient of .70, then η_I is 49%. If $\eta_M = \eta_H = \eta_A = 0.90$, then η_o becomes equal to 35.7%. An overall efficiency of 35% has recently been achieved at United Stirling in their latest 4-cylinder double-acting engine of about 100 bhp, and is based on net power after driving all accessories including the combustion blower and radiator fan, at an ambient of 86°F. Two years ago efficiency was under 32%; the goal is 37% in the next two years.

By the end of 1978, efficiency running on fossil fuels is expected to be at least 36% based on net bhp with all accessories including radiator fan operating, for an ambient of 30°C. The effect of cooling water temperature on efficiency is shown in Figure 5-12. For engine applications requiring radiator cooling, accessories include the following:

- Combustion air blower
- Atomization air pump
- Gas compressor
- Fuel pump
- Lubricating oil pump
- Electric generator
- Radiator fan
- Cooling water pump

Power requirements for driving accessories varies from one application to another, with the speed of the engine and with matching conditions.

APPENDIX D

HEATER TUBE SCALING ANALYSIS

According to the scaling rules, the length and diameter of the heater tubes are proportional to the scaling factor, γ , but as will be seen below, it is necessary to scale the tube spacing approximately as $\gamma^{1/3}$. To a first approximation this means that the number and L/D of the tubes will remain approximately constant; and since piston speed is held constant, the flow losses inside the tubes do not change very much.

Considering the effects of scaling on the combustion gas side, it is seen that the heat transfer area will scale as γ^2, or proportional to bhp. Thus, if the heater head thermal effectiveness is to remain approximately constant, the combustion gas side heat transfer coefficient must also remain constant. In general, the heat transfer coefficient for flow over tube tanks is given by:

$$h = kG^m d^{m-1}$$

where G is the gas mass velocity, d is the tube diameter, k is a constant relating the fluid property effects, and m typically has a value of about .6*. Thus, if d scales with γ, and h is to remain approximately constant, the mass velocity must scale as $\gamma^{(1-m)/m}$, or as $\gamma^{2/3}$. Since the combustion gas flow rate scales with horsepower, or as γ^2, the flow cross sectional area must scale roughly as $\gamma^{4/3}$. For a fixed number of tubes whose length scales with γ, this requires that the tube spacing vary as $\gamma^{1/3}$.

Strict application of the foregoing rules would have the result that the combustion gas pressure drop would vary as $\gamma^{4/3}$, or as $bhp^{2/3}$. A detailed engine optimization would likely show that the scaling rules could be relaxed somewhat so that combustion air blower power need not increase

* Colburn, A. P., Transactions of the American Institute of Chemical Engineers, Vol. 29, pp 174-210 (1933).

as rapidly. Nevertheless, this approximate scaling argument indicates the tendency for specific combustion air blower power to increase roughly as the 2/3 power of cylinder horsepower in a direct-fired engine. Thus, practical limitations on auxiliary power must place an eventual upper bound on cylinder size. Considering the fact that Philips has designed cylinders as large as 225 bhp per cylinder with predicted efficiencies of the order of 40% at specific power levels of 60 bhp per liter of piston swept volume (Ref. 12), this factor should not limit the design of the engines currently envisioned.

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